Improving the performance of wooden journal bearings

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Abstract

Journal bearings made of wood have long been used in traditional devices such as oxcarts and waterwheels. They are currently used in industrial materials-handling equipment and other specialized applications. Wooden bearings have potential use in developing regions in "appropriate technology" devices such as animal-drawn carts and manual water pumps. This pilot study examined a number of factors that affect wooden bearing performance, including wood properties, fabrication methods, and operating conditions. A testing procedure for wooden bearings was established, and a testing machine was constructed. Bearings were fabricated of different wood species, treated with various lubricants, and run on the machine under varying conditions of load and speed. Measurements were made of bearing and axle wear, static and dynamic friction, and bearing operating temperature. The results of this study show that lubricant characteristics, speed and load of operation, steel axle properties, and wood density and permeability can have a considerable effect on bearing performance.

The journal bearing is a fundamental mechanical component that supports and positions an object while allowing that object to rotate. Bearings made of wood have long been used in cartwheels, windmills, lathes, and other technical devices (Mumford 1934). Archeological evidence shows that wooden wheels and bearings were first used in the Tigris-Euphrates valley circa 3500 BC (Jenkins 1981). These solid cart wheels were crafted of flat planks and rotated on fixed wooden poles. Evolving wood-shaping sophistication in the ensuing centuries led to lighter spoked wheels by the second millennium BC (Piggott 1983). Bearing lubricant has been used since at least 1400 BC, which is the date of a cart axle anointed with animal tallow that was found in an Egyptian tomb (Davison 1957).

Metals were later introduced into bearing designs, but wooden bearings continued to be used in many applications such as water wheels, grain mills, construction cranes, and military machines such as catapults. Leonardo da Vinci studied the friction and wear of bearings in the 15th century (Reti 1971), and the recent industrial revolution sparked great advances in bearing technology. Wooden bearings were widely used for power transmission shafts in factories during the early industrial revolution. Farm machinery such as harvesters and threshers employed wooden bearings (Martin 1892), continuing into the 20th century. Wood known as lignum vitae, from *Guaiacum* spp. trees grown in tropical America, has been used for centuries in rudder- and propeller-shaft bearings on boats and ships. It performs very well, however, the slow-growing tree is increasingly scarce and lignum vitae is rarely used in new applications (Steuernagle 2001).

In Russia, bearings made of compressed wood have been used successfully since the 1940s in certain industrial applications including water pumps on the Moscow Canal (Apostol and Yanin 1990) and roller veneer dryers (Lazarev 1991). In production, deciduous wood is first plasticized by immersion in steam or hot machine oil, compressed under high temperature and pressure to about half its original volume, then machined to shape using metal-working equipment. These bearings reportedly performed with low friction and wear, and had high resistance to abrasive particles.

Wooden journal bearings continue to be used in specialized industrial applications in North America (Steuernagle 2001). At least three companies currently manufacture wooden bearings in the United States with total annual sales estimated at \$2 to 2.5 million (Steuernagle 2002). The bearings are used in

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screw-auger conveyers to transport bulk materials, because of the bearings' ability to work well under dry abrasive conditions with irregular lubrication. Wood is also used in roll-end bearings for roller conveyers and other specialty applications, where their low cost is an advantage. These bearings are generally made of hard maple wood (*Acer saccharum,* 'rock maple') saturated with wax or oil lubricant, and compete successfully with a wide range of other bearing materials including metals, plastics and ceramics. Accumulated experience has led to design guidelines regarding radial clearance, grain direction, load-speed relationships, and other operational factors (Steuernagle 2001, Anon. 1977).

Wooden bearings have strong potential for "appropriate technology" applications in engineered devices designed to be constructed, used, and maintained by rural populations in developing countries. The advantages of wooden bearings over other types of bearings in these applications include their low cost, relative ease of fabrication, and local access to required materials. The "Bush Pump" manual water pump originating in Zimbabwe is a successful example of this use (Erpf 1998). The pump employs a wooden fulcrum bearing that is inexpensive, durable, and can be replaced by a local carpenter when required.

In recent decades, several projects have been implemented in sub-Saharan Africa to develop improved appropriate technology animal-drawn carts using wooden bearings (Starkey 1989). More widespread use of animal-drawn carts could improve transport links and reduce the burden currently carried by people, principally women. The greatest challenge in cart design is the wheel/axle assembly, including bearings, the cost of which may constitute 40 to 70 percent of the production cost of a cart (Dennis and Anderson 1994). The use of wooden bearings can reduce the cost and production effort of carts, thus increasing their accessibility to rural populations; however, wooden bearing performance in these projects has been erratic and unacceptably high friction and wear was sometimes exhibited. The motivation for the present study was the need to deepen our understanding of the general principles of wooden bearings, a resource potential for local efficiency and global equity.

Current knowledge of wooden bearing behavior is informed by a limited set of sources. Descriptions of historical usage contain empirical solutions from the past that may suggest leads for further study, but may not be directly applicable to current bearing design. Modern industrial bearing producers have accumulated application-specific experience that may not be transferable to other uses or geographic regions in which the wood species, lubricant type, or other factors may be different. Soviet researchers have apparently studied wooden sliding bearings in considerable depth, although little of that work is currently available in English and the translation of additional publications (e.g., Denisenko 1962) could provide potentially useful information. Rural development projects in Africa to develop improved animal-drawn carts have produced comprehensive guidelines for overall cart and harness design (Barwell and Hathway 1986, Dennis 1997), but information specific to the performance of wooden bearings is limited and sometimes contradictory (e.g., Collett 1976, Wirth 1992).

Despite the long and continuing use of wooden bearings, this topic has benefited from very little formal study. The available literature contains few studies directly and quantitatively comparing the effects of basic wood properties, fabrication methods, and operating conditions on wooden bearing behavior. Little is understood about the factors that affect the performance of wooden journal bearings and how those factors might be optimized to improve bearing performance. Thus, the objectives of this pilot study were to 1) develop a testing procedure for wooden bearings; and 2) identify wood properties, fabrication methods, and operating conditions that have a major effect on the performance of wooden bearings.

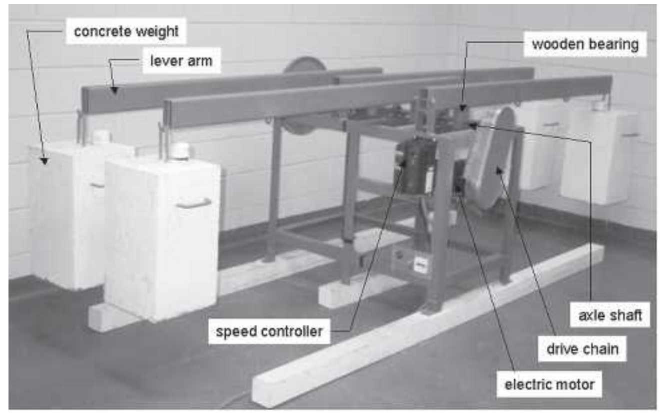

Figure 1. — Bearing testing machine, capable of testing four bearings at a time.

Materials and methods

Test machine and bearings

A testing machine was constructed to subject wooden bearings to high load while rotating at low to moderate speed (**Fig. 1**). It consisted of a steel frame upon which a 19.05-mm- (nominal 0.75-in-) diameter steel shaft was supported by ball-bearing pillow blocks. The shaft was rotated by a DC electric motor through a drive chain mechanism. Four wooden bearings at a time were placed on the axle shaft. Downward force was applied to the bearings by weights suspended from four lever arms attached to the machine frame. The rotational speed was varied by an electronic speed controller and by changing drive-chain sprocket size. The load force on the bearings was adjusted by varying weight suspended from the lever arms.

A series of block bearings was made of different wood types and treated with different lubricants. The wood was first cut roughly to size and allowed to air-dry to 6 to 9 percent moisture content (MC), then cut and planed to final bearing size. The wood grain was oriented such that the axis of the steel axle ran in the radial grain direction and the load force was applied in the longitudinal grain direction. Holes for the axle shaft were bored in the bearing blocks with a 19.05-mm (nominal 0.75-in) Forstner bit. Wood density and MC were calculated using air-dry and ovendry weights of hole boring chips.

The air-dried bearings were then treated with various lubricants including olive oil, motor oil, beeswax, mineral oil, peanut oil, axle grease, pork tallow, petrolatum wax, and graphite. The treatment method depended on lubricant properties. For most liquid lubricants, the wooden bearing was submerged in lubricant, heated at 70°C for 1 hour, subjected to a vacuum of 600 mm Hg for 15 minutes, then maintained submerged in the lubricant for 24 hours. Waxes and tallow, solid at room temperature, were melted before vacuum treatment. Powdered graphite was applied to the wooden bearing surface as a paste mixed with either water or peanut oil. Grease was applied to the bearing surface. Further details on lubricant treatment and testing methods in general are described by Sathre (2002). The bearings were weighed before and after treatment to determine lubricant retention. The holes for the axle shaft were rebored with a Forstner bit after lubricant treatment.

Performance indicators

A series of independent experiments was conducted. The independent variables studied were wood density, wood permeability, load stress and sliding speed, lubricant type and load stress, and steel axle type. The dependent variables measured were static rotational friction, dynamic rotational friction, wear of bearing material, wear of shaft material, and operating temperature of the bearing material. Bearings were considered to have better performance when they had lower measurements for static and dynamic friction, bearing and axle wear, and operating temperature.

Static rotational friction was determined by measuring the moment required to induce rotation of the shaft. Testing one bearing at a time, weight was incrementally suspended from a cord wound around a pulley on the axle. When the force exerted by gravity on the weight, acting upon a moment arm equal to the radius of the pulley, produced a rotational moment that exceeded the static friction between the bearing and the shaft, the axle would rotate and the cord would unwind. Repeated at several intervals around the axle shaft, the average weight required to produce rotation allowed calculation of the coefficient of static friction. Correction was made for the friction of the ball-bearing pillow blocks supporting the axle shaft.

Dynamic rotational friction was similarly measured; however, after each increment of weight was added to the fixture, the pulley was "nudged" slightly by hand to overcome the static frictional forces. The weight that allowed the unwinding of the pulley on its own similarly allowed calculation of the coefficient of dynamic friction.

Wear of the bearing material was measured with a digital caliper accurate to 0.01 mm. The distance was measured between the axle surface and a steel pin inserted in each bearing block during fabrication, the distance becoming less and less as the bearing wore. Measurements of both sides of the bearing were averaged to account for asymmetrical wear.

Wear of the steel axle was measured with a digital caliper accurate to 0.01 mm. The shaft diameter was measured at the beginning and end of each bearing test. A new steel axle shaft was installed on the test machine with each new set of bearings. Wear of the axle shaft must, in general, remain very low, and a bearing material that caused high shaft wear would be considered a failure. No measurable axle wear was encountered in this study.

The operating temperature of the bearing was found by inserting the probe of an electronic thermometer, accurate to 0.1°C, into a hole in the bearing material. Measurements were taken with the testing machine in operation and the bearing under load. Ambient air temperature was measured with the same thermometer to allow calculation of temperature rise of bearing over ambient. Because heat is from friction that in a well performing bearing will remain low, cooler operating

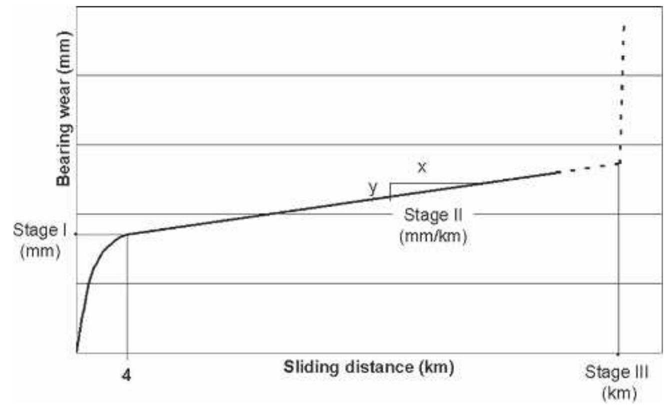

Figure 2. — Typical bearing wear pattern and method of quantification.

temperatures were generally considered an indication of better bearing performance. Limited temperature increase may be beneficial, for example, to melt wax lubricant, but excessive heat will cause wood damage.

Results and discussion

Wear pattern

Bearing wear was plotted as mm of wear per km of sliding distance (sliding distance = revolutions × axle circumference). Wear of most wooden bearings was observed to follow three distinct stages. This pattern and its quantification are shown in **Figure 2**. Wear rates were high during an initial run-in period (Stage I) of up to 4 km sliding distance. Wear rates then declined and stabilized during a linear wear period (Stage II). Later, some bearings failed suddenly either by splitting or burning (Stage III). These three stages were quantified by fitting a least squares regression line to the wear measurement points between the initial run-in period and failure. The slope of the regression line is the parameter of Stage II linear wear rate (mm/km). The y value of the regression line at $x = 4$ km is the parameter of Stage I initial run-in amount (mm). The x value when failure occurs is the parameter of Stage III sliding distance to failure (km).

Wood density and permeability

Bearings were fabricated of maple (*Acer saccharum*) ($n = 5$) and basswood (*Tilia americana*) ($n = 4$), two woods sharing similar anatomical characteristics including diffuse pore distribution and high permeability. Differentially, the specific gravity (SG) of the maple measured 0.72, twice the 0.36 of the basswood. The bearings were identically treated with olive oil, the maple retaining a mean 48 percent lubricant (weight of lubricant per weight of air-dried wood) and basswood retaining 155 percent. The bearings were tested to 42 km sliding distance at identical speed and load levels. Bearing wear was recorded, and is plotted in **Figure 3**. During Stage I initial run-in, the basswood bearings had significantly greater wear than the maple bearings [$t(7) = 5.90, p < 0.05$]; but after reaching Stage II, the basswood bearings had a mean linear wear rate not significantly different from the maple bearings [$t(7) = 1.18, p > 0.05$]. Moreover, variation in wear between individual basswood bearings was greater than the variation between maple bearings (**Fig. 3**). None of the bearings reached Stage III failure.

Limited comparison was made between maple bearings ($n = 5$) and muninga (*Pterocarpus angolensis*) ($n = 2$), two

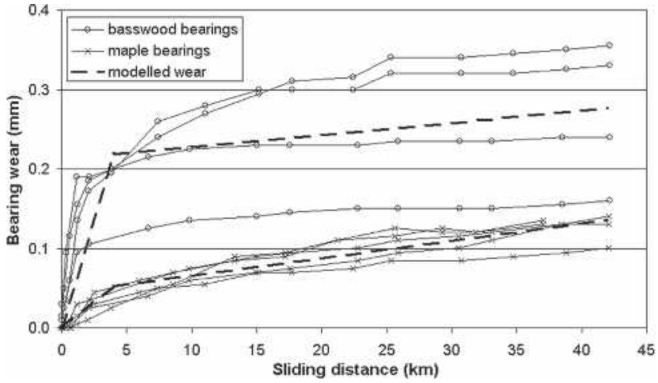

Figure 3. — Wear of bearings made of woods of different density.

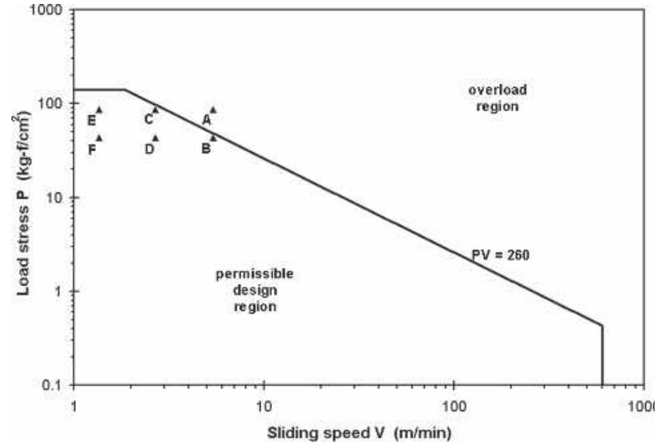

Figure 4. — Load-speed (PV) relationship of wooden bearings (from literature), with test conditions of this study indicated by letters A-F.

woods of similar density (maple = 0.72 SG, muninga = 0.62 SG) and diffuse pore distribution. After treatment with olive oil using the vacuum method described previously, the maple bearings retained a mean 48 percent of their air-dry weight in lubricant while the muninga bearings retained only 4.2 percent lubricant. On the testing machine at identical operating speed and load, the muninga bearings experienced Stage I run-in amounts averaging 3.1 times greater than the maple bearings and Stage II linear wear rates averaging 2.6 times greater than the maple bearings. One muninga bearing failed and no maple bearings failed. The operating temperature of the muninga bearings averaged 12.5°C greater than the maple bearings.

The wood matrix in a prepared wooden bearing has at least two functions: 1) it must provide physical support for the load imposed on it; 2) it must act as a vehicle for the bearing lubricant. To fulfill the first function, the wood structure must be sufficiently robust to withstand the compressive forces and retain its form under load. This requires a certain density of wood matter to provide the needed mechanical strength, as woods of higher density generally have higher compressive strength (Panshin and de Zeeuw 1980). However, to satisfy the second function, the wood must have sufficient porosity to contain and conduct the lubricant. Thus these two functions are contradictory. A very dense wood will provide high mechanical strength but have low lubricant capacity, while a wood of low density may contain a large quantity of lubricant but have inadequate physical strength.

In this context, a distinction must be made between the porosity and the permeability of a wood. Wood may be highly porous, but because of pit aspiration, tyloses, or other impediments to flow, have low permeability and not allow the movement of lubricant through its structure. Such a wood would provide neither high mechanical strength nor high lubricant capacity.

Experimental results suggest that wood permeability may be more important to bearing longevity than is wood density. Wood with high permeability and low density (basswood) had greater Stage I run-in than wood of high permeability and high density (maple); however, the Stage II linear wear rates of the two woods were comparable. Wood with low permeability and high density (muninga) experienced both high run-in amounts and high linear wear rates. Because Stage I initial run-in occurs a single time in the life of a bearing, while Stage II linear wear rate continues throughout the working lifespan of the bearing and ultimately determines its failure due to "wearing out," bearing longevity may depend more on wood permeability than on wood density.

Load and speed

Lancaster (1978) explained that the maximum load a journal bearing can support is generally limited by the compressive strength of the bearing material, while the maximum speed is determined by the ability of the bearing assembly to dissipate heat generated by friction. A "PV index" is commonly used to rate journal bearing materials for their suitability for use under different operating conditions. The product of loading stress (P) and sliding velocity (V) should not exceed a designated value that varies with material. Limiting individual values for maximum P and for maximum V are also specified. For wooden bearings, with the loading stress P expressed in units of kg-f/cm^2 and sliding velocity V in units of m/min, the PV_{max} index found in the literature ranges from 260 to 320, with individual P_{max} of 140 and V_{max} of 600 (Wilcock and Booser 1957, Anon. 1977, Steuernagle 2001). This range of values is plotted on logarithmic axes in **Figure 4**.

In this experiment, 12 bearings were prepared of maple wood and treated with olive oil. Two bearings were tested at each of six combinations of load and speed. Levels of sliding speed were low (1.35 m/min), medium (2.69 m/min), and high (5.39 m/min), and load stress levels were low (43.1 kg-f/cm^2) and high (86.1 kg-f/cm^2). The load and speed conditions of the bearings tested are shown on **Figure 4** as points A, B, C, D, E, and F. **Table 1** summarizes the measured Stage I run-in amount and Stage II linear wear rates for the bearings tested.

Failure occurred in both bearings tested at high load and high speed (point A). Note that this operational condition is outside the permissible design region because the product of speed (86.1 kg-f/cm^2) and load (5.39 m/min) exceeds the illustrated PV_{max} of 260. The failure of both bearings tested under these conditions of speed and load supports the literature's suggested PV_{max} range of 260 to 320.

None of the other bearings failed during the tests, although their wear amounts and rates were not linear with changes in load and speed. Results suggest that load has a greater effect on bearing wear than does speed. Sets of bearings were tested at differing loads and speeds, but with the product of load and

Table 1. — Wear bearings operated under different load and speed conditions.[a]

Sliding velocity (m/min)	Load stress ($kg\text{-}f/cm^2$) 43.1	86.1
Stage I, initial run-in (mm)		
1.35	F **0.0687**	E **0.0969**
	(0.0047)	(0.0181)
2.69	D **0.0358**	C **0.0819**
	(0.0171)	(0.0030)
5.39	B **0.0476**	A **failed**
	(0.0073)	(n.a.)
Stage II, linear wear rate (mm/km)		
1.35	F **0.00112**	E **0.00494**
	(0.00004)	(0.00180)
2.69	D **0.00178**	C **0.00748**
	(0.00006)	(0.00209)
5.39	B **0.00274**	A **failed**
	(0.00028)	(n.a.)

[a]Boldface values are sample means; values in parentheses are standard deviations.

speed resulting in constant PV values. Bearings at points B and C all had total PV values of 232; bearings at point B operated at twice the speed and half the load of bearings at point C. Similarly, bearings at points D and E all had total PV values of 116. In all cases, wear was greater in bearings operated at higher load and lower speed. Bearings at points B and D, corresponding to conditions of higher load and lower speed, averaged 2.1 times greater Stage I run-in amounts and 2.8 times greater Stage II linear wear rates (**Table 1**) than bearings tested at points C and D under lower load and higher speed.

Axle properties

Tests investigated the effect of steel axle properties on wooden bearing performance. Maple bearings treated with a variety of lubricants were tested on two types of steel shafts, cold-drawn 1018 mild steel and ground-and-polished 4140 heat-treated steel. Bearing life was remarkably shorter when run on the ground and polished heat-treated axles. **Figure 5** shows wear curves for these bearings, to be compared with **Figure 3** showing typical wear of bearings run on mild steel axles, noting differences in plot scales. Of six lubricated maple bearings tested on heat-treated shafts, most failed within 0.15 km sliding distance and the longest lasting failed at 0.65 km. Conversely, of 20 lubricated maple bearings tested on mild steel axles under identical load and speed conditions (bearings lubricated with dry graphite not included), the earliest bearing failure occurred at 5.8 km and many bearings continued operating past 30 km. Coefficients of dynamic friction with the heat-treated shafts were 17 to 42 percent higher than those for the same lubricant with a mild steel shaft. Operating temperatures were much higher with the heat-treated shafts, and all of the bearings failed by burning. No measurable axle wear was noted in any test.

It is likely that the surface properties of the steel shafts account for the difference in wear rather than the metallurgical properties of the different metals. Photomicrographs of the surfaces of the axles (Sathre 2002) show parallel ridges on the heat-treated steel caused by the grinding and polishing process, while the surface of the cold-drawn mild steel axle had a more mottled appearance. The ridges on the heat-treated shaft ran in the direction of rotation and may have dug into the wood surface and prevented any accumulation of lubricant or wood debris in the interface. The irregular topography of the cold-drawn steel surface possibly allowed lubricant or debris to build up between the sliding surfaces, reducing wear.

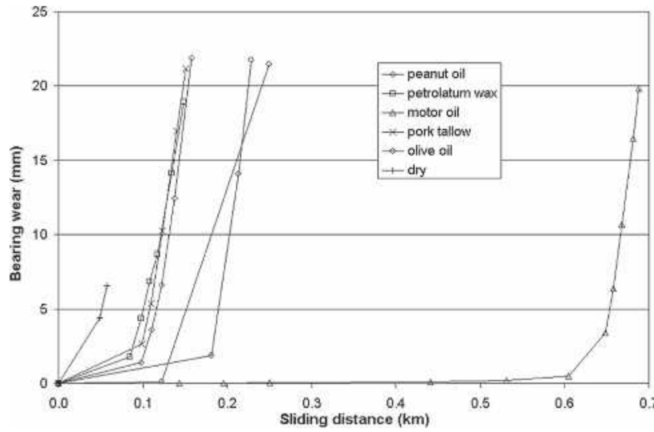

Figure 5. — Wear of bearings run on ground-and-polished heat-treated steel axles.

Static friction

Tests were conducted to determine the coefficient of static friction at three load stress levels for maple bearings treated with eight types of lubricant. Static friction of dry bearings without lubrication was also measured. Two bearings treated with each of the following lubricants were tested: peanut oil, olive oil, motor oil, mineral oil, petrolatum wax, beeswax, axle grease, pork tallow. Three dry unlubricated maple bearings were also tested.

As shown in **Figure 6**, coefficients of static friction ranged from 0.11 to 0.48. There was little difference in static friction between dry bearings and some of the worst performing lubricants. Coefficients of friction were often, but not always, slightly lower at higher load stress levels. Bearings lubricated with beeswax had the lowest friction. There was no apparent difference in friction between lubricants that are liquid at room temperature (peanut oil, olive oil, motor oil, mineral oil) and those that are solid (pork tallow, beeswax, petrolatum wax, axle grease). There was also no apparent difference between petroleum-based lubricants as a group (motor oil, mineral oil, petrolatum wax, and axle grease) and animal- and vegetable-based lubricants (olive oil, peanut oil, pork tallow, beeswax).

Dynamic friction

The same bearings and lubricants tested for static friction were also tested for dynamic friction. As shown in **Figure 7**, there was a large difference in dynamic friction between dry and lubricated bearings. Petroleum-based lubricants, as a group, had significantly higher friction than animal- and vegetable-based lubricants at all three load levels tested: low, medium, and high [Fs $(1, 14) = 33.17$, 59.90, and 103.30, respectively, $p < 0.05$]. There was no significant difference between solid and liquid lubricants at any of the three load levels tested: low, medium, and high [Fs $(1, 14) = 0.13$, 0.11, and 0.82, respectively, $p > 0.05$]. All lubricants tested showed a slight downward trend in coefficient of dynamic friction as

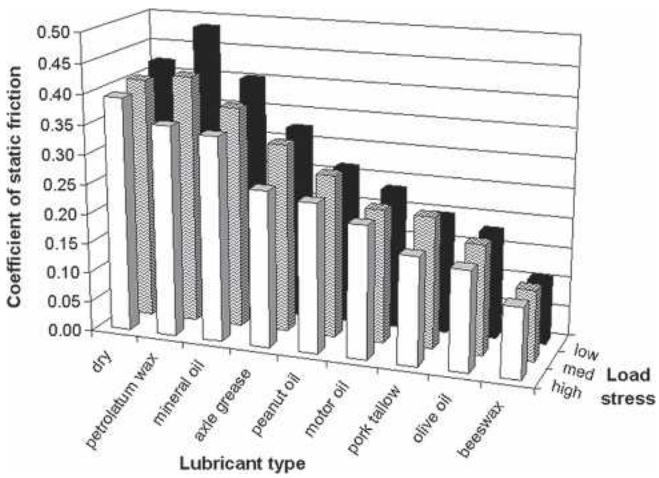

Figure 6. — Static friction of nine different lubricants and three load stress levels.

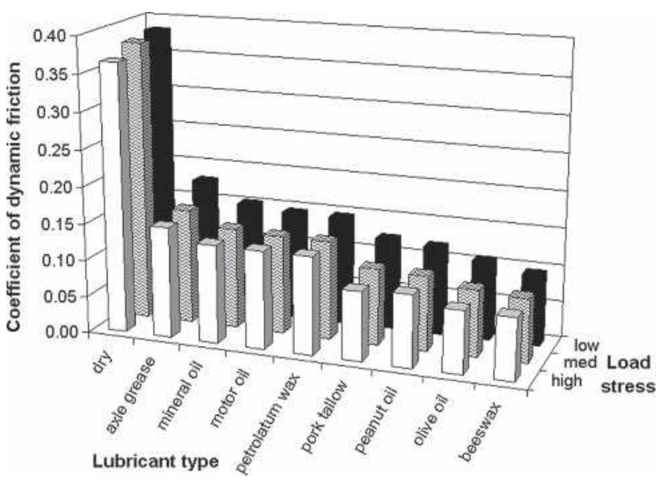

Figure 7. — Dynamic friction of nine different lubricants and three load stress levels.

the load level increased. Bearings lubricated with beeswax had the lowest friction levels of all lubricants tested.

Dynamic friction of the petroleum-based lubricants was in all cases greater than that of the animal- and vegetable-based lubricants. No difference in dynamic friction was noted between solid and liquid lubricants. These phenomena suggest that the bearings operate in the boundary lubrication regime, in which intimate physical contact is made between the bearing and shaft, rather than the hydrodynamic regime in which the two surfaces are separated by a layer of oil. Friction level in the boundary regime is primarily determined by the chemical makeup of the lubricant, while in the hydrodynamic regime the friction depends on the viscosity of the lubricant (Moore 1975). Animal- and vegetable-based lubricants are composed of polar fatty acid molecules that attach strongly to the sliding surfaces, forming layers that slide against each other with lower friction. Nonpolar petroleum-based lubricant molecules cannot attach strongly to the surfaces and do not form a low-friction layer.

Limited tests were conducted using powdered graphite as lubricant, in paste form mixed with water or peanut oil. The presence of graphite produced no apparent reduction in friction or wear. A single maple bearing lubricated with dry powdered graphite exhibited high friction and a very short lifespan (coefficient of dynamic friction > 0.185, Stage III failure at 0.06 km sliding distance), similar to an unlubricated maple bearing tested under the same load and speed conditions (coeff. of dyn. friction > 0.185, Stage III failure at 0.12 km sliding distance). Two maple bearings lubricated with powdered graphite mixed with peanut oil showed fairly low friction and wear rates (1: coeff. of dyn. friction = 0.093, Stage I wear = 0.152 mm, Stage II wear rate = 0.0102 mm/km; 2: coeff. of dyn. friction = 0.090, Stage I and II wear data not available), similar to two maple bearings lubricated with peanut oil alone (1: coeff. of dyn. friction = 0.094, Stage I wear = 0.148 mm, Stage II wear rate = 0.0038 mm/km; 2: coeff. of dyn. friction = 0.086, Stage I wear = 0.214 mm, Stage II wear rate = 0.0114 mm/km). Thus the addition of dry graphite appeared to bring no noticeable improvement to dry wood, and graphite plus oil was no better than oil alone.

The poor performance of graphite as a wooden bearing lubricant may possibly be explained at the interface between the graphite crystals and the wood cell structure. Graphite in theory is a good boundary lubricant because it has a crystal lattice structure composed of layers (Ellis 1970). As the graphite is compressed between two sliding metal surfaces, it shears relatively easily between layers resulting in low sliding friction. In a wooden bearing, however, the compressive strength of the wood is roughly the same as that of the graphite crystal, so that the graphite crystals may become embedded in the wood rather than shear between the wooden bearing and steel axle. The compressive strength of graphite is about 28,000 to 55,000 kPa (Ellis 1970), while the compressive strength parallel to the grain of air-dried maple wood is about 36,000 to 54,000 kPa (FPL 1999).

Conclusion

Wooden bearings have a long history of use, continue to be used successfully in various applications, and have potential for more widespread use. Many factors affect bearing behavior, few of which are well understood and documented. The present pilot study examined several wood properties, fabrication methods, and operating conditions in an effort to deepen our understanding of the factors that influence bearing performance. The results confirm that under certain conditions wooden bearings can operate with reasonably low levels of friction and wear. The study has also shown that under unsuitable conditions wooden bearings can exhibit high friction and extreme wear.

Specifically, the study results indicate that:

- Higher wood density gave lower wear;
- Higher wood permeability gave lower wear;
- Permeability may be more important than density to bearing longevity;
- Within the range of loads and speeds tested, load stress level had a greater effect on wear than did speed level;
- The maximum PV index from the literature was confirmed;
- Extremely high wear rates occurred with ground-and-polished heat-treated steel axles;
- Petroleum-based lubricants gave higher dynamic friction than animal- and vegetable-based lubricants;
- Lubricant viscosity had no apparent effect on friction;
- Wooden bearings appear to operate in the boundary lubrication regime;

- Beeswax lubricant gave the lowest static and dynamic friction levels;
- The coefficients of friction decreased slightly at higher load stress levels.

This study is offered as a step toward illuminating several factors that influence wooden bearing behavior. Additional work is needed, however, not only to confirm and expand on these findings but also to address additional wood characteristics, fabrication methods, and operating conditions affecting wooden bearing behavior that could not be feasibly addressed in this investigation.

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