

IMPROVING THE PERFORMANCE OF HARDWOOD JOURNAL BEARINGS

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AUTHORIZATION TO SUBMIT THESIS

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**Abstract:**

Journal bearings made of wood have been used for centuries in traditional devices such as carts and waterwheels. They are currently used in industrial

materials handling equipment and other specialized applications. Wooden bearings have potential for use in “appropriate technology” applications in developing countries such as animal-drawn carts and manual water pumps. This pilot study examines a number of factors that affect hardwood 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, steel axle properties, speed and load of operation, and density, permeability, and pore distribution of wood bearing material can have considerable effect on bearing performance.

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This work is dedicated to my son Savana, to whom I hope this information will someday be useful.

## Table of Contents

Title Page		i
Authorization to Submit Thesis	ii	
Abstract		iii
Acknowledgements		iv
Table of Contents		v
List of Figures	vii	
List of Graphs	viii	
<b>1. Introduction</b>		<b>1</b>
1.1 Classification of bearings	1	
1.2 History of wooden bearings	1	
1.3 Current applications of wooden bearings	3	
1.4 Factors affecting bearing performance	4	
<b>2. Objectives and Methods</b>		<b>7</b>
2.1 Project objectives	7	
2.2 Testing machine	7	
2.3 Fabrication of test bearings	8	
2.4 Experimental design	9	
2.5 Bearing performance		10
2.5.1 Static rotational friction	10	
2.5.2 Dynamic rotational friction	11	
2.5.3 Bearing wear		12
2.5.4 Shaft wear		12
2.5.5 Bearing operating temperature	13	
<b>3. Experimental Results</b>		<b>14</b>
3.1 Wear model	14	
3.2 Wood density		15
3.3 Load and speed		15
3.4 Wood permeability	16	
3.5 Wood pore size and distribution	17	
3.6 Axle properties		18
3.7 Static friction		19
3.8 Dynamic friction		20
3.9 Wear vs. friction		21

<b>4. Discussion</b>		<b>23</b>
4.1 Lubricants	23	
4.2 Load and speed		24
4.3 Wood properties	26	
4.4 Axle properties		28

## **5. Conclusion**

30

References		32
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Appendix 1: Dimensions of test bearings	34
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Appendix 2: Summary of bearings tested	35
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Appendix 3: Test data	37
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### List of Figures

Figure 1. Factors potentially affecting hardwood bearing performance	5
Figure 2. Wood bearing testing machine	7
Figure 3. Summary of experimental conditions	9
Figure 4. Diagram of friction test setup	11
Figure 5. Measuring wear of wooden bearing	12
Figure 6. Bearing wear model showing parameters of three stages of wear	14
Figure 7. P-V diagram showing load and speed of bearings tested	25
Figure 8. Magnified images (24x) of surfaces of wooden bearings	27
Figure 9. Magnified images (48x) of surfaces of steel axles	29

### List of Graphs

Graph 1. Bearing wear vs. sliding distance for woods of different density	15
Graph 2. Stage I run-in amounts for bearings tested at different loads and speeds	16
Graph 3. Stage II linear wear rates for bearings tested at different loads and speeds	17
Graph 4. Wear vs. sliding distance for bearings run on heat-treated steel axles	18
Graph 5. Static friction of eight lubricants at three load stress levels	20
Graph 6. Dynamic friction of eight lubricants at three load stress levels	21
Graph 7. Bearing wear rates vs. dynamic friction for various lubricants	22

## **1. Introduction**

## **1.1 Classification of bearings**

A bearing is a mechanical component that supports and positions an object while allowing that object to rotate. Bearings are an integral part of commonly used devices such as pulleys, wheels, motors, pumps, and most other machines that have rotating parts. Bearings can be classified according to the direction of the loading that they support: journal bearings support loads perpendicular to the axis of rotation, while thrust bearings support loads parallel to that axis. A journal bearing can take the form of a fixed bearing, inside of which rotates a shaft (termed the “journal”), or a fixed shaft, around which rotates the bearing (usually termed the “bushing”).

Bearings can further be classified according to the nature of contact between the two surfaces: sliding bearings and rolling element bearings. In a sliding bearing there is intimate contact between the fixed and rotating surfaces. The sliding surfaces can be in direct contact (known as boundary condition) or can be separated by a thin film of oil or other lubricant (hydrodynamic condition). The bearings supporting the crankshaft of an internal combustion engine are examples of sliding bearings. In a rolling element bearing the surfaces are separated by a number of round (spherical, cylindrical, or tapered) elements, as for example the ball bearings of a bicycle wheel.

## **1.2 History of wooden bearings**

Sliding journal bearings made of wood have been used for thousands of years, although today they are much less common than bearings made of metal or plastic. Wooden wheels and bearings were first used in the Tigris-Euphrates valley around 3500 BC, and spread throughout much of Europe and the Near East in the centuries that followed (Jenkins 1981). These solid cart wheels were crafted of planks of wood split from logs. Generally the wheels rotated on wooden poles fixed to the cart frame. In some instances a pair of wheels was fixed to a pole that rotated in notches cut in the cart frame.

Cart and wheel technology gradually became more sophisticated while continuing to employ wooden bearings. By the second millennium BC lighter spoked wheels were developed and began to replace solid wheels in some applications, particularly in “chariots” used by soldiers and rulers (Piggott 1992). Solid wheels continued to be used in other applications, principally farm wagons carrying heavy loads. Bearing lubricant also began to be used at this time, the earliest known example being animal tallow encountered on a cart axle found in an Egyptian tomb from 1400 BC (Davison 1957).

As human metalworking skills increased, bronze, and later iron, was introduced into bearing designs. Wooden bearings continued to be used in many applications such as water wheels, grain mills, construction cranes, and military machines such as catapults. Dutch windmills constructed in the 16<sup>th</sup> and 17<sup>th</sup> centuries rotated on iron shafts supported by oak bearings lubricated with pork tallow (Molen van Sloten 2001). 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, continuing well into the 20<sup>th</sup> century.

Wood known as lignum vitae, from *Guaiaicum sp.* trees grown in tropical America, has been used for centuries in rudder- and propeller-shaft bearings on boats and ships. It performs very well, particularly when kept wet, and is the bearing of choice not only for ships but also water wheels and turbines. However, the slow-growing tree is increasingly scarce and lignum vitae is rarely used in new applications (Steuernagle 2001).

The industrial revolution sparked great advances in bearing technology during the 18<sup>th</sup> and 19<sup>th</sup> centuries, including the development of rolling-element bearings and metal alloys with low-friction properties. Of particular importance was the development of a low friction metal alloy by Charles Babbitt in 1839. Today, most bearings in industrialized countries are roller bearings made of hardened steel, or journal bearings made of metal alloys or plastics (Wilson 1986).

### 1.3 Current applications of wooden bearings

Wooden bearings, however, continue to be used in a number of modern applications. In the United States, wooden bearings are used in some industrial materials handling equipment (Steuernagle 2001). These bearings are generally made of hard maple wood (*Acer saccharum*, 'rock maple') and lubricated with waxes or oils. Screw conveyers used to transport bulk materials are often supported by wooden bearings, due to the bearings' ability to work well under dry, abrasive conditions with irregular lubrication. Wood is also used in roll-end bearings for roller conveyers on, for example, loading docks to unload boxes from trucks. In this application their relative low cost is an advantage over other bearing types, because of the large number of bearings used in a single conveyer. At least three companies currently manufacture wooden bearings in the United States, with total sales volume estimated at \$2 million per year (Steuernagle, pers. corr. 26 April 2002).

In Russia, bearings made of laminated wood have been used successfully in large water pumps in the Moscow Canal since the end of World War II (Lazarev 1991). Compressed wood has also been used successfully as a bearing material in these pumps. In this technique, poplar and birch wood is compressed under high temperature and pressure to about half of its original volume and then machined to shape using metal-working equipment. These wood bearings reportedly performed with low friction and high resistance to abrasive particles in the water. Compressed wood bearings have also been used successfully in roller veneer dryers (Apostol and Yanin 1990). These bearings were made by boiling and soaking soft deciduous wood in machine oil before pressing them to shape under high pressure. They reportedly worked well under high-temperature operating conditions and showed little wear.

In rural areas of many developing countries, wooden bearings continue to be used in traditional devices made by local craftsmen, including wheelbarrows, animal-drawn carts, and pulleys for lifting water from wells. These devices are generally made with simple hand tools from locally available materials.

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

Projects were implemented in several African countries during recent decades to develop improved “appropriate technology” ox-carts using wooden bearings. Greater use of animal-drawn carts could improve transport links and reduce the burden currently carried by people, principally women. However, the carts with wooden bearings suffered higher friction and wear and were not widely adopted by the population (Starkey 1989). The present study was motivated, in part, to develop an ox-cart bearing that is locally accessible and has adequate performance.

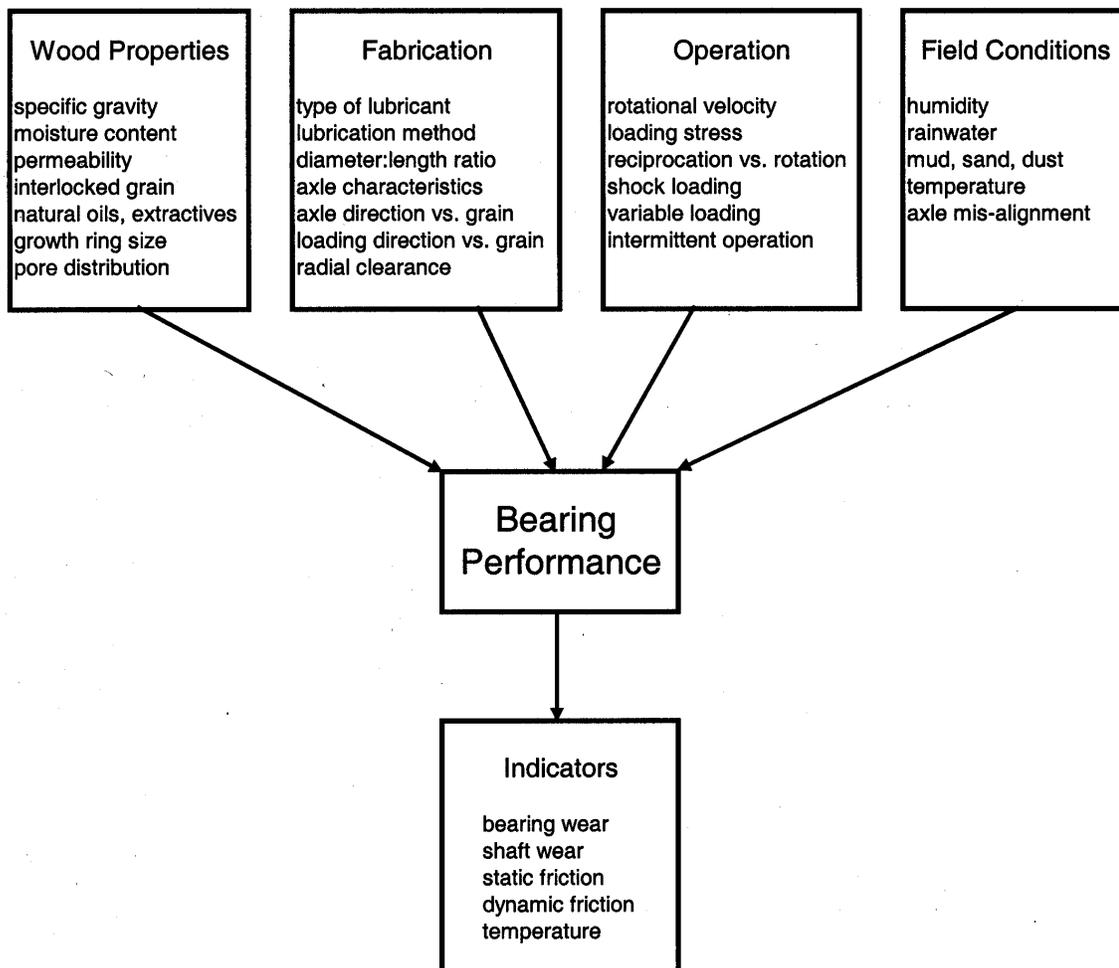
#### **1.4 Factors affecting bearing performance**

Preliminary consideration of wooden bearing usage suggests that many different factors might influence bearing performance. Figure 1 illustrates this range of factors, beginning with the basic properties of the wood from which the bearing is made. Some woods may be more suitable than others for use as bearings because of their density, permeability, extractive content, or other inherent characteristics.

Once a particular wood is chosen as a bearing material, different fabrication methods may also affect the performance of the resulting bearings. For example the bearing geometry, wood grain orientation, and type of lubricant may impact bearing behavior. Thus two bearings made from the same piece of wood may perform differently if their fabrication methods are different.

Furthermore, after a bearing is fabricated and put to use, factors related to its operating conditions may influence the bearing's performance. Loading stress, rotational velocity, and shock loading can be expected to influence its performance.

Finally, a group of factors more or less beyond the control of the designer may affect the behavior of the bearing. These include humidity, temperature, rain, dust, and other ambiantal conditions. While steps may be taken to protect the wooden bearing from some of these factors, it may be impractical to completely shield the bearing from all environmental forces.



**Figure 1.** Factors potentially affecting hardwood bearing performance

The mechanical designer who wishes to rationally incorporate wooden bearings in an engineered device may therefore be discouraged by the range of

factors that potentially affect bearing performance. This is complicated by the lack of information currently available on how those many factors actually influence bearing behavior. Despite the long and continuing use of wooden bearings, this topic has benefited from very little formal study. Sources of information on wooden bearing behavior are few and incomplete:

- Historical accounts from 16<sup>th</sup> century Holland (Molen van Sloten 2001) and 19<sup>th</sup> century United States (Martin 1892) contain local, anecdotal information on bearing manufacture that may suggest leads for further study, but is not directly applicable to current bearing design.
- Modern wooden bearing manufacturers in the United States rely on accumulated trial-and-error experience to ensure adequate bearing performance (Stearnagle, pers. corr. 19 March 2001). This application-specific experience may not be transferable to other uses or geographic regions in which the wood species, lubricant type, or other factors are different.
- Projects in Africa to develop improved animal-drawn carts produced comprehensive guidelines for overall cart and harness design (Dennis 1997), but information specific to the performance of wooden bearings is limited and sometimes contradictory. Collett (1976) described a method of wooden bearing manufacture, and Wirth (1992) made recommendations of suitable local wood types, though no comparative testing was recorded in the literature.
- Soviet researchers have studied wooden sliding bearings in considerable depth, although little of that work is currently available in English. The information available suggests that translation of additional publications (e.g. Denisenko 1962) could provide potentially useful information.

The available literature contains very few studies directly comparing the effects of basic wood properties, fabrication methods, and operating conditions on wooden bearing behavior. Very little is known about the factors that affect the performance of wooden journal bearings and how those factors might be optimized to improve bearing performance.

## 2 Objectives and Methods

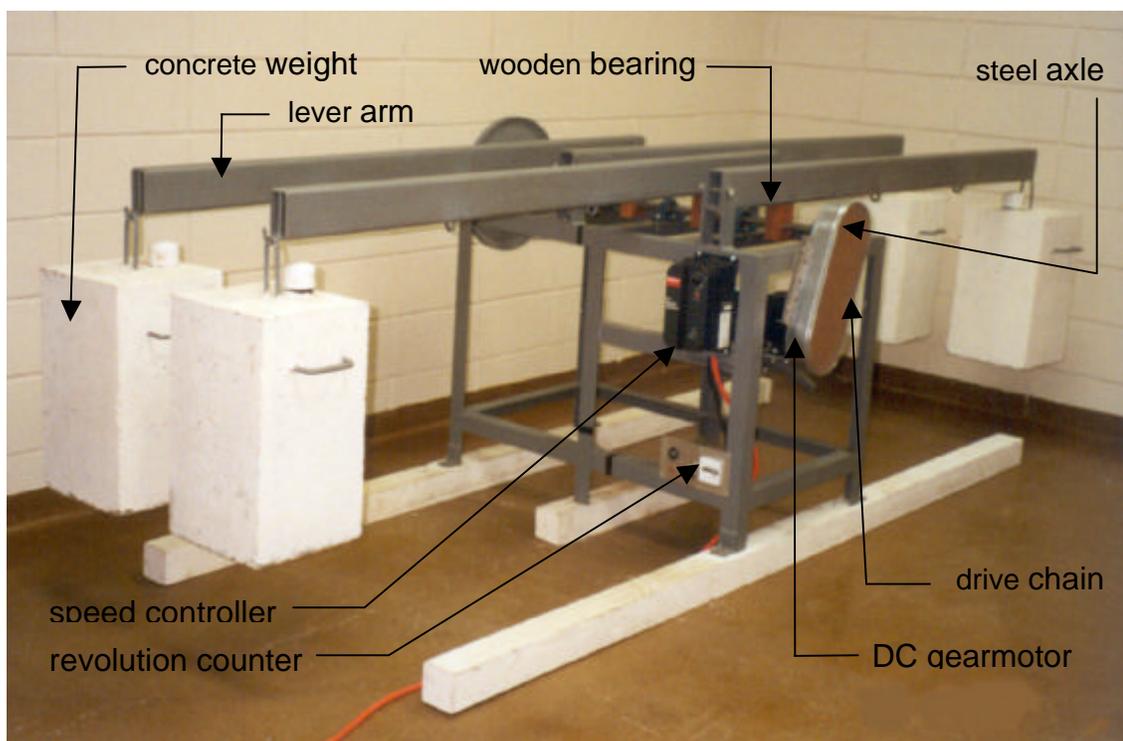
### 2.1 Project objectives

The objectives of this pilot study were:

1. to develop a testing procedure for wooden bearings,
2. to identify wood properties, fabrication methods, and operating conditions that have major effects on the performance of hardwood bearings, and
3. to the extent possible, identify methods to improve the performance of wooden bearings.

### 2.2 Testing machine

To fulfill these objectives, a testing machine was constructed to subject wooden bearings to high loads while rotating at low to moderate speeds. The machine is shown in Figure 2. It consisted of a steel frame upon which a 19.05 mm (0.75 inch) diameter steel shaft was supported by ball-bearing pillow blocks.



**Figure 2.** Wood bearing testing machine

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 lever arms attached to the machine frame. The rotational speed was adjusted with an electronic speed controller linked to the electric motor, and by changing the size of the drive-chain sprockets. The force on the bearings was adjusted by varying the weights suspended from the lever arms.

### **2.3 Fabrication of test bearings**

A series of bearings was made of different wood types and treated with different lubricants, and run on the testing machine at varying conditions of speed and load. To fabricate the bearings, the wood was first cut roughly to size and allowed to air-dry to 6-9% moisture content. The bearings were then cut and planed to final size. Dimensions of the test bearings are shown in Appendix 1. The wood grain was oriented such that the axis of the axle ran in the radial grain direction and the bearing load was applied in the longitudinal grain direction. Holes for the axle shaft were bored in the bearings with a 19.05 mm (0.75 inch) Forstner bit. To determine the density and moisture content of the wood, the bearings were weighed before and after boring the holes, and the chips from the holes were weighed before and after oven-drying.

The air-dried bearings were then treated with various lubricants. The exact treatment process depended upon the properties of the lubricant. For lubricants that were liquid at room temperature (peanut oil, olive oil, motor oil, mineral oil, liquid soap), the dry bearings were submerged in the lubricant, heated in an oven at 70°C for one hour, subjected to a vacuum of 600mm Hg for 15 minutes, then maintained submerged in the lubricant for 24 hours. For bearings treated with pork tallow and beeswax, the lubricant was melted in an oven before the bearings were introduced, then returned to the oven for one hour at 70°C, subjected to the vacuum treatment described above, and then returned to the oven for one hour at which time the bearings were removed from the lubricant. Bearings treated with petrolatum wax were sent to the Woodex Bearing

Company where they were submerged in heated liquid wax for one week. Bearings treated with axle grease were liberally coated with grease and allowed to stand for six weeks, the melting point of the grease being at a temperature that would have damaged the wood. For bearings treated with graphite, a paste of powdered graphite and water was applied to the inner surface of the bearing and allowed to dry. For bearings treated with graphite and peanut oil, a paste of powdered graphite and peanut oil was applied to the inner surface of the bearing, and the bearing was then vacuum-treated in peanut oil as described above. The bearings were weighed before and after treatment to determine lubricant retention. The holes for the axle shaft were re-bored with a Forstner bit after lubricant treatment.

## 2.4 Experimental design

This study consisted of a series of independent experiments. In each experiment, one or two factors of interest were varied while all other factors remained constant. As shown in Figure 3, the independent variables studied were wood density, wood permeability, wood pore distribution, load stress and sliding speed, lubricant type and load stress, and steel axle type. Further information on each bearing tested is contained in Appendix 2. Complete test data for all bearings are found in Appendix 3.

Independent variable(s)	Wood type	Lubricant	Speed (m/min)	Load (kg-f/cm <sup>2</sup> )	Axle type
wood density	maple/basswood	olive oil	5.39	43.1	mild steel
wood permeability	maple/muninga	olive oil	5.39	43.1	mild steel
wood pore distribution	maple/red oak	olive oil	5.39	43.1	mild steel
load stress, speed	maple	olive oil	1.35-5.39	43.1-86.1	mild steel
lubricant, load stress	maple	various	n.a.	21.5-86.1	mild steel
axle properties	maple	various	2.69	86.1	mild/treated

**Figure 3.** Summary of experimental conditions

## **2.5 Bearing performance**

In this study, “bearing performance” was defined by measurements of the following dependent variables:

1. static rotational friction
2. dynamic rotational friction
3. wear of bearing material
4. wear of shaft material
5. operating temperature of the bearing material

These measurements are operationally defined below.

### **2.5.1 Static rotational friction**

Static rotational friction was determined by measuring the moment required to induce rotation of the shaft. One bearing at a time was tested, with the axle shaft disconnected from the drive chain and motor. A cord was wound around a pulley on the shaft. Weights were incrementally added to a fixture on the end of the cord. When the force exerted by gravity on the weights, acting upon a moment arm equal to the radius of the pulley, produced a moment that exceeded the static friction between the bearing and the shaft, the shaft would rotate and the cord would unwind. A diagram of the test setup is shown in Figure 4. The weight required to produce this rotation was recorded. This procedure was repeated four times at quarter-rotation ( $90^\circ$ ) intervals around the axle shaft. The average weight required to produce rotation of the pulley was then calculated, and the coefficient of static friction was calculated based on this weight, the force exerted on the bearing by the lever arm, and the diameters of the shaft and pulley. The friction of the ball bearing pillow blocks supporting the axle shaft was measured separately, and a correction was made for this when calculating the friction of the wooden bearings. Bearings with lower static friction were considered to have better performance than those with higher friction.

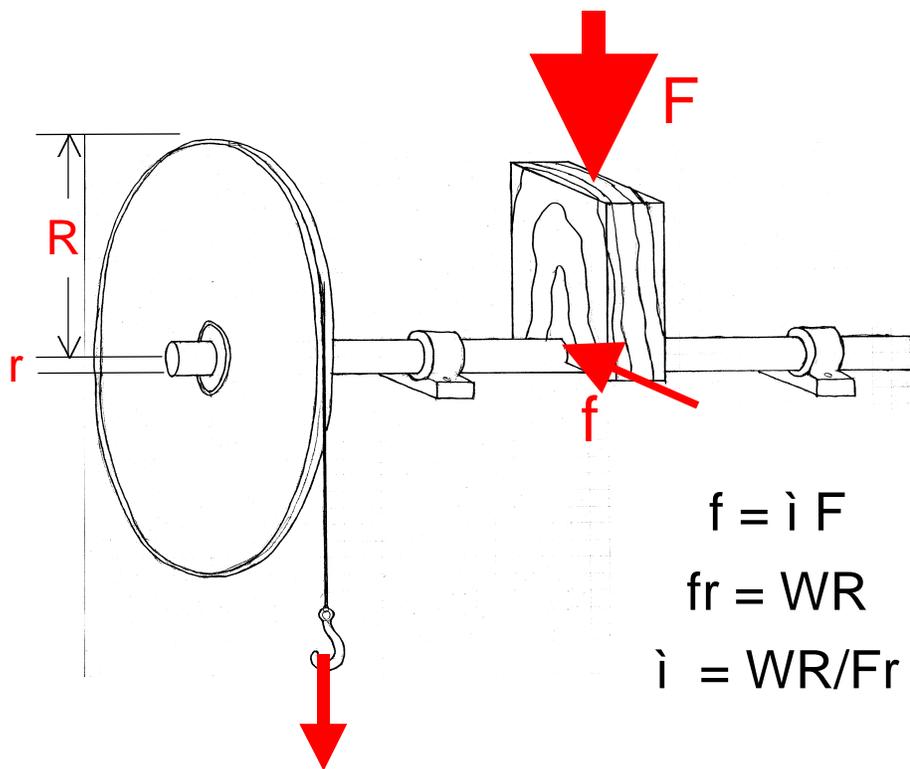


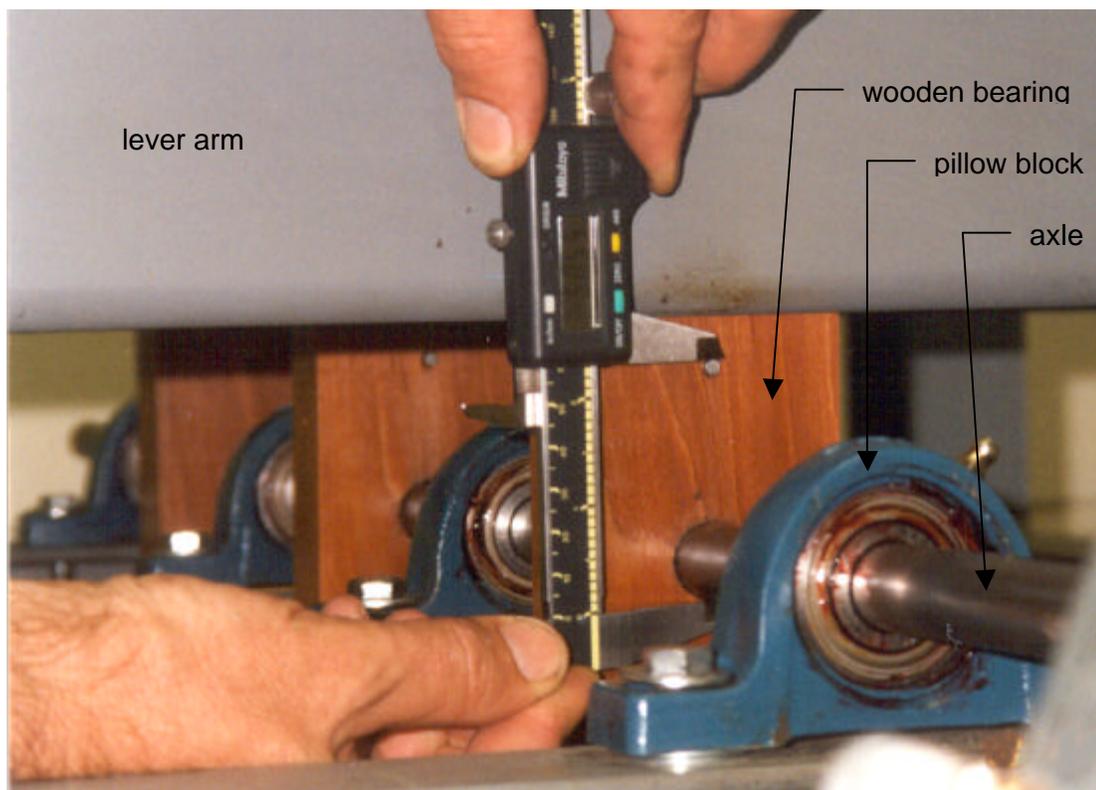
Figure 4. Diagram of friction test setup

### 2.5.2 Dynamic rotational friction

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. If the cord then proceeded to unwind from the pulley on its own, the moment produced by the weights would have exceeded the dynamic frictional forces between the bearing and the shaft. If not, more weight was added and the procedure repeated. The weight required to cause unwinding of the cord was recorded. This procedure was repeated two times at half-rotation ( $180^\circ$ ) intervals around the axle. The average weight required to produce unwinding of the cord from the pulley was then calculated, and the coefficient of dynamic friction was calculated as above. Bearings with lower dynamic friction were considered to have better performance than those with higher friction.

### 2.5.3 Bearing wear

Wear of the bearing material was measured with a digital caliper accurate to 0.01 mm (see Figure 5). A steel pin was inserted in each bearing during fabrication, and the distance between the top of the pin and the bottom of the shaft was measured to the nearest 0.01 mm. As the bearing wore, the distance measured became less and less. Measurements were made on both sides of the bearing and averaged, to account for asymmetrical wear. Bearings with lower rates of wear were considered to perform better than those with higher wear.



**Figure 5.** Measuring wear of wooden bearing

#### **2.5.4 Shaft wear**

Wear of the steel shaft was measured with a digital caliper accurate to 0.01 mm. The shaft diameter was measured to the nearest 0.01 mm at the beginning of each bearing test, and again at the end of each 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. A bearing material that causes high shaft wear is a failure.

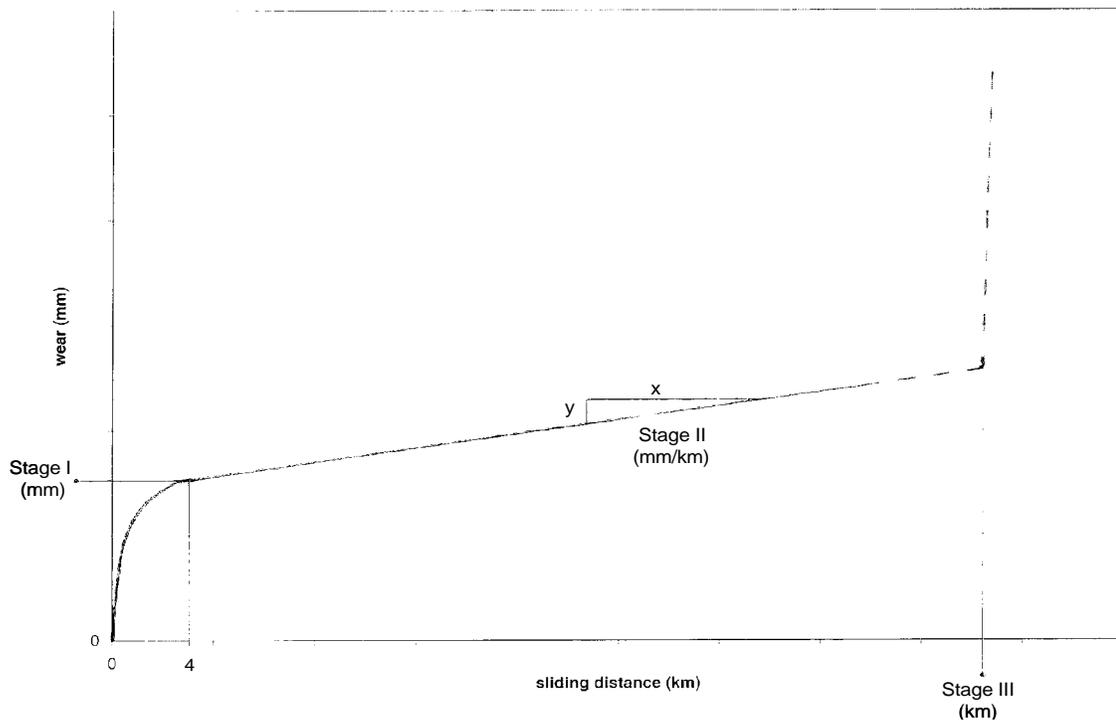
### 2.5.5 Bearing operating temperature

The operating temperature of the bearing was measured by inserting the probe of an electronic thermometer, accurate to 0.1°C, into a hole in the bearing material. Measurements were taken while the testing machine was in operation and the bearing was under load. The temperature reading was recorded to the nearest degree Celsius. The ambient air temperature was measured with the same thermometer and the difference between the bearing temperature and the ambient temperature was calculated. Lower operating temperatures were an indication of better bearing performance. Although high bearing temperature *per se* may not be disadvantageous, provided that the heat does not damage the wood or other components, the heat is a result of friction that in a well performing bearing will remain low.

## 3 Experimental Results

### 3.1 Wear model

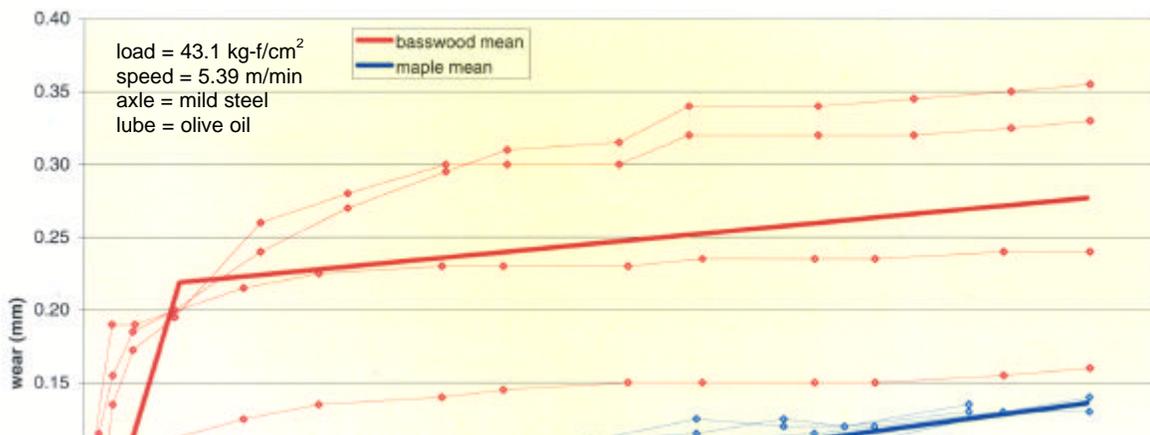
Wear of the wooden bearings was observed to follow three distinct stages. This pattern, shown in Figure 6, is typical of sliding wear (Czichos 1992). Wear rates were high during an initial run-in period (Stage I) of up to 4 km sliding distance (sliding distance = revolutions x shaft circumference). Wear rates then declined and stabilized (Stage II). Later, some bearings failed suddenly either by splitting or burning (Stage III). These three stages were quantified by fitting a regression line to the wear measurements between the initial run-in period and failure. Coefficients of determination ( $r^2$ ) for most bearings ranged from 0.88 to 0.99, indicating quite linear wear. A few bearings had more irregular wear with  $r^2$  values as low as 0.73. The slope of the regression line was the parameter of Stage II linear wear rate (mm/km). The y value of the regression line at x=4km was the parameter of Stage I initial run-in amount (mm). The x value when failure occurs was the parameter of Stage III sliding distance to failure (km).



**Figure 6.** Bearing wear model showing parameters of three stages of wear

### 3.2 Wood density

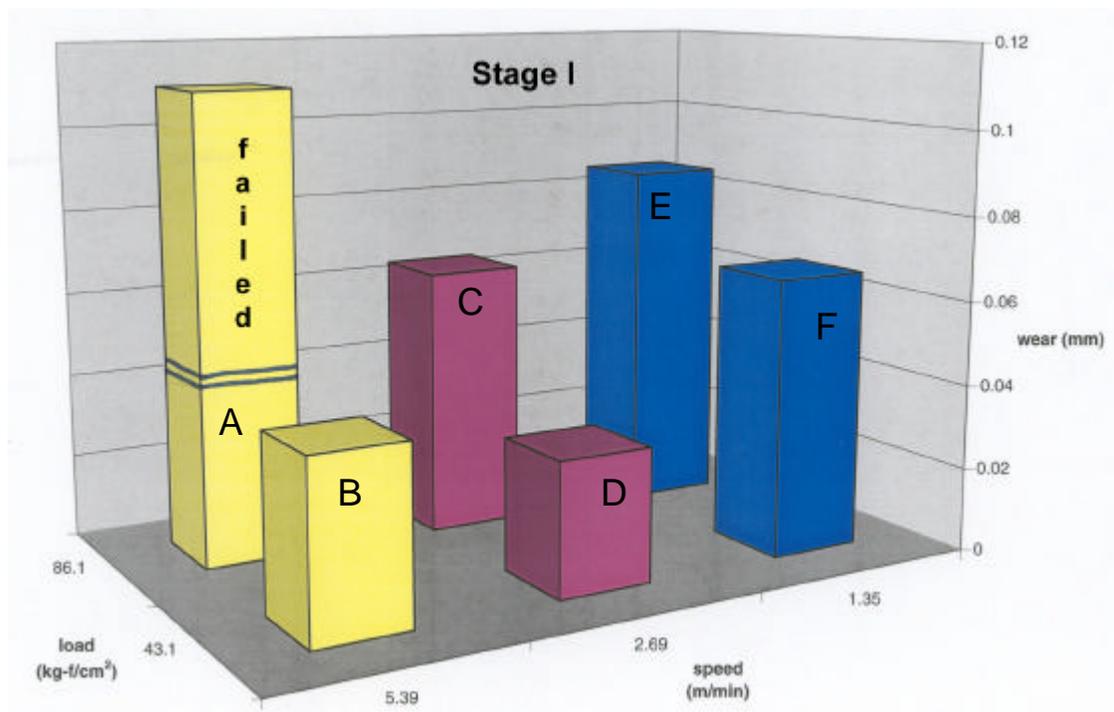
Bearings were fabricated of basswood (*Tilia sp.*) (bearings # 94,95,96,97) and maple (*Acer sp.*) (bearings # 51,54,55,63,64), two woods having similar physical characteristics including diffuse pore distribution and high permeability. The specific gravity of the basswood was 0.36 and that of the maple was 0.72. The bearings were treated with olive oil and tested to 42 km sliding distance at the same speed and load levels. As shown in Graph 1, the basswood bearings had greater wear during Stage I run-in, but upon reaching Stage II the basswood bearings had a slightly lower average linear wear rate than the maple bearings. Variation in wear between individual basswood bearings was greater than the variation between maple bearings. None of the bearings reached Stage III failure.



**Graph 1.** Bearing wear vs. sliding distance for woods of different density

### 3.3 Load and speed:

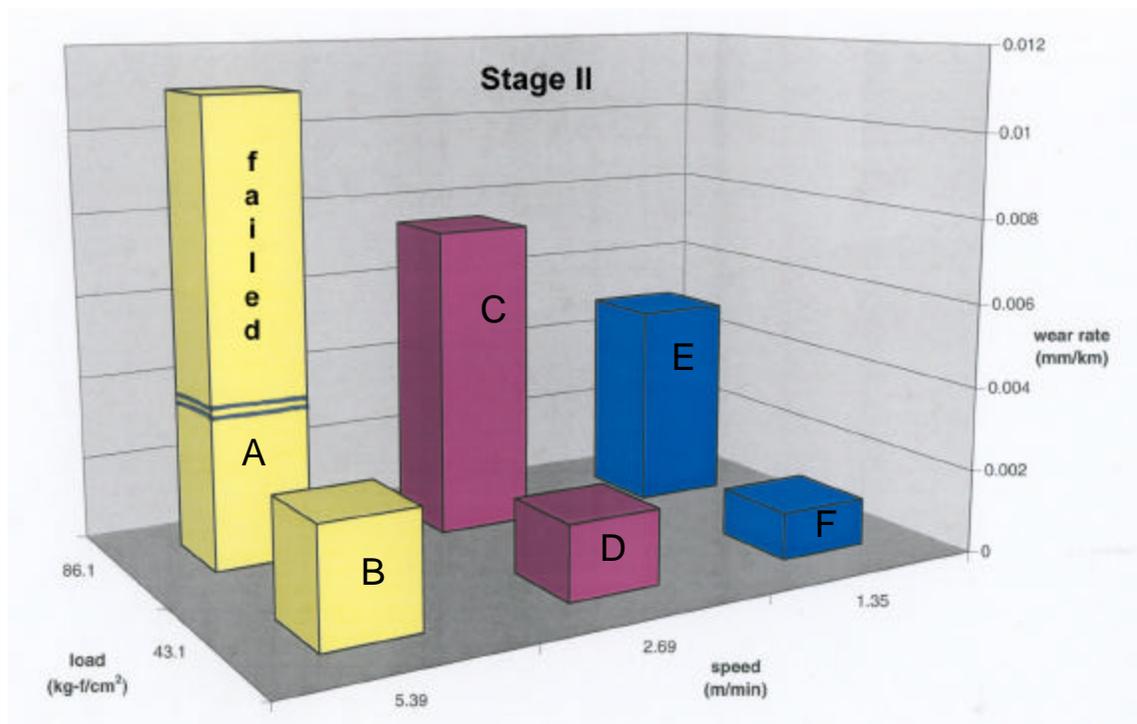
Twelve bearings were prepared of maple wood and treated with olive oil (bearings # 50,52,53,56,61,62,63,65,66,67,70,71). Two bearings were tested at each of six combinations of load and speed. Levels of sliding speed were low (1.35 m/min), medium (2.69 m/min) and high (5.39 m/min), and load stress levels were low (43.1 kg-f/cm<sup>2</sup>) and high (86.1 kg-f/cm<sup>2</sup>). Both bearings tested at high load and high speed failed. None of the other bearings failed during the tests, although their wear amounts and rates were not linear with changes in load and speed. Stage I run-in amount for the six groups of bearings are shown in Graph 2, while Stage II linear wear rates are shown in Graph 3. Bearing wear was more dependent on load stress than on sliding speed. This is explained in greater depth in section 4.2.



**Graph 2.** Stage I run-in amounts for bearings tested at different loads and speeds

### 3.4 Wood permeability

Bearings were prepared of maple (bearings # 51,54,55,63,64) and muninga (*Pterocarpus angolensis*) (bearings # 108,109) two woods of similar density (SG=0.62-0.72) and diffuse pore distribution. The bearings were treated with olive oil using the vacuum method described above. The maple bearings retained an average of 48.4% of their air-dry weight in lubricant while the muninga bearings retained only 4.2% lubricant. On the testing machine, the



maple bearings uniformly ran to 40+ km sliding distance with less than 0.15 mm wear. None of the maple bearings failed. At the same operating speed and load,

**Graph 3.** Stage II linear wear rates for bearings tested at different loads and speeds

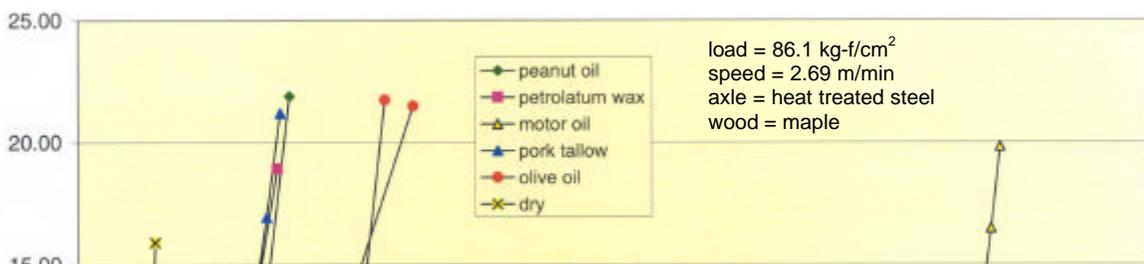
the muninga bearings experienced Stage I run-in amounts averaging 3.1 times greater than the maple bearings, and Stage II linear wear rates averaging 2.6 times greater than the maple bearings. One of the muninga bearings failed within 32 km sliding distance. The operating temperature of the muninga bearings averaged 12.5°C greater than the maple bearings.

### 3.5 Wood pore size and distribution

Bearings were prepared of maple (bearings # 51,54,55,63,64) and red oak (*Quercus sp.*) (bearings # 104,105) two woods of fairly similar density (SG=0.59-0.72) and high permeability. The maple wood was diffuse porous, with small conduction vessels uniformly distributed throughout its cross-section. The oak wood was ring porous, with larger vessels concentrated in growth rings about the tree's cross-section. The bearings were treated with olive oil and retained from 47.5 to 48.4% of their air-dry weight in lubricant. The bearings were tested at the same load and speed for 40+ km sliding distance. The oak bearings experienced Stage I run-in amounts averaging 2.7 times greater than the maple bearings. After running-in for 4 km sliding distance, the two wood types experienced similar Stage II linear wear rates. None of the bearings failed.

### 3.6 Axle properties

Tests were conducted with axle shafts made of cold-drawn 1018 mild steel and ground-and-polished 4140 heat-treated steel. Bearings made of maple wood were treated with a variety of lubricants and tested on the two types of steel shafts. Bearing life was remarkably shorter when run on the ground and polished heat-treated shafts, as shown in Graph 4. Of six lubricated maple bearings tested on heat treated shafts (bearings # 25,26,27,28,29,31), most failed within 0.15 km sliding distance and the longest lasting failed after 0.65 km. Conversely, of 20



**Graph 4.** Wear vs. sliding distance for bearings run on heat-treated steel axles

lubricated maple bearings tested on mild steel axles under identical load and speed conditions (bearings # 6,9,11,12,13,14,15,16,17,18,19,20,21,24,32,33, 34,35,66,70, bearings lubricated with dry graphite and liquid soap not included), many bearings continued operating past 30 km, and the shortest-lived bearing stopped after 4.2 km because of axle breakage. Coefficients of dynamic friction for the heat-treated shafts were 17% to 42% 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.

### **3.7 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. Dry bearings without lubrication were also tested. Two bearings treated with each of the following lubricants were tested, as well as three unlubricated bearings:

peanut oil (bearings # 82,83)

olive oil (bearings # 78,79)

30W motor oil (bearings # 90,91)

mineral oil USP (bearings # 88,89)

petrolatum wax (bearings # 22, 23)

beeswax (bearings # 74,75)

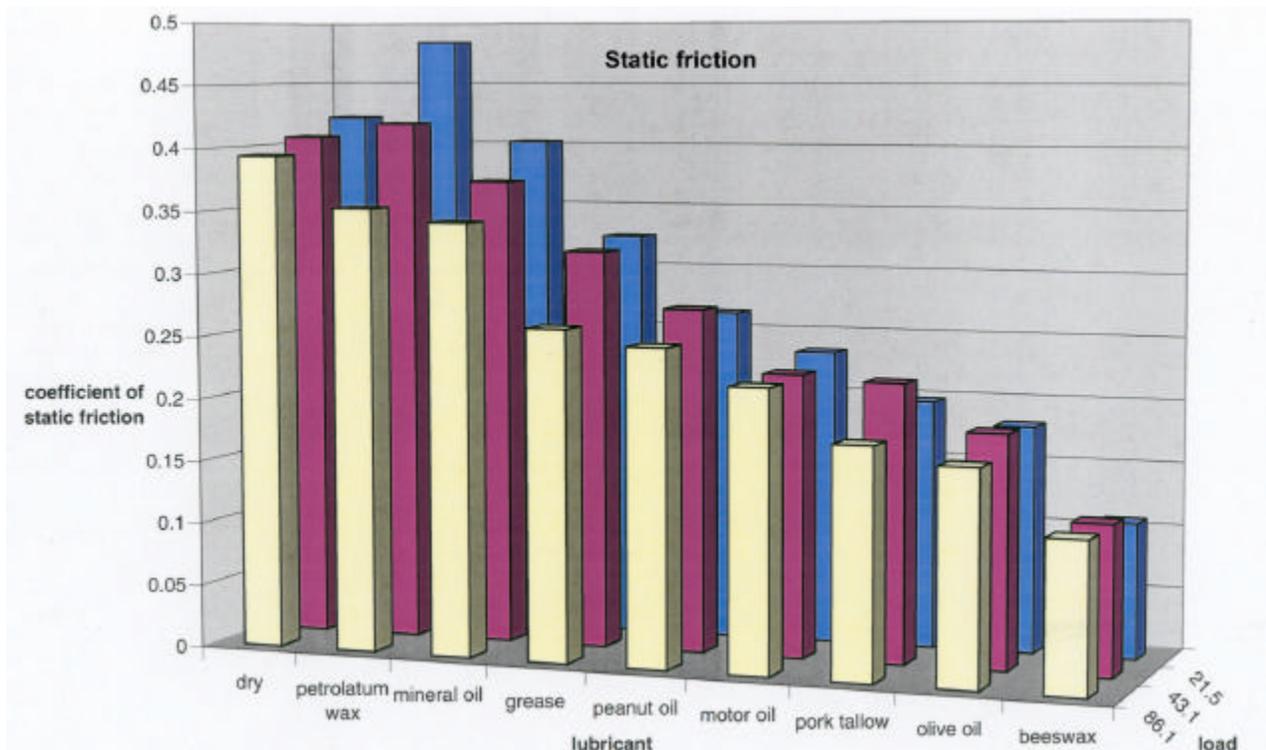
axle grease (bearings # 76,77)

pork tallow (bearings # 86,87)

dry (unlubricated) (bearings # 80,84,85)

As shown in Graph 5, 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 liquid lubricants as a group (peanut oil, olive oil, motor oil, mineral oil) and solid lubricants (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).

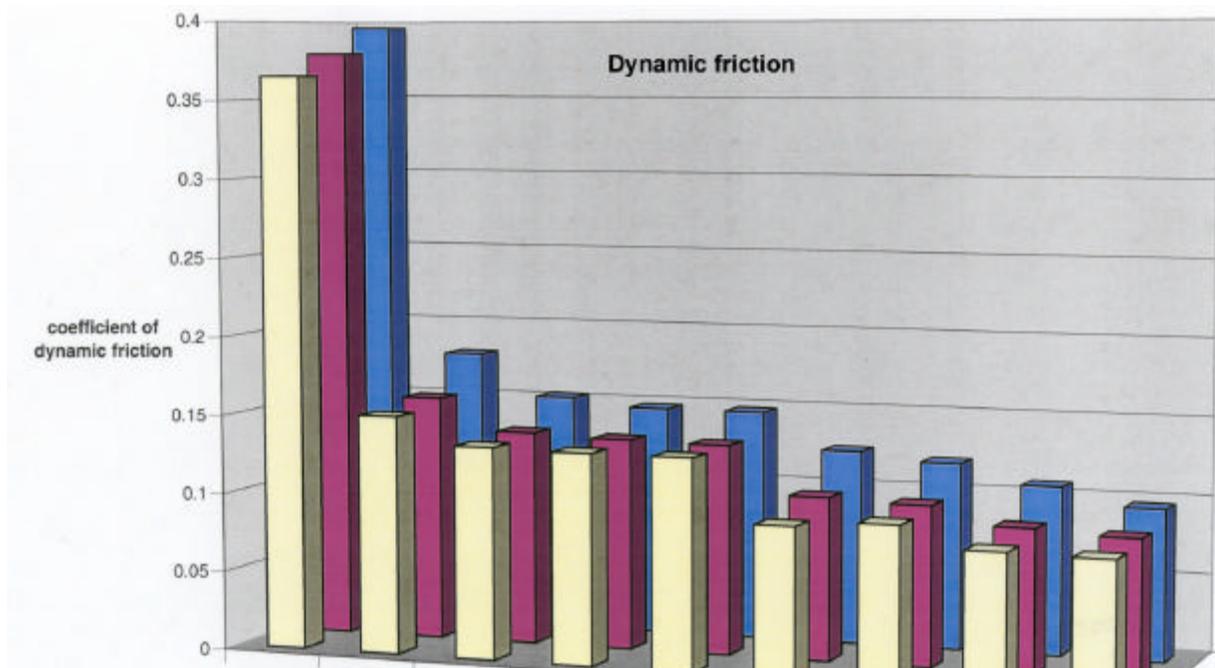


**Graph 5.** Static friction of eight lubricants at three load stress levels

### 3.8 Dynamic friction

Tests were conducted to determine the coefficient of dynamic friction of eight different lubricants at three load stress levels. The same bearings and lubricants as tested above for static friction were also tested for dynamic friction. As shown in Graph 6, there was a large difference in dynamic friction between dry and lubricated bearings. Petroleum-based lubricants, as a group, had higher friction than animal- and vegetable-based lubricants. There was no apparent difference between solid and liquid lubricants. All lubricants tested showed a slight downward trend in coefficient of dynamic friction as the load level increased. Bearings lubricated with beeswax had the lowest friction levels of all lubricants tested.

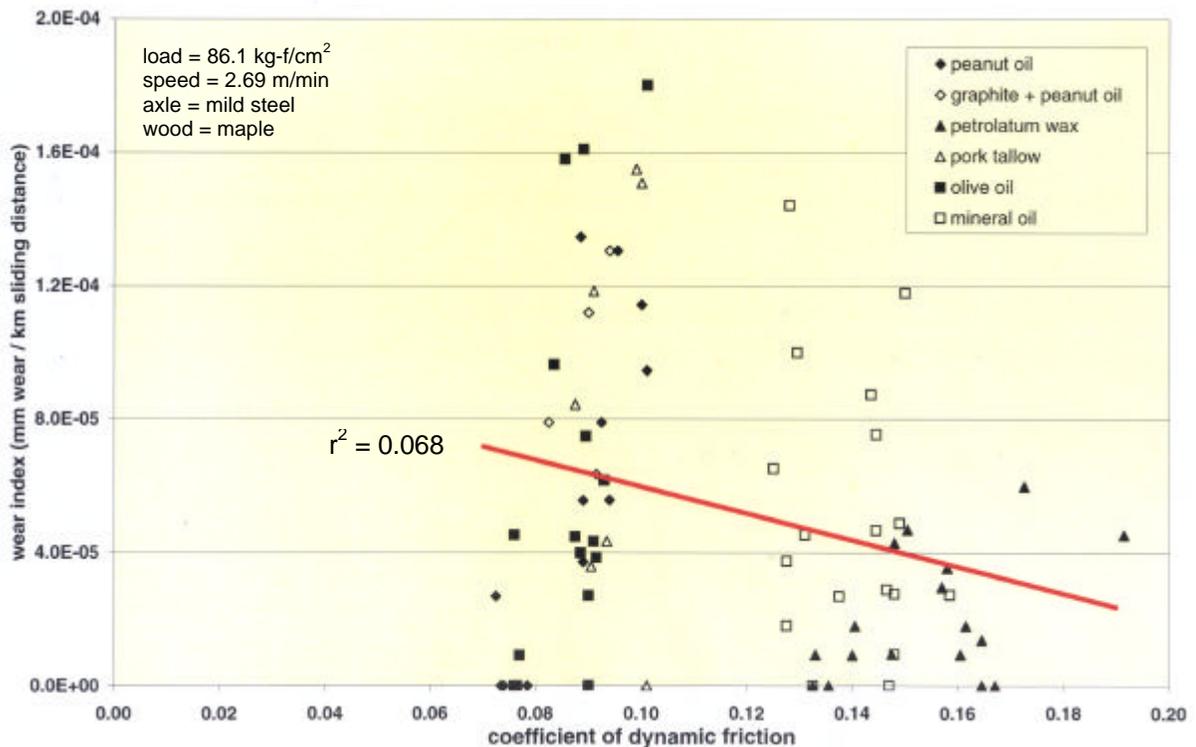
Limited tests were conducted using powdered graphite, and graphite mixed with peanut oil, as lubricant. The presence of graphite produced no apparent reduction in friction or wear. One maple bearing (#10) lubricated only with powdered graphite exhibited high friction and very short lifespan, similar to an unlubricated maple bearing (#7) tested under the same load and speed conditions. Two maple bearings (#19,33) lubricated with powdered graphite mixed with peanut oil showed fairly low friction and wear rates, similar to two maple bearings (#6,11) lubricated with peanut oil alone.



**Graph 6.** Dynamic friction of eight lubricants at three load stress levels

### 3.9 Wear vs. friction

To establish a relationship between bearing wear and friction, a series of maple bearings (# 6,11,12,13,14,15,17,19,20,21,24,32,33,34,35) treated with various lubricants were tested for up to 45 km sliding distance, with periodic measurements made of both wear and dynamic friction. For each consecutive pair of measurement points, the wear rate and mean friction were calculated and plotted in Graph 7. There was no apparent correlation between bearing friction and wear. Data points were somewhat clustered according to lubricant type. Bearings lubricated with petrolatum wax registered the highest friction levels but low wear rates. A distinction was evident between the petroleum-based lubricants



**Graph 7.** Bearing wear rates vs. dynamic friction for various lubricants

(mineral oil, petrolatum wax), which had coefficients of friction in the range of 0.13 to 0.17, and the animal- and vegetable-based lubricants (peanut oil, olive oil, pork tallow) with coefficients of friction of 0.08 to 0.10. Within each of these two groups, however, wear rates ranged from zero to high.

## **4 Discussion**

### **4.1 Lubricants**

The choice of lubricant can greatly affect bearing performance. All unlubricated bearings tested showed very high friction and wear rates. The presence of any lubricant (except dry graphite) reduced dynamic friction levels to less than half that of dry bearings. 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. This suggests that the bearings operate in the boundary lubrication regime, in which 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 levels in the boundary regime are 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. Non-polar petroleum-based lubricant molecules cannot attach strongly to the surfaces and do not form a low-friction layer.

Graphite should also be a good boundary lubricant because it has a crystal lattice structure composed of layers, the bonds between atoms in the same layer being strong while the bonds between adjacent layers are weak. As the graphite is compressed between two sliding surfaces, it shears relatively easily between layers resulting in low sliding friction (Ellis 1970). Experimental

results, however, show that graphite was not an effective lubricant for wooden bearings. A possible explanation is that the compressive strength of the wood is roughly the same as that of the graphite crystal, so that the graphite may become embedded in the wood rather than shear between the wood bearing and steel axle. The compressive strength of graphite is about 3,000 to 6,000 kPa (Ellis 1970), while the compressive strength parallel to grain of maple wood is about 4,000 kPa (Forest Products Laboratory 1999).

Another consideration when choosing a lubricant is the tendency of liquid lubricants to drip out of the bearing during storage and use. The dripping of lubricant from the bearing, and the dust that may be attracted to the liquid, creates a cleanliness problem that may be more disadvantageous in some applications than others. Beyond the cleanliness issue, however, oil that drips from the bearing is no longer available to lubricate the bearing, resulting in possible long-term longevity problems. For this reason, lubricants that are solid at room temperature may be preferred to liquid lubricants.

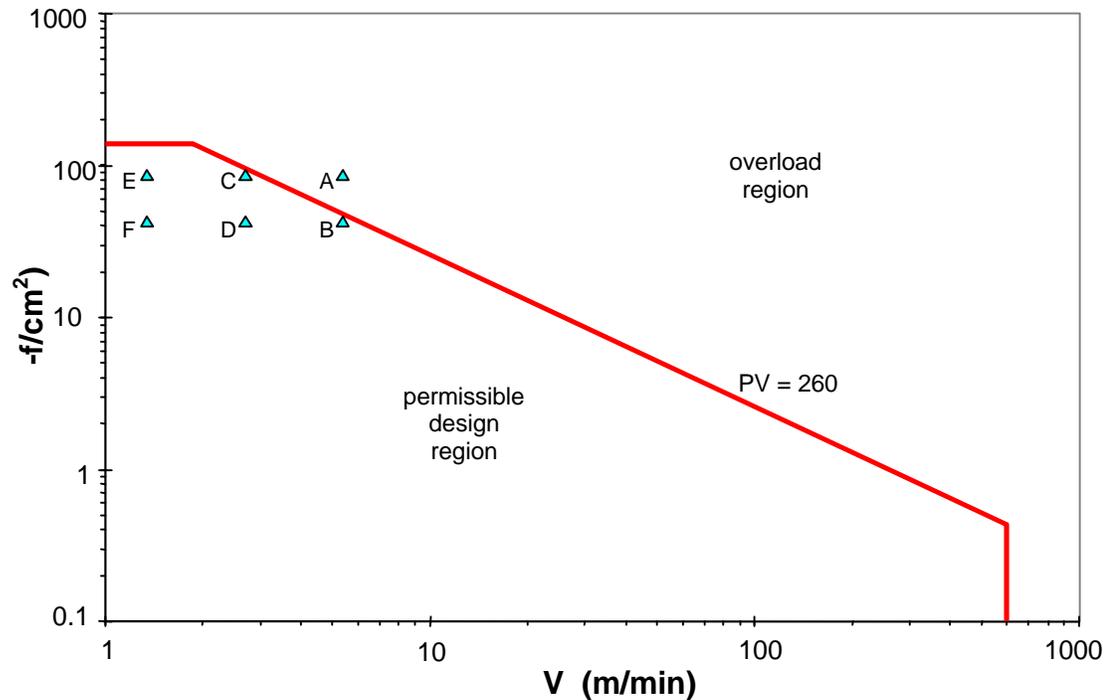
Based on all these considerations, a recommendation can be made to use animal- or vegetable-based lubricants that are solid at room temperature. Two such lubricants tested in this experiment were pork tallow and beeswax. Pork tallow showed low static and dynamic friction and low wear rates. Beeswax showed even lower friction levels but was not tested for wear. A potential disadvantage of these two lubricants, however, is the possibility that insects or rodents may be attracted to the treated bearings.

## **4.2 Load and speed**

A “P-V index” is commonly used to rate journal bearing materials for their suitability for use under different working 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. This implies that, as long as the product of load and speed does not exceed  $PV_{\max}$  and the individual limits of  $P_{\max}$  and  $V_{\max}$  are not exceeded, a bearing should perform equally well if the speed is doubled and the

load is halved. For wooden bearings, values of the  $PV_{\max}$  found in the literature range from 260 to 320 ( $\text{kg-f/cm}^2$ )( $\text{m/min}$ ), with  $P_{\max}$  of 140 ( $\text{kg-f/cm}^2$ ) and  $V_{\max}$  of 600 ( $\text{m/min}$ ) (Wilcock and Booser 1957; Product Engineering 1977; Steuernagle 2001). This range of values is plotted on logarithmic axes in Figure 7.

Lancaster (1978) explained that the maximum load that a journal bearing can support is generally limited by the compressive strength of the material, while



**Figure 7.** P-V diagram showing load and speed of bearings tested

the limiting speed is determined by the ability of the bearing assembly to dissipate heat generated by friction. The P-V relationship is linear only in the middle ranges of load and speed, and becomes non-linear as load or speed approaches maximum value asymptotically.

The tests conducted in this experiment focused on the high-load, low-speed end of the P-V continuum, the operating regime of animal-drawn carts. The operating conditions of the bearings tested are shown on Figure 7 as points A, B, C, D, E, and F. Note that point A is outside the permissible design region because the product of speed ( $86.1 \text{ kg-f/cm}^2$ ) and load ( $5.39 \text{ m/min}$ ) exceeds the

$PV_{\max}$  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.

Data from the other bearings tested 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 speed resulting in constant P-V values. Bearings at points B and C all had total P-V values of 232, although bearings 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 P-V values of 116, while bearings D operated at twice the speed and half the load of bearings at point E. 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 (shown in Graph 2) and 2.8 times greater Stage II linear wear rates (shown in Graph 3) than bearings tested at points C and E under lower load and higher speed.

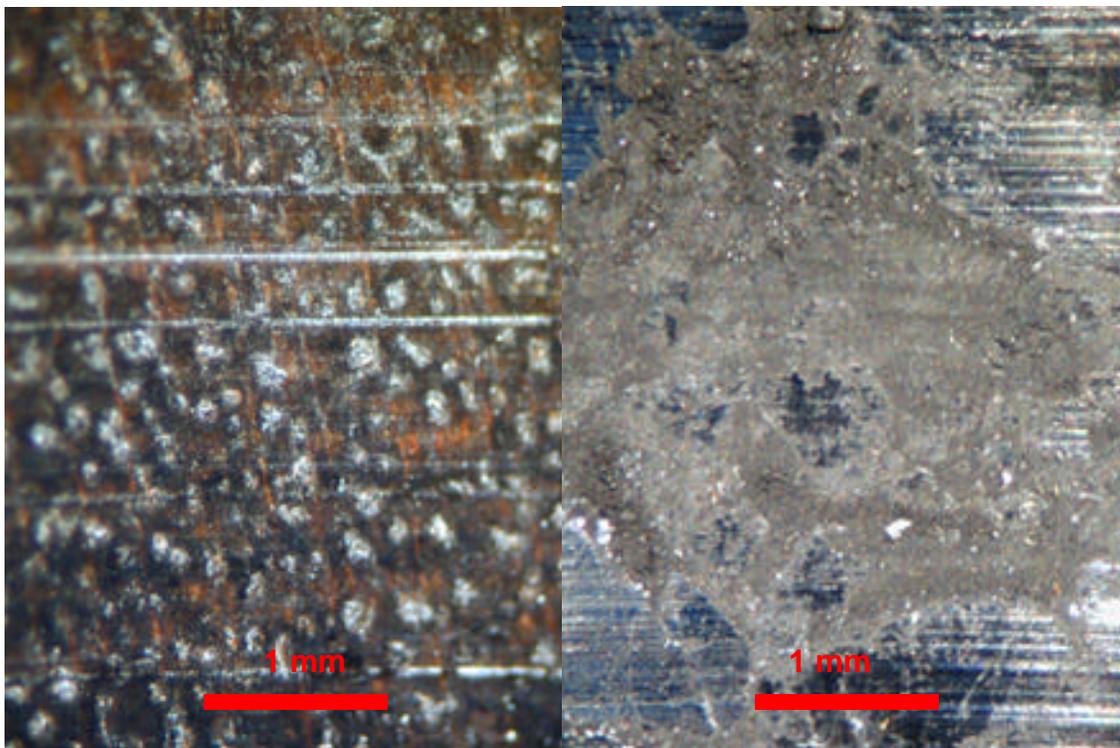
This study explored only a part of the possible range of loads and speeds under which wooden bearings can operate. Further studies are required to elaborate the true nature of the P-V curve at higher speeds and lower loads, as well as with different types of wood and lubricant.

#### **4.3 Wood properties**

The wood matrix has at least two functions in a wooden bearing. First, it must provide physical support for the load imposed on it. Second, it must act as a vehicle for the bearing lubricant. To satisfy 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. 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, and a wood of low density may contain a large quantity of lubricant but have inadequate physical strength.

This trade-off is understood by manufacturers of porous metal bearings. This type of bearing is made by fusing (“sintering”) metal particles together under heat and pressure. The density and porosity of the bearings can be varied by changing the particle size, heat, and pressure. It has been found that the best porosity for most porous metal bearing applications ranges from 25 to 35 per cent (Morgan and Cameron 1957). Applied to oven-dry wood, this range of porosity corresponds to wood with specific gravity ranging from 0.97 to 1.12.

However, a distinction should 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.



**Figure 8.** Magnified images (24x) of surfaces of maple bearing (left) and muninga bearing (right)

Limited wear tests were conducted on wood species of different density and permeability. Basswood, with high permeability and low density, had a greater Stage I run-in amount than did maple, a wood of high permeability and

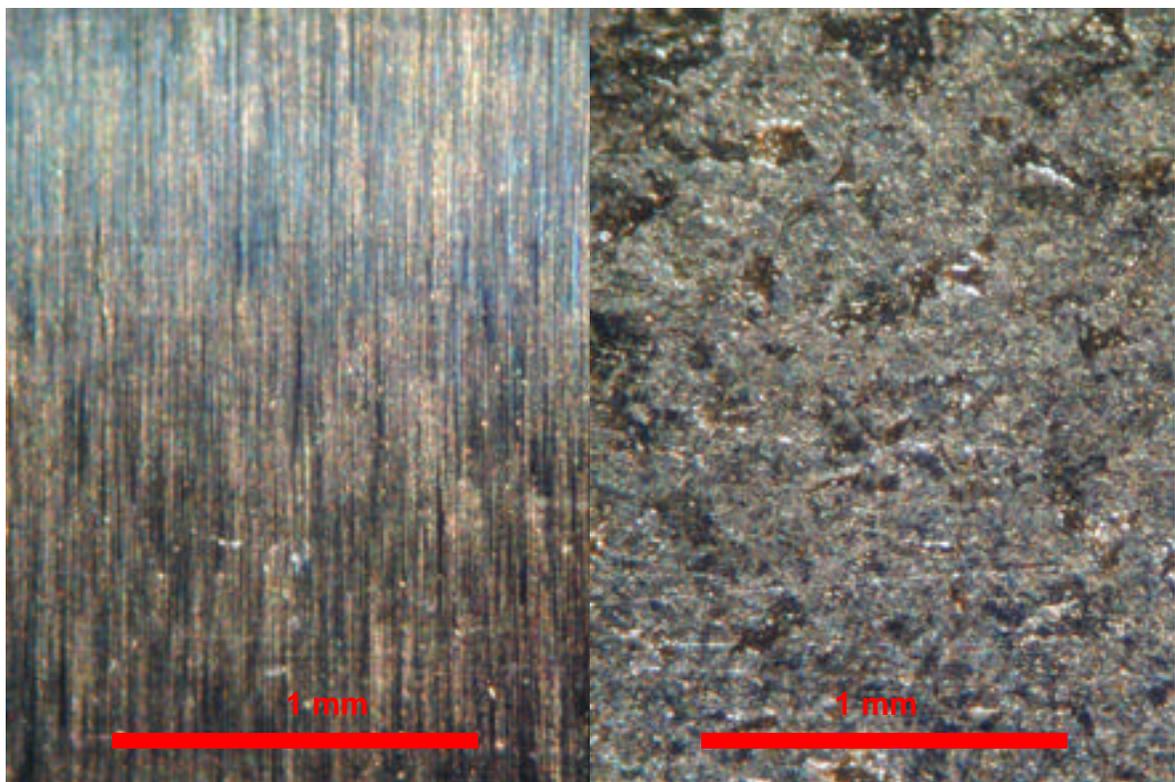
comparatively high density. However, the Stage II linear wear rates of the two woods were comparable. Muninga wood, fairly dense and yet with low permeability, had both high run-in amounts and high linear wear rates. Because initial run-in is a one-time event that occurs when a bearing is first put into service, while linear wear continues throughout the lifespan of the bearing, these results suggest that permeability is more important to bearing longevity than is density. Magnified images of the bearing surfaces (Figure 8) show that the maple bearing remained oil-soaked after 42 km sliding distance, while the surface of the muninga bearing was dry and charred.

The limited tests conducted in this study of woods of different pore distribution suggest that diffuse porous woods perform better than ring porous woods. Further studies of the effect of pore distribution and pore size on bearing performance should be undertaken. With porous metal bearings, it has been shown that the size distribution of pores is an important feature (Morgan 1957). The capillary forces holding the lubricant varies with pore size, allowing oil to first be removed from the larger size pores and later from progressively smaller pores. A bearing material with a range of pore sizes should thus have a longer lifespan than a material with uniform pore size. If the same relationship holds for wooden bearings, a semi-ring porous wood with a large range of pore sizes may make a good bearing material.

#### **4.4 Axle properties**

The difference in wear rates between bearings run on cold-drawn 1018 mild steel axles and bearings run on ground and polished 4140 heat-treated steel axle shafts is striking. Wood/lubricant combinations that performed very well on the mild steel shafts burned up within minutes or hours when run on the heat-treated shafts. It seems likely that the surface properties of the steel shafts account for the difference in wear, rather than the metallurgical properties of the different metals. Magnified images of the surfaces of the axles (Figure 9) show parallel ridges on the heat-treated steel caused by the grinding and polishing process. The ridges on the heat-treated shaft ran in the direction of rotation, so

that as the shaft rotated the ridges tended to dig into the wood surface and prevented any accumulation of lubricant or wood debris. In contrast, the surface of the cold-drawn mild steel axle had a more mottled appearance. It may be that the irregular topography of the cold-drawn steel surface allowed lubricant or debris to build-up between the sliding surfaces, thereby reducing wear. Further tests should be conducted with axle shafts of varying roughness and surface feature orientation, to see the effect on bearing wear of different axle surface characteristics.



**Figure 9.** Magnified images (48x) of surfaces of heat treated steel axle (left) and mild steel axle (right)

## 5 Conclusion

Wooden bearings have a long history of use, and they continue to be used in a number of applications. Many factors affect bearing behavior, none of which are well understood. Successful applications of wooden bearings have relied on trial-and-error experience to establish reliable performance. The present pilot

study has examined several wood properties, fabrication methods, and operating conditions in an attempt to provide an understanding of some of the more important factors that influence bearing performance. The results indicate that under certain conditions hardwood 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,
- diffuse porous wood gave lower wear than ring porous wood,
- 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,
- some lubricants gave static friction levels comparable to unlubricated bearings,
- petroleum-based lubricants gave higher friction than animal- and vegetable-based lubricants,
- lubricant viscosity had no apparent effect on friction,
- wood 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, and
- no correlation was apparent between friction and wear.

This study begins to illuminate some of the factors that control bearing behavior, which is a critical step to using wooden bearings in a rational manner. Additional work is needed, however, not only to confirm and expand on these findings but also to address the many additional factors not covered by this study.

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