EFFECT OF BLADE DESIGN ON SHEARING FORCES OF HYDRAULIC SHEAR FOR ROD WOOD

Thesis for the Degree of M. S.

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ABSTRACT

EFFECT OF BLADE DESIGN ON SHEARING FORCES OF HYDRAULIC SHEAR FOR ROD WOOD

By

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Felling of rod trees with machines using hydraulic powered blade has become a standard practice in forest mechnization. These machines have proven to be fast, flexible and safe. However further work must be done to improve their performance. The magnitude of the shearing force is dependent on the blade characteristic and the geometry of the link through which the force is transmitted.

A model two blade felling machine was designed and constructed. The blades were hydraulically operated and moved toward each other in the same plane.

The objectives of this study were to investigate:

a. the blade thickness, edge angle and shape that will be the best for this model.

b. the force required and blade design for a prototype, which is expected to work with 18 in. tree diameter.

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Three blade thicknesses, each one with three different edge angles were made of steel plate. They were tested with fresh wood samples 3.125, 4.125, and 5.5 in. in diameter.

The 0.50 in. blade had the poorest performance. However, after changing its shape by grooving both sides, a reduction of 30% of the force required was obtained.

The 0.187 in. blade had the best efficiency and it was selected for model test.

The model tests were done with 0.187 in. blade having 30 degrees edge angle by using wood samples of 4.5 in. diameter. Twenty-two tests were made which gave the data for computing the Pi terms. From the Pi terms the force required for the prototype was calculated to be 42,000 lb. with parameters:

t = 0.75 in. grooved blade

d = 18 in tree diameter

α = 30 degree blade edge angle

However, during the model tests, the design condition equations were not satisfied completely due to the capacity of the model hydraulic system and mechanical properties of the samples.

Approved [signature]
Major Professor

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Department Chairman

EFFECT OF BLADE DESIGN ON SHEARING FORCES

OF HYDRAULIC SHEAR FOR ROD WOOD

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Manuel L. Lecca R.

A THESIS

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CHAPTER I

INTRODUCTION AND OBJECTIVES

In the last ten years, tremendous effort has been devoted to forest mechanization, especially in harvesting. The use of the new hydraulic shear blade, when properly designed, has proven to be a faster, flexible and safer way of felling rod trees.

However there are still some factors which reduce the efficiency of this device. The force required to operate the blades is large and is a function of the shape, size and cutting direction. Other factors such as the wood itself and environments also affect the magnitude of applied force.

There are very few experimental results which relate the force required with different parameters. Generally one blade was used against an anvil, with no axial pressure on the sample. These laboratory processes are not similar to those actually occurring in the field, but still these laboratory results help to evaluate different parameters for the purpose of modeling or designing of prototypes.

A careful study of these works indicates that the applied force and damage to the fiber can be reduced with proper blade shape and size and by applying the pushing force in the right direction.

The objectives of this study are to:

a. Study selected variables affecting the shearing force for design purposes.

b. Design a prototype shearing mechanism with two shearing blades traveling toward each other without overlapping.

c. Design, construct and test a model built to one-half scale.

d. Test three blade thicknesses (.50, .25, and .187 in.) each one with three different edge angle (30, 45 and 60 degrees).

e. Determine the force required and blade design for a prototype through data obtained from the model tests.

CHAPTER II

REVIEW OF THE LITERATURE

2.1 Common Shearing Blade Machines for Felling Rod Trees

Of all machines used in forest mechanization the tree felling device has become most important. The device shears trees with diameters up to 40 inches. The blades, which differ in design are activated by a most sophisticated hydraulic system; some of them are combine operating machines with auxiliary systems which fell and bunch simultaneously in one operation by using only blades.

The monoblade type is shown in Figure 1; the blade rotates on a pin. In this way the blade severes the fibers by compressing the tree stem against an anvil which does not move.

The guillotine type is shown in Figure 2. This device usually works in horizontal position and its blade is longitudinally stroked by one hydraulic cylinder. In front of the blade there is the "latch" mechanism which embraces the stem and holds it while the blade is forced through the wood. The principal feature of this type is that it allows the severing of the trees without the transfer of force to the tractor.

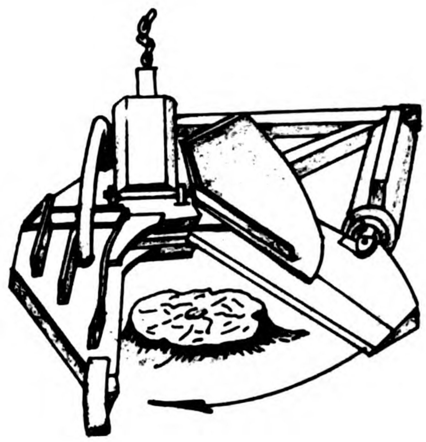
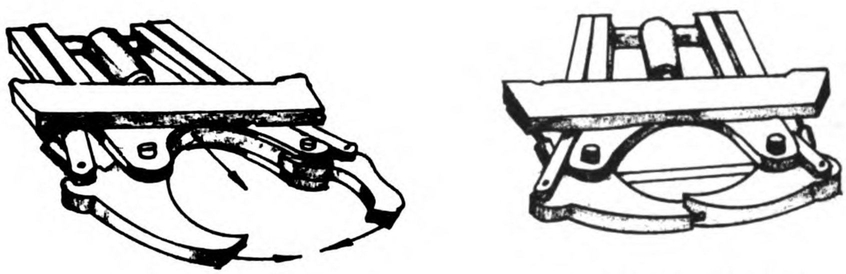


Figure 1.--Monohydraulic blade type.



Latch open

Latch closed

Figure 2.--Guillotine blade type.

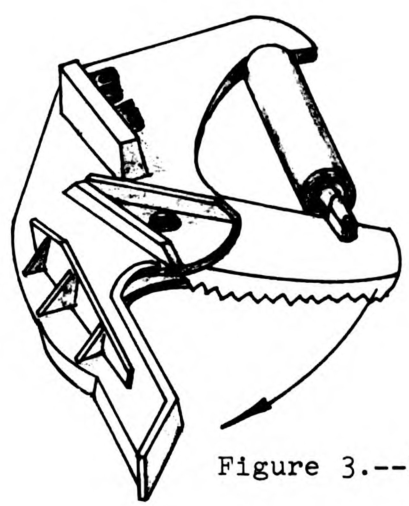


Figure 3.--Sawtooth blade type.

The sawtooth type is shown in Figure 3. This device is similar to the monoblade type except that the cutting edge is provided of exaggerative teeth.

2.2 Research with Hydraulic Blades in Laboratory

So far the research had been concerned with the study of the influence of factors that affect the force required in severing "square wood samples" perpendicular to the fiber and also to study damage resulting from the cutting process. It has been shown under laboratory conditions that the force required as well as the damage can be reduced to a minimum by a correct selection of the blade.

The factors which affect the force required can be grouped into three types: those inherent to the blade itself, those inherent to the material to be cut, and those due to the machine operation. The blade factors are the most important for design purposes.

The blade factors are: section shape, thickness, edge angle, wedge and quality of blade surface.

The principal factors inherent to the material to be cut are: moisture content, wood temperature, density, sample size and part of the trunk where the sample was obtained.

The factors important to the machine operation are: direction of the force applied and blade speed.

Review of one of the laboratory methods now available in literature will give a better understanding of principal features and comparative results of equipment, methods etc.

2.3 Equipment Commonly Used in Laboratory

Carl Kample (1965), Swedish mechanical engineer designed and used successfully a testing machine consisting of:

a. Power generator

b. Hydraulic system

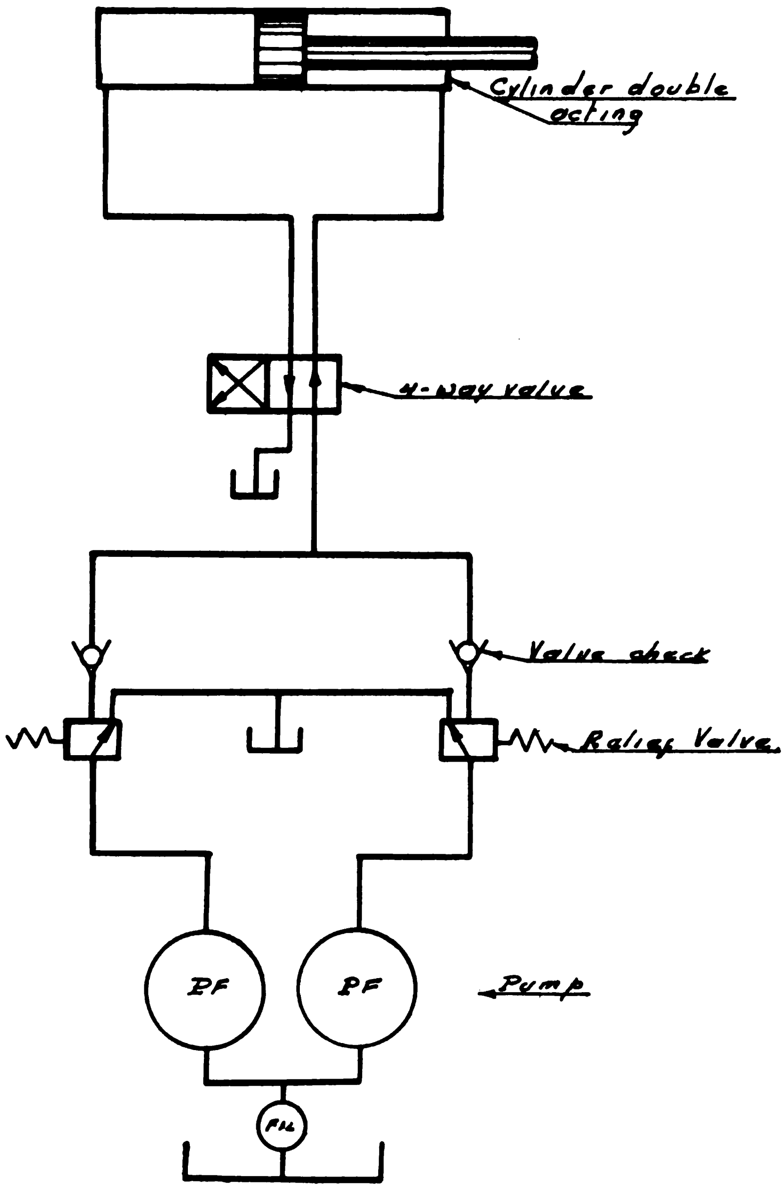
c. Shearing blades

d. Force metering

A Volkswagon industrial engine of 30 hp. and speed of 1500-300 RPM was selected as a power source. The output engine shaft was coupled to a single dry disc clutch to allow the operator to start or stop the mechanism. Between the clutch and the hydraulic pump there was a gear case with a 1.5:1 ratio and an elastic coupling to reduce wear to the pump due to mis-alignment of the shaft. A sketch of the coupling mechanism is given in Figure 5.

The hydraulic system consisted of a pump, two check valves, two pressure relieve valves, a reservoir, one hydraulic cylinder and a four way control valve. The pump was a Vicer's double vane type with a capacity

Figure 4.--Diagram of hydraulic system used in laboratory testing machine.



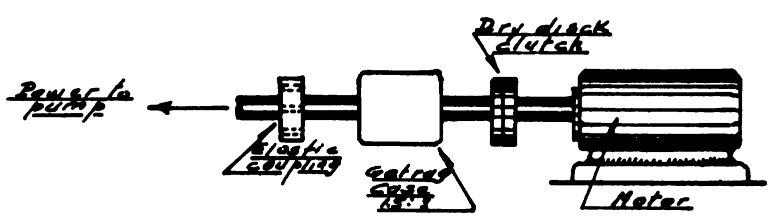


Figure 5.--Motor, gear case and couplings.

of approximately 75 lts./min. at 1990 psi. The four way control valve when in its rest position guided the flow back to the tank, and in its extreme position the oil was forced into the working cylinder. The hydraulic cylinder had a 3.75 in. internal diameter, 11.8 in. stroke and 2000 psi. maximum working pressure. A sketch of the hydraulic system is given in Figure 4.

The blades were made of heat-tempered steel. The testing was done with six blades, each one having distinct shape and size. The principal feature of them being:

Blade No.	1	2	3	4	5	6
Edge angle, degree.	45	45	30	45	60	45
Thickness, in.	.397	.198	.475	.345	.258	.358/.159

The blades were attached to the machine by two bolts, one of them to the blade-holder and the other to the hydraulic cylinder. The form of the blade-holder was such that it permitted the blade to work only on bending moment.

These blades are shown in Figures 6 and 7. It should be noted that blades numbered 3 and 4 are a little eccentric.

Besides, a special blade was used to determine the coefficient of friction between wood and metal. It was 18 in. long, 6 in. wide and 0.75 in. in thickness. The edge angle was 45 degrees.

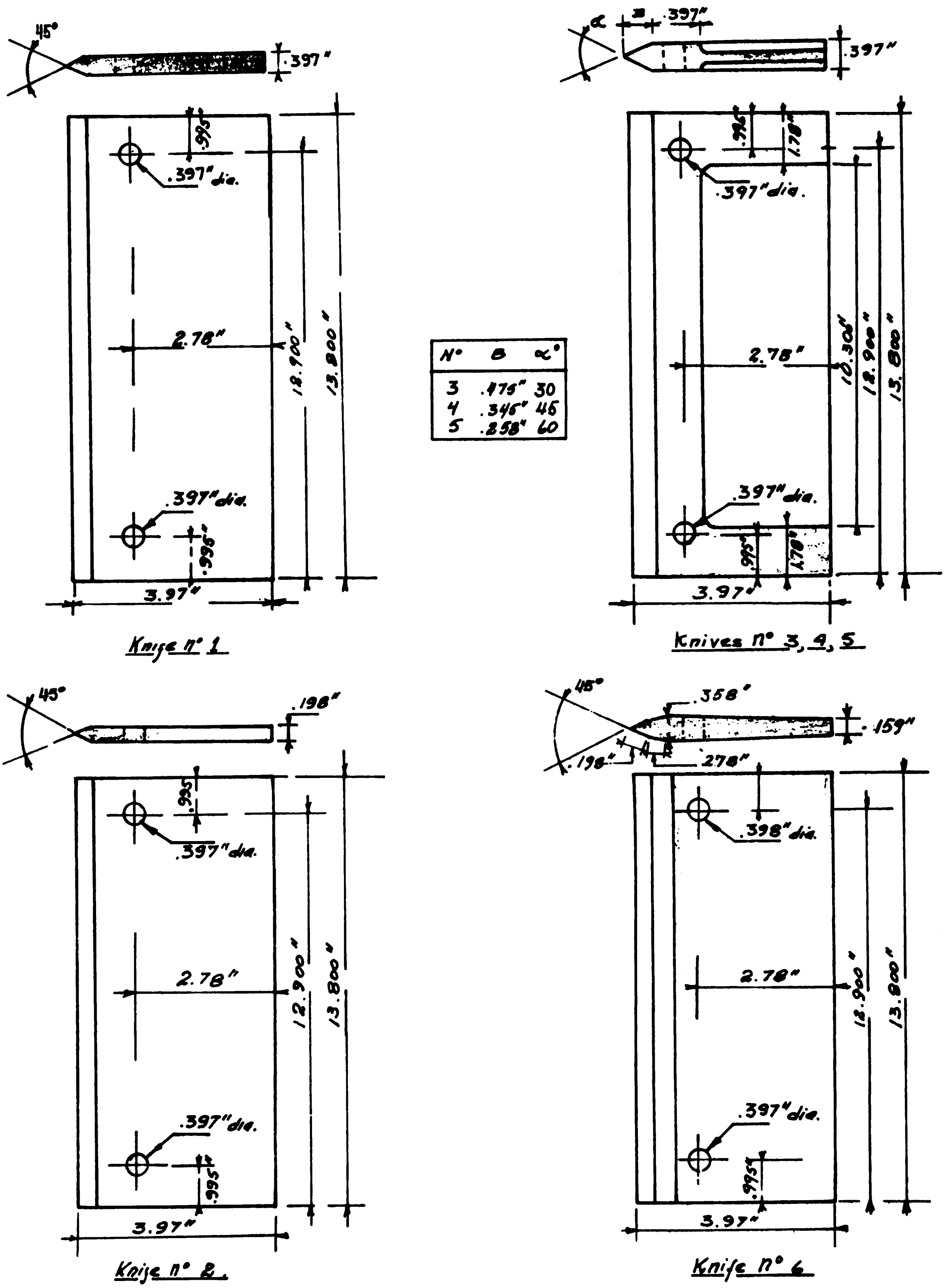


Figure 6.--Blades used in Laboratory.

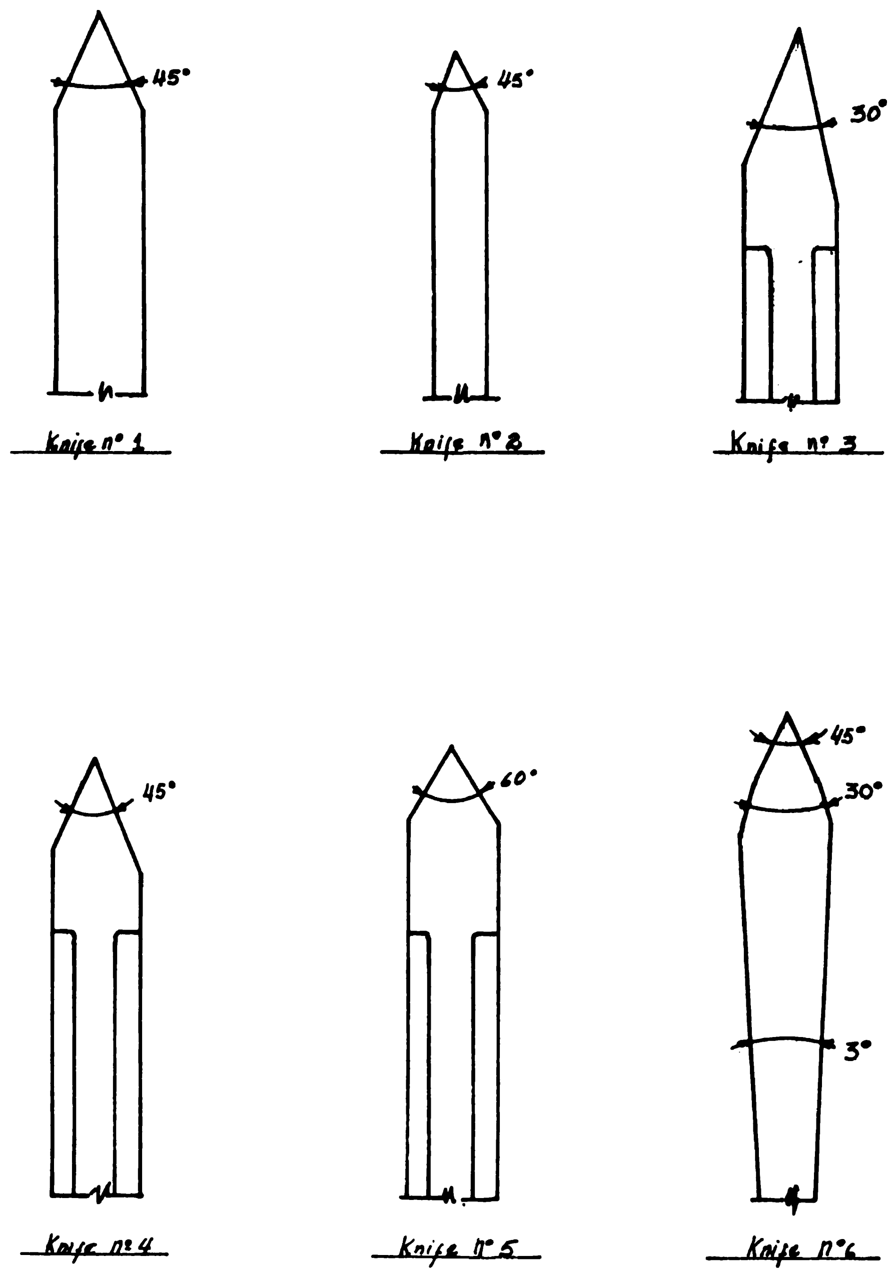


Figure 7.--Blade shapes used in laboratory.

The force was measured axially by two strain gauges; which were glued to the connecting rod between the piston and blade frame.

The output of the strain gauges was amplified and recorded by a Visicorder. The output from this recorder was in the form of a curve giving force per unit time. Since the knife and the recorder speed is an invariant, the curve also gives force per unit distance. A sketch of the testing machine is given in Figure 8.

In order to find the coefficient of friction, the same testing machine was used with a special blade. The blade used for friction was mounted directly on the hydraulic cylinder. The anvil consisted of two steel bars with a narrow spacing between them to allow the knife to pass through. In this way the friction force could be measured when the blade edge passed the anvil. The axial force was provided by a 25 Ton hydraulic jack, and any variation of this force was measured with a manometer. Knowing the axial force, and the pushing force exerted by the hydraulic cylinder, the coefficient of friction is obtained through the following relationship:

$$P = 2P_f$$

$$P_f = uPa$$

$$P = 2uPa$$

and $$u = \frac{P}{2Pa} \tag{1}$$

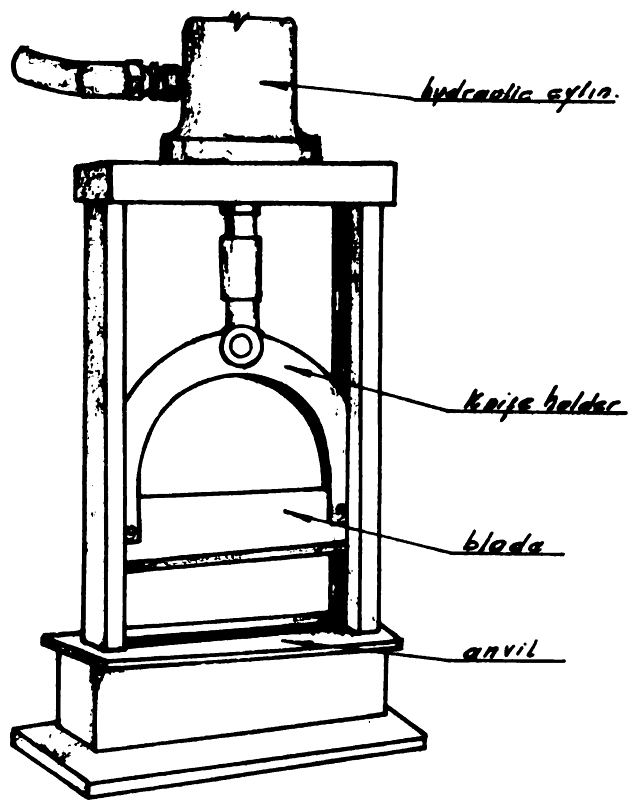


Figure 8.--Hydraulic machine used in laboratory for cutting wood.

Where:

P = knife pushing force

Pa = axial force on the jack

Pf = friction force

u = coefficient of friction

A sketch of the friction testing machine is given in Figure 9.

2.4 Sample Used for Laboratory Testing

The samples used were of rectangular section, because tests with a 10 in. outside diameter circular rod produced considerable damage due to compression of the rod smaple against the anvil.

The common rectangular sample section were as follows:

a. 2.75 x 2.75 in.

b. 2.75 x 3.95 in.

c. 3.95 x 3.95 in.

d. 3.95 x 5.55 in.

e. 3.94 x 3.17 in.

f. 5.55 x 5.55 in.

These samples were taken from tree sections in such a way that some of them were sapwood and others heartwood. The samples b, c, and d were also used in friction tests. They were 27.5 in. lenth to avoid the risk of buckling and an uneven distribution of strain due to axial force.

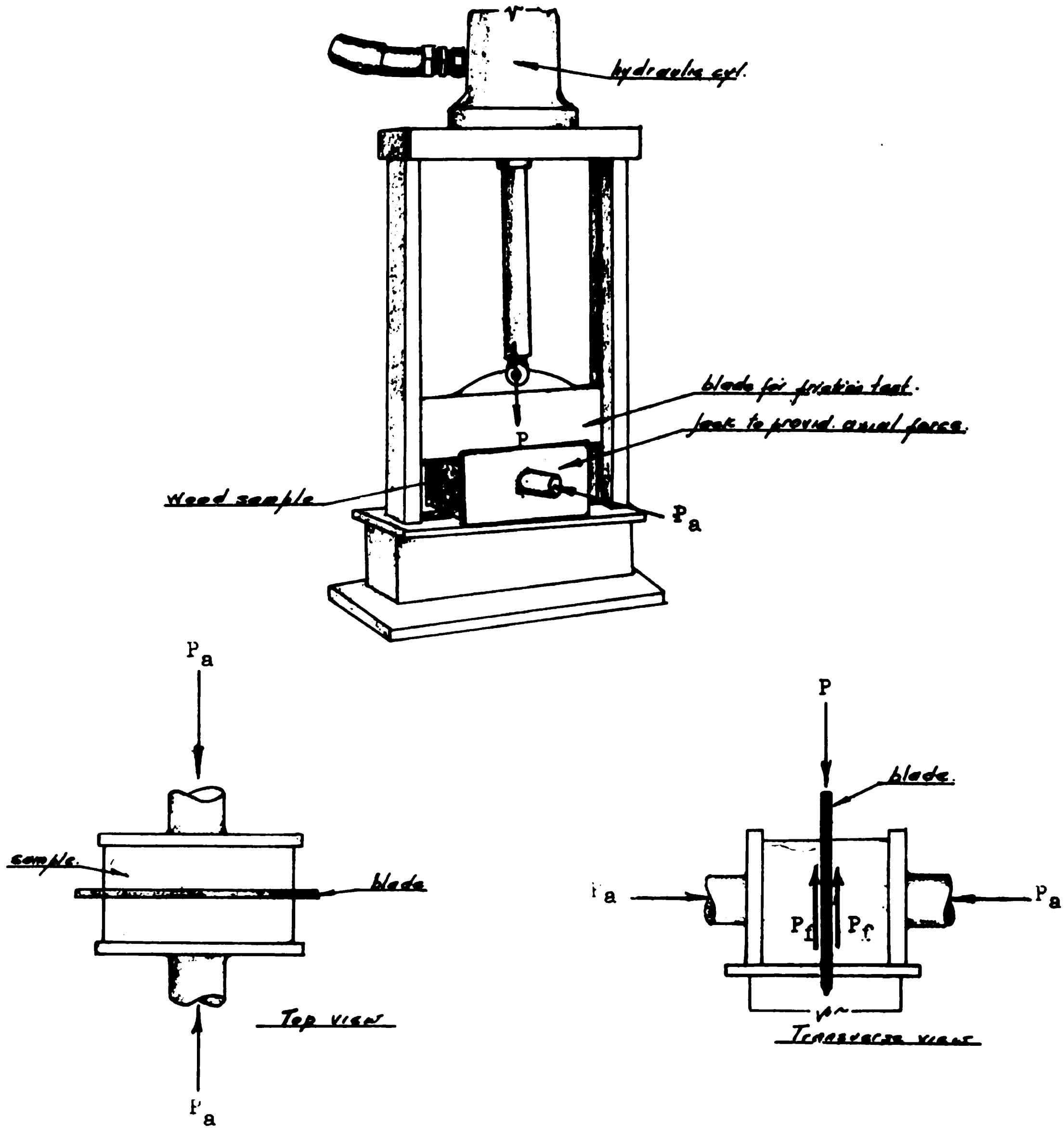


Figure 9.--Blade and frame used in determination of coefficient of friction.

The damage to the wood fiber produced by compressing the rod sample against the anvil suggests that shearing under field practice would be very well performed with a machine which has two blades acting parallel and against each other.

2.5 Some Results Obtained in Laboratory Tests

The results for rectangular sections previously mentioned are given below:

2.5.1 Influence of the Blade on the Force Required

2.5.1a Effect of the blade shape.--Blades 1, 4 and 6 represent three different shapes with the same edge angle of 45°. If one elects 100 units of force for blade 1, for comparison the result can be written as follows:

Blade 1	100 units
Blade 4	75 units
Blade 6	75 units

As a conclusion one could say that the cut-away shape (blade 4) and the sloping backward shape (blade 6) tend to reduce the force. In practice, however, these shapes are not common, perhaps due to manufacturing difficulties.

2.5.1b Effect of blade thickness.--Blades 1 and 2 represent two different thicknesses with the same edge angle of 45°. With blade 1 as the basis (100 units of force), the results to describe the effect of blade thickness can be reported as:

Blade 1	100 units
Blade 2	50 units

It is evident that the thinner blade required less force, however in practice the thickness is limited by bending stress and deflection.

2.5.1c Effect of blade edge angle.--Blades 3, 4 and 5 represent three different blade edge angles, but all of them have the same shape. Taking the force required for blade 4 as 100 units we can write the result of effect on edge angle as:

Blade 3	104 units
Blade 4	100 units
Blade 5	140 units

The results do not show any significant difference between blade 3 and 4, but the blade 5 shows much higher force. This suggests that the edge angle should remain in the range 30°-45°.

2.5.2 Influence of Wood Specimen on Force Required

2.5.2a Effect of specimen knots.--The cut over a knot showed about 50% greater force than a cut over a sample without knots.

2.5.2b Effect of specific gravity.--There was a significant difference between samples of different specific gravity. Samples with lower values required lower force. It was estimated that for each per cent increase of specific gravity, the force required increases by about 1-2 per cent.

2.5.2c Effect of moisture content and temperature.--Moisture content and temperature are factors related to each other. To show this, samples of sapwood and heartwood were exposed to three different temperatures of 15, -5 and -15 degrees Centigrade. The results showed that sapwood required more force under frozen than in warm storage. In any case sapwood required more force than heartwood which is contrary to what is expected, however this can be explained by the fact that sapwood has higher visco-elasticity.

2.5.3 Influence of Machine Operation on Force

Only blade speed and pushing force direction are considered here. As far as blade speed is concerned, laboratory results indicated that speeds between the range of 1.58-8.7 in. per second had no significant effect on force.

The importance of direction of pushing force need not be over emphasized. It is obvious that the force shall be mximum if the direction of the pushing force radial to the stem, i.e. two blades travelling parallel and against each other radially to the stem may be more efficient.

2.5.4 Coefficient of Friction

From the results of many experiments under different moisture content, speed and temperature conditions, the following conclusions can be drawn:

a. The coefficient of friction is lower in radial shearing than in tangential. Felling trees involves only radial shearing.

b. Friction seems to decrease with increases in speed expecially for the frozen sapwood.

c. Friction seems to increase with speed for heartwood.

d. Knots increase friction considerably.

2.5.5 Damage to Wood Fiber

To determine a qualitative damage to the fiber, thin samples were taken close to the cut surface. The samples were studied with a microscope under polarized light. Cracks and crushing were observed. Less damage resulted when using the thinnest blade at 30-45 degree of edge angle.

CHAPTER III

THEORETICAL EQUATION FOR SHEARING WOOD

Theoretically, the total force required in shearing can be divided into three components.

Edge force or cutting force

Wedging force or separating force

Friction force

The edge force or cutting force is defined as the force required to cut the fibers by a knife edge not ideally sharp.

The wedging force is defined as the force required for a knife wedge triangle to penetrate into the wood.

The friction force is defined as the force caused by friction between wood and the flat blade surface, except the wedge portion of the knife.

The theory involved in these forces is discussed below.

3.1 Edge Force

This force originates when the blade edge, not ideally sharp, penetrates perpendicular to the wood grain. As a result of this the fibers deflect and absorb energy before they are cut off. Figure 10 shows the fiber deflection during cutting.

Mckenzie (1961) formulated a theory concerned with edge penetration perpendicular to the grain which allows the theoretical calculation of the edge force. This theory is based on the fact that the edge force is due only to the bending moment under which the fiber deflects in front of the blade edge and breaks at certain deflection. However, it was observed that besides that, there is another component: the pulling force along the axis of the fiber, which is the result of the shear stress between the fiber sides. The shear stress between fibers causes separation from each other when the stresses are considerable. Mckenzie's theory neglects this force, because he was concerned with cutting small chips where axial forces would not really occur.

The cutting force is obtained by assuming a beam on an elastic foundation with a concentrated load at one end without axial force. Sketches of the assumed beams are shown in Figures 11 and 12.

In figure 11:

P_1 is the cutting force acting on a layer or ring of thickness h_1; this layer is considered as a beam and the next layers below as the elastic foundation.

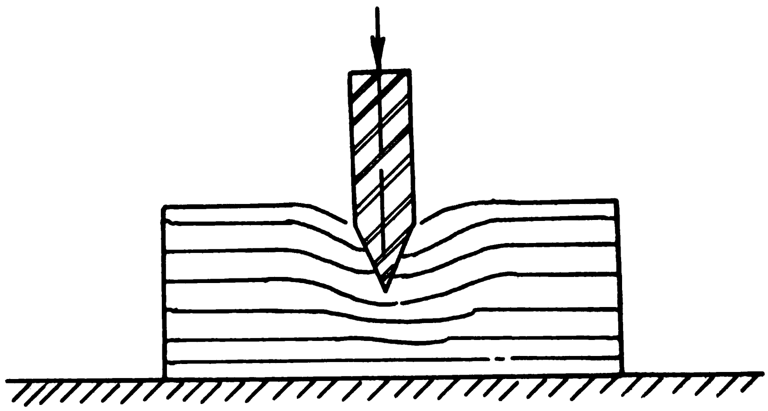


Figure 10.--Fiber deflection during cutting process.

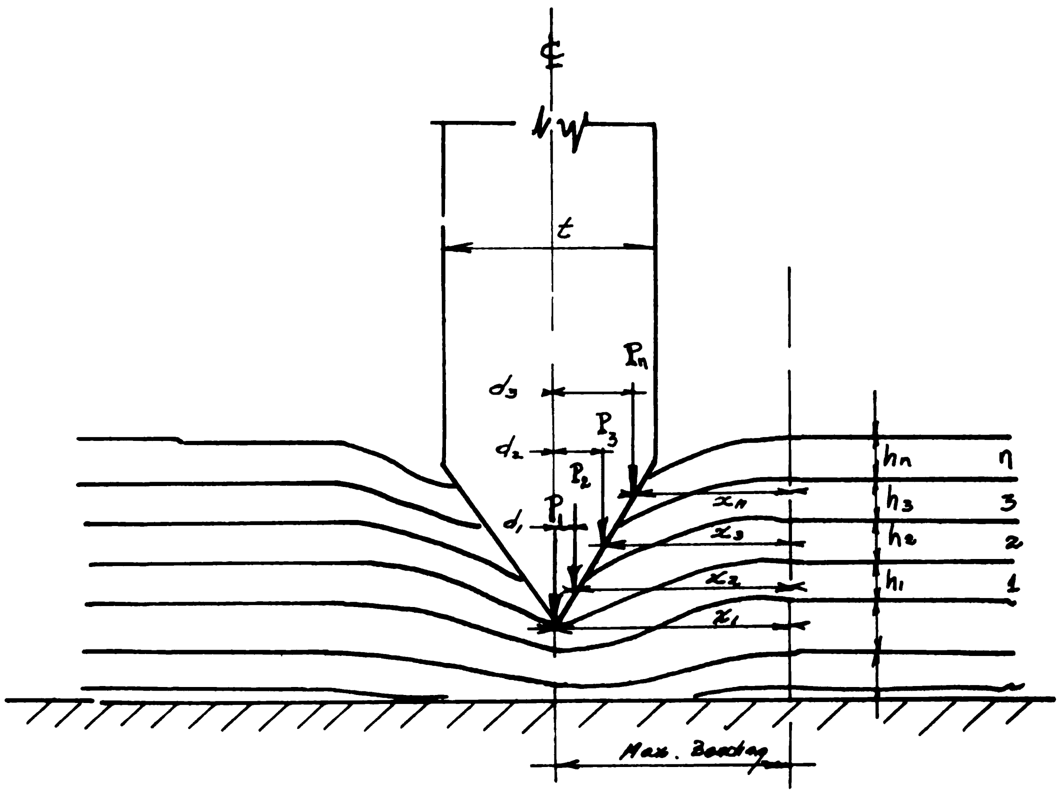


Figure 11.--Theoretical beam on elastic foundation.

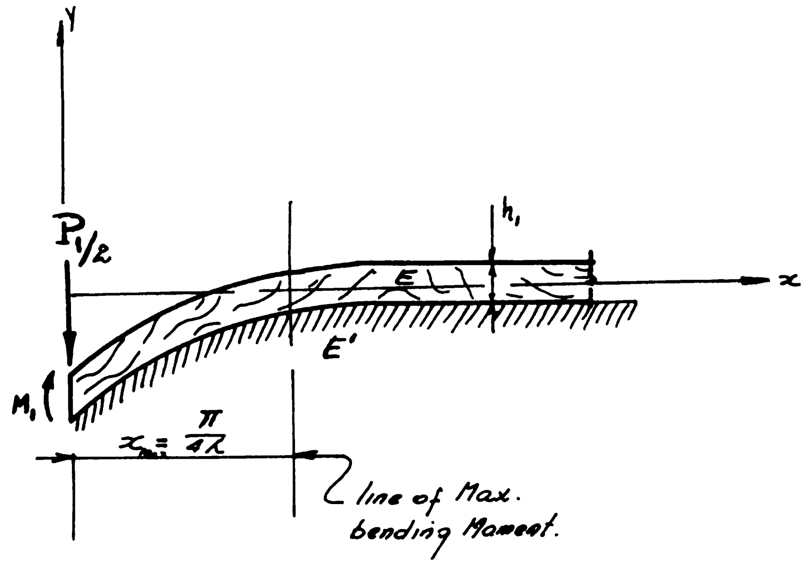


Figure 12.--Theoretical deflection of a fiber beam on elastic foundation.

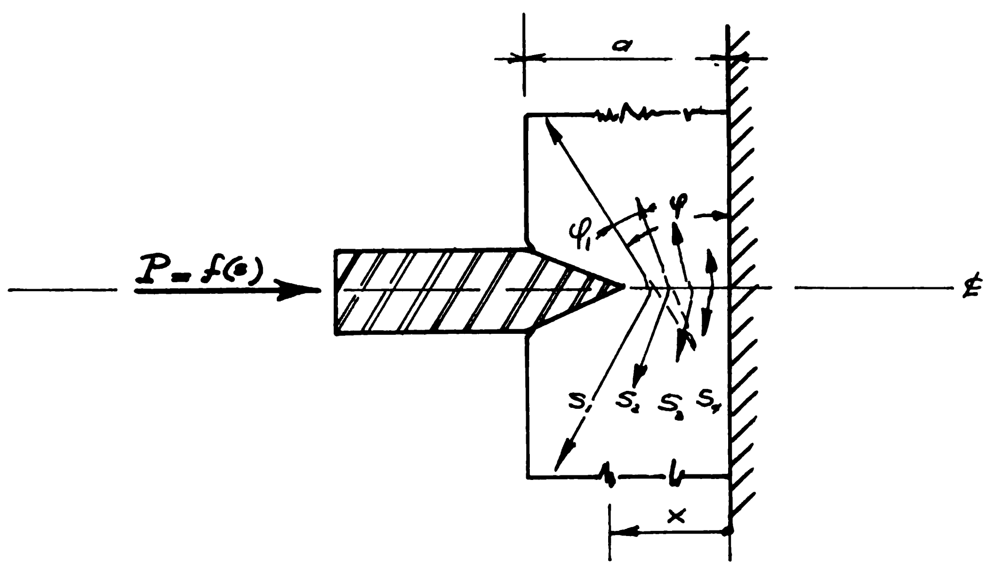


Figure 13.--Theoretical axial pulling stress.

P_2 , . . . , P_n are the wedging forces acting at the end of the cut layers.

t is the thickness of the blade

α is the edge angle

d_1 , . . . , d_n are distances from the line of symmetry of the blade to P_1 , . . . , P_n forces.

According to Hetenyi (1946), the bending moment in Mckenzie's beam is:

$$M = - P'_1 \, e^{-\lambda x} \frac{\sin \lambda x}{\lambda} \tag{2}$$

where $P'_1 = P_1/2$

$$\lambda = \sqrt[4]{\frac{K}{4EI}} \text{ ; foundation modulus} \tag{3}$$

K is a constant.

E is the beam elastic modulus

I is the moment of inertia of beam section.

According to Biot (1937), the constant K for a rectangular beam section is:

$$K = .645 \, \frac{bE'}{h} \left(\frac{E'}{E}\right)^{1/3} \tag{4}$$

where

b is the beam width

h is the beam height

E' is the foundation elastic modulus.

Equations (3) and (4) give:

$$\lambda = \frac{1.18}{h} \left(\frac{E'}{E}\right)^{1/3}$$

The location of the maximum bending moment is obtained as follows:

setting $\frac{d}{dx}(M) = 0$

$$\sin \lambda x = \text{cox } \lambda x$$

$$\tan \; x = 1$$

$$x_{max.} = \frac{\pi}{4\lambda} \qquad (5)$$

By replacing Equation (5) in Equation (2), the maximum bending moment becomes:

$$M_{max.} = -P_1' \, e^{-\pi/4} \, \frac{\sin \pi/4}{\lambda} \qquad (6)$$

Relating Equation (6) to the maximum stress (σ max.) that the beam could stand in compression,

$$\sigma_{max.} = \frac{M_{max.} \, h}{2I}$$

$$\sigma_{max.} = \frac{2.2}{bh} \; P_1' \; (\frac{E}{E'})^{1/3}$$

which yields:

$$P_1' = .455 \; bh \; \sigma_{max.} \; (\frac{E'}{E})^{1/3} \qquad (7)$$

And finally, Equation (7) gives the edge force:

$$P_1 = .910 \; bh \; \sigma_{max.} \; (\frac{E'}{E})^{1/3} \qquad (8)$$

Mckenzie relates the values E' and E with the elastic modulus perpendicular and longitudinal to the grain. E is equal to the longitudinal elastic modulus and E' is equal to 3.92 times the transverse elastic modulus.

When it is necessary to consider the pulling force along the fiber axis, as shown in Figure 13, Kempe (1965) gives the following Equation:

$$P_s' = \int_o^a 2 \; S \sin \phi \; dx \qquad (9)$$

where

$$S = S_1 \frac{\phi}{\phi_1}$$

$$\phi = \phi_1 \frac{x}{a}$$

P'_s is the edge force necessary to overcome the axial fiber tension in lb. per unit blade length. Therefore the force in terms of lbs. only is:

$$P_s = \ell \int_o^a 2 \frac{S_1 x}{a} \sin \frac{\phi_1 x}{a} dx. \qquad (10)$$

where ℓ is the effective blade length.

Adding up the components, the edge force finally becomes:

$$P_{ed} = P_1 + P_s \qquad (11)$$

It should be noticed that the edge force defined here is not completely analogous to the force experienced in actual practice. In realtiy the rings are circles of finite diameter whereas for theoretical consideration the rings were assumed to have infinite diameter.

3.2 Wedging Force

This force is the separating force which permits penetration of the blade through the wood. It originated from the normal pressure against the slope and the friction as the blade wedge moves. In order to find the theoretical value of the wedging force (P_w), the following assumptions are made: the pressure is uniform along the slope and the wedge section is symetric. The forces acting on one side of the symetric wedge are shown in Figure 14. From this figure the following relation can be derived:

$$Qh = Qv \tan \alpha/2$$

$$R = \frac{Qv}{\cos \alpha/2}$$

$$F = uR = \frac{u\ Qv}{\cos \alpha/2}$$

From conditions of equilibrium:

$$Pw/2 - F \cos \alpha/2 - Qh = 0$$

which yields:

$$Pw = 2\ Qv\ (u + \tan \alpha/2) \qquad (12)$$

where Qv is the vertical force acting on the slope and in practice represents the weight of the tree supported by the horizontal projection of the wedge slope, u is the coefficient of friction and α is the wedge angle.

3.3 Friction Force

These resisting forces act on both sides of the blade as it moves through the wood. In order to overcome these (F_f) forces, it is necessary to push the blade. Acting and reacting forces are shown in Figure 15. Assuming that pressure on both sides are uniform, the following relations are obtained:

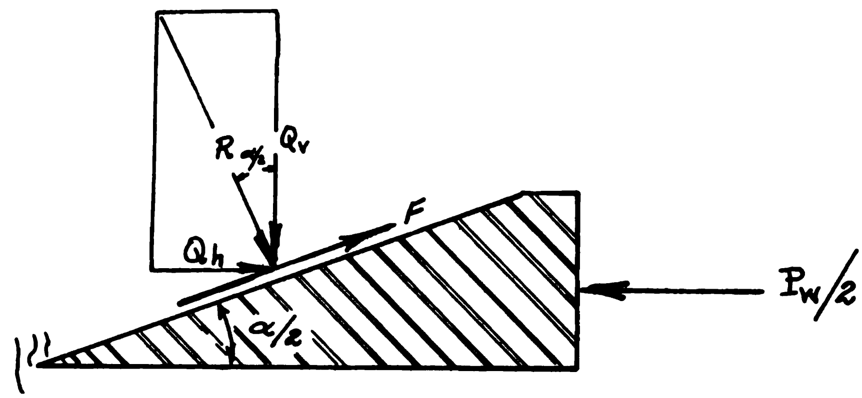


Figure 14.--Wedge force components.

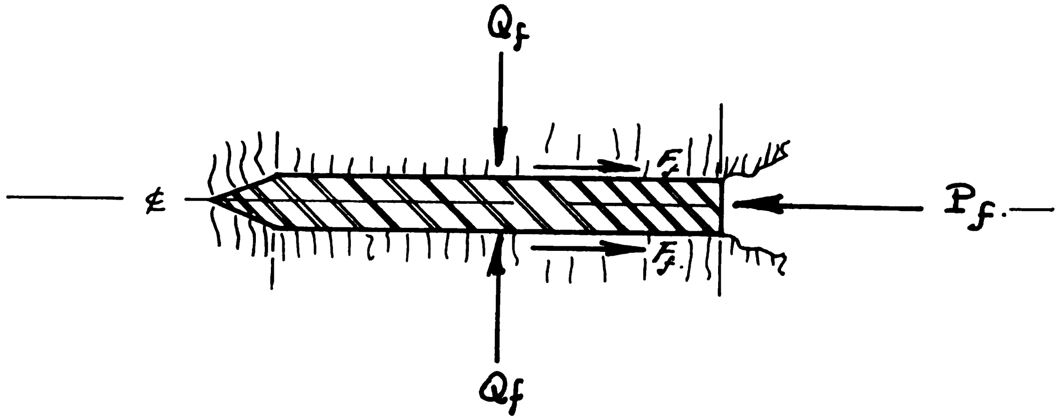


Figure 15.--Friction force components; flat blade.

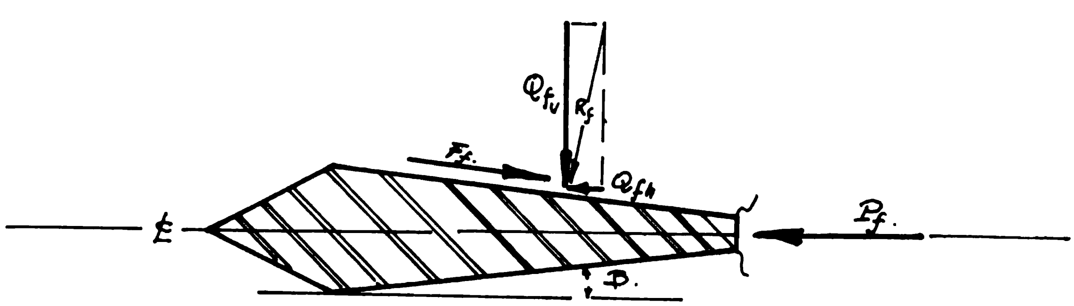


Figure 16.--Friction force components; backward sloped blade.

3.3.1 For Blade with Sides Parallel

From equilibrium condition:

$$P_f - 2u\ Qf = 0$$

which yields

$$P_f = 2\ u\ Qf \tag{13}$$

where Qf is the force perpendicular to the blade face, which in practice represents part of the tree weight supported by the blade, and u is the coefficient of friction.

3.3.2 For Blade with Faces Sloping Backward

This case is shown in Figure 16. By assuming a uniform pressure on the blade the following relations are obtained:

$$Q_{fh} = Q_{fv} \tan \beta$$

$$R = \frac{Q_{fv}}{\cos \beta}$$

$$F_f = u\ R$$

From equilibrium conditions:

$$P_f = F_f \cos \beta - Q_{fh}$$

Which yield:

$$P_f = 2Q_{fv}\ (u - \tan \beta) \tag{14}$$

where β is called the clearance angle. The clearance angle reduces the friction force as can be seen by comparing P_f in cases mentioned before.

In summary the theoretical equation for the force required by the blade is:

$$P_t = P_l + P_s + P_w + P_f \tag{15}$$

CHAPTER IV

SIMILITUDE AND MODELING

It is desired to determine the behavior of a prototype tyrough a model which is designed according to a predetermined scale.

Modeling is a laboratory technique which gives the possibility of simulating, as nearly as possible, any process as it happens in practice. The only inconvenience here is that scaling the material to be cut is difficult. Distortion between the model and the prototype is to be expected when working with biological material. The aim of this work is to measure the force required by the model and to predict that required by the prototype.

4.1 Characteristic of the Prototype

The assumed prototype is a double blade hydraulic shearing machine for felling rod wood to be used in mechanical harvesting of trees. It could be mounted on any standard forest tractor.

The prototype is designed to cut trees up to 18 in. in diameter but using geometrical similitude it is possible to design devices for trees of any diameter.

Under actual conditions directional felling is done by a hydraulic boom which is not a part of this study.

The machine's operating parts are two blades and two latches. The latches embrace the tree and guide the blades while cutting. The blades work against each other without overlapping. This orientation of the blades theoretically give the following advantages:

a. The force required could be less than that for a single blade, since with the absence of the anvil, the value E' from the theoretical equation is expected to be small.

b. Since the blade frame, tree and tractor constitute a static frame, the influence of P_s in Equation (10) and P_1 in Equation (8) may be reduced with two moving blades. P_s and P_1 are proportional to the blade displacement.

c. The damage to the fiber may be reduced since there is no anvil compression effect.

The blades are activated by two hydraulic cylinders through links with a mechanical advantage of approximately 1.2. The hydraulic power normally is supplied by the same tractor on which the machine is mounted, provided that the pump pressure is in the order of 2000 psi. and the minimum capacity is 20 GPM.

In Figure 18 the linkage that transmits the hydraulic power from the cylinder to the blade is shown; any linkage could be designed for this purpose provided that it occupies the same space and offers a mechanical advantage greater or equal to one. Figures 17, 18 and 18a show sketches of the prototype and latch mechanism.

4.2 Modeling Variables

The variables to be studies can be separated into three groups as follows:

a. Those relating to the material to be cut

- d diameter of the rod wood
- Q_v axial load which represents the tree weight
- σ maximum compressive stress that the wood can stand in bending.

b. Those relating to the model design

- α blade edge angle
- t blade thickness

c. Those relating to the machine oepration

- P total force required to operate the blade.

For the experiment, moisture content, coefficient of friction, weight density, blade speed, temperature and modulus ratio were considered constant.

The functional equation which involves the dependent and independent variables may be represented as follows:

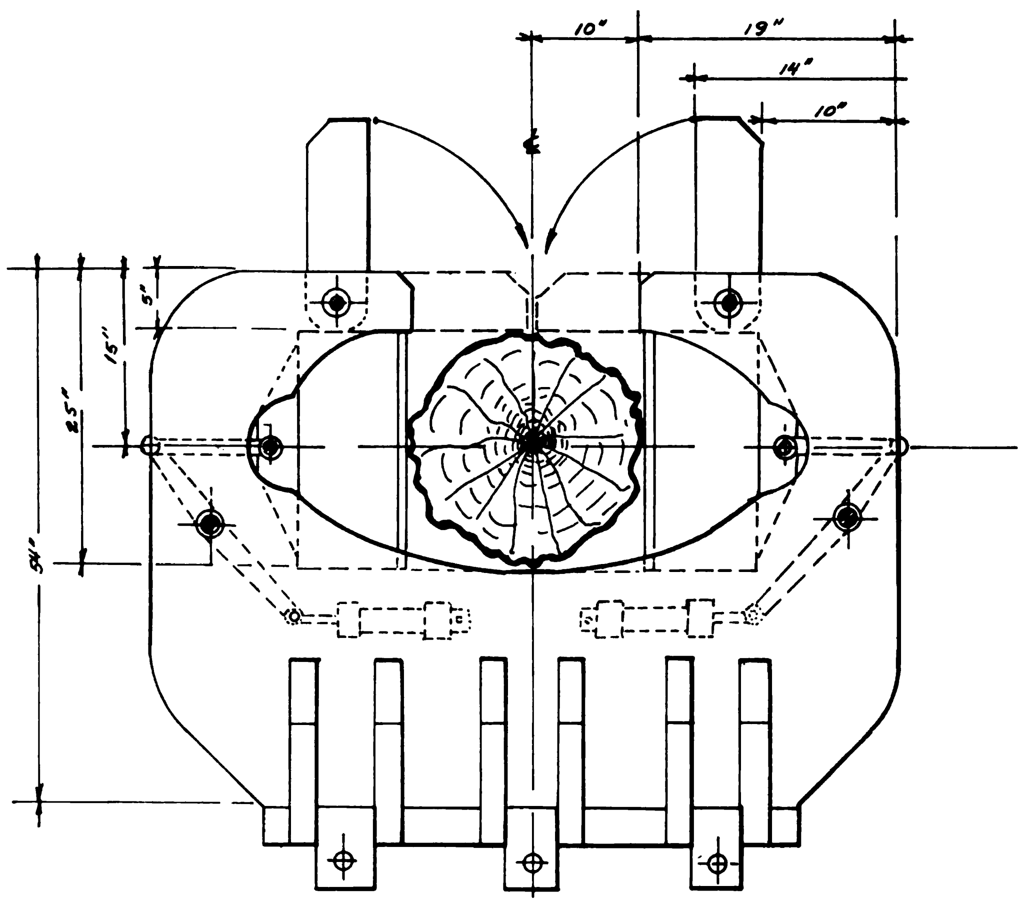


Figure 17.--Sketch of top view of the prototype.

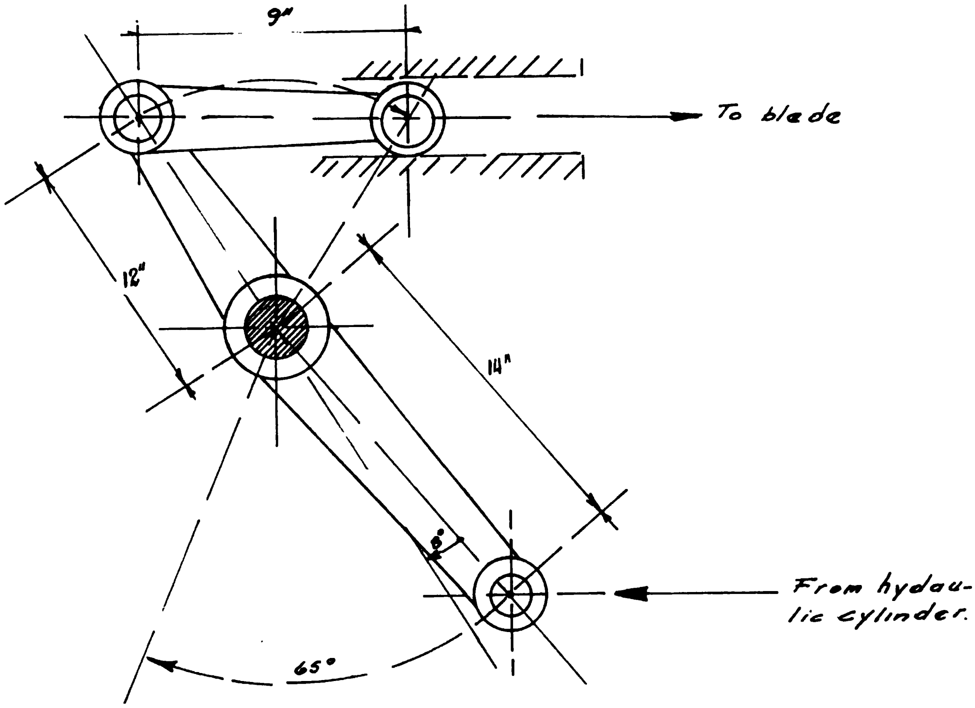
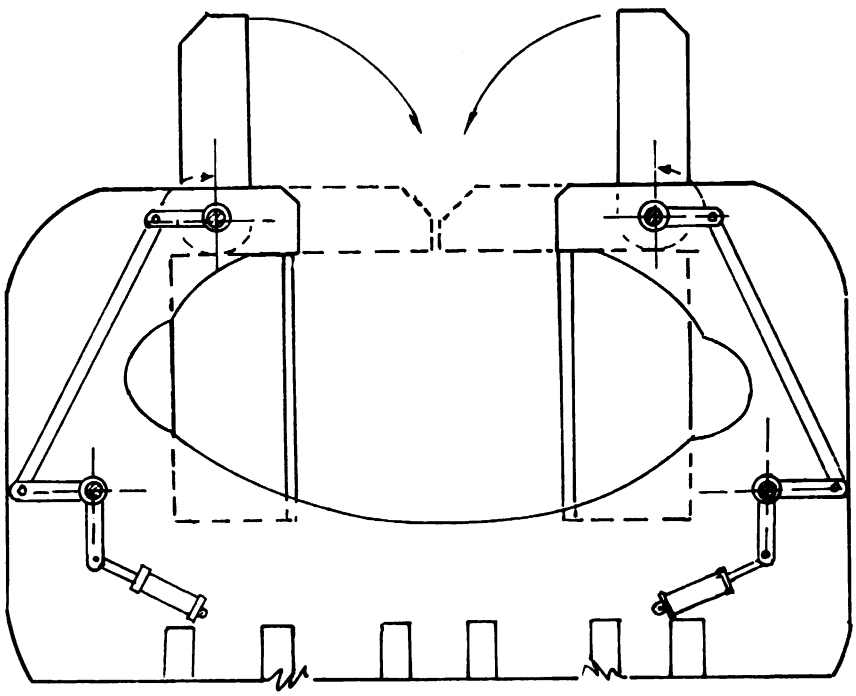


Figure 18.--Sketch of blade driver linkage.

Figure 18a.--Sketch of "latch" driver linkage.



$$f\ (P,\ t,\ d,\ Qv,\ \sigma,\ \alpha) = 0 \tag{15}$$

which yields:

$$P = F(t,\ d,\ Qv,\ \sigma,\ \alpha) \tag{16}$$

By applying the Buckingham Pi theorem from Dimensional Analysis the representative dimensionless equation was obtained:

$$\pi_1 = F(\pi_2,\ \pi_3,\ \pi_4) \tag{17}$$

where:

$$\pi_1 = P/Qv$$

$$\pi_2 = t/d$$

$$\pi_3 = \sigma d^2/Qv$$

$$\pi_4 = \alpha$$

then:

$$P/Qv = F(t/d,\ \sigma d^2/Qv,\ \alpha) \tag{18}$$

4.3 Prediction Equations

If we assume that no variable which affects the force required for the blade has been omitted, and no distortion is expected, the prediction equation will be:

$$P/Qv = P_p/Qv_p \tag{19}$$

and the design condition equations are:

$$t/d = t_p/d_p \tag{20}$$

$$\sigma d^2/Qv = \sigma p\ d_p^2/Qv_p \tag{21}$$

$$\alpha = \alpha p \tag{22}$$

where the subscript p represents the prototype.

CHAPTER V

MODEL CHARACTERISTICS

It is important to notice that if we assume that the arc displacement (Figure 18), formed by the link between the hydraulic cylinder and the blade in the prototype is negligible and the structure of the model is rigid enough, i.e., with very small deformation during loading, then the model is exactly geometrically similar to the prototype and the prediction equation can be applied.

The model was constructed to a 1/2 scale in the Agricultural Engineering Research Laboratory at Michigan State University. It mainly consisted of a steel frame supported by four legs; and two blades which slide toward each other in a severing operation (Figure 19). The blades are supported and guided by four square bars which can be adjusted to guide the blade motion in the plane of the cylinder.

In order to exert an axial force Qv when operating the model, a sliding table was built in the middle and over the blade frame (Figure 19).

The hydraulic system consisted of a pump, a power generator, two hydraulic cylinders, a relief valve, a reservoir and a control valve. The pump was a single

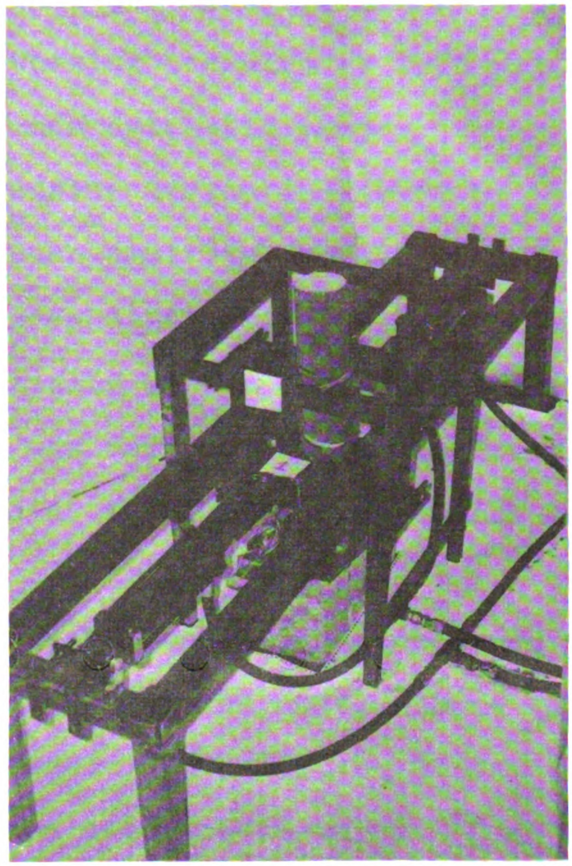


Figure 19.--Top view of the testing model.

1. Model blade
2. Sliding table support

vane Vickers, model V 111 E; the pump capacity was 4.3 GPM and a working pressure of 1000 psi. The power generator was a 5 hp., 1170 RPM, electric motor. The hydraulic cylinders had a 3 in. bore and an 8 in. stroke with a recommended working pressure of 1500 psi. Figure 20 shows the driver and hydraulic pump.

The force measuring device consisted of two strain gauges, one Bush amplifier (model BL 520) and a single Bush oscillograph (Figure 21). The two strain gauges were fixed axially on the blade rod driver; the strain gauge output was recorded by the oscillograph which was calibrated at 10 micro-inch per inch of strain per chart line.

For the first tests, which were concerned with determining the blade to fit the model, three blade thicknesses were chosen: 0.500 in., 0.250 in. and 0.187 in. These blades were not heat-tempered. Edge angles of 60°, 45° and 30° were used.

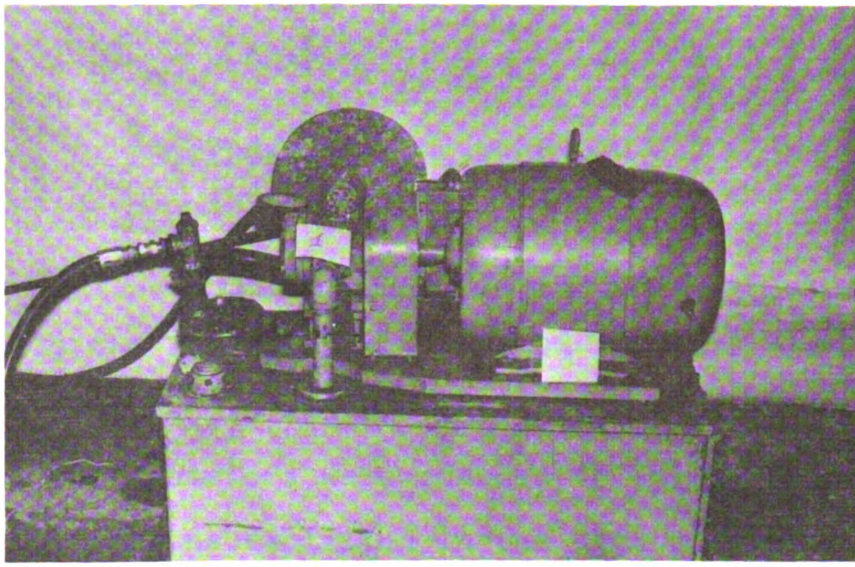


Figure 20.--Hydraulic power generator.

1. Vickers hydraulic pump
2. Electric motor

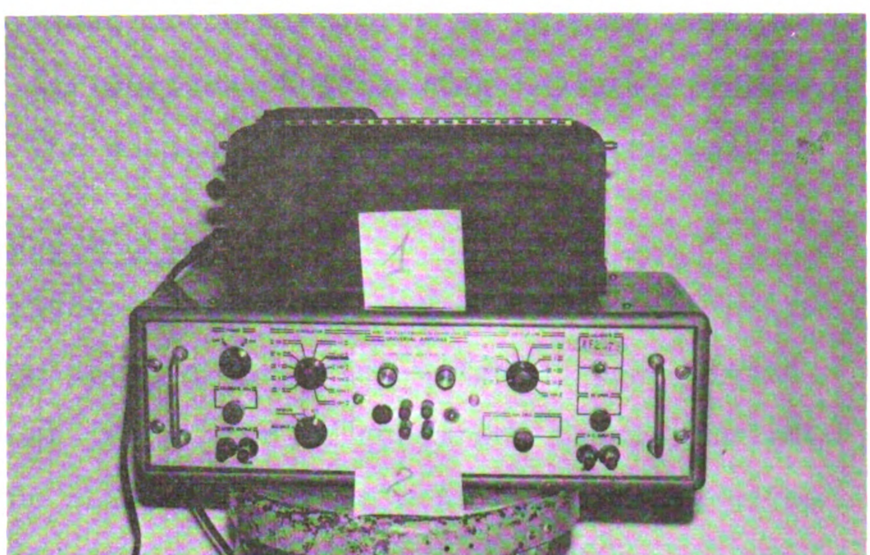


Figure 21.--Measuring equipments.

1. Bush oscillograph
2. Bush amplifier

CHAPTER VI

PROCEDURE FOR THE TEST

6.1 Blade Test

Initially the log specimens were 28 in. long and had diameters of 4.0, 6.0 and 8.0 in. After a preliminary test the diameters were changed to 3.125, 4.125, and 5.5 in. for reasons explained in the next chapter.

The samples were obtained from tree stems and were very heterogeneous. They showed an irregular distribution of knots and twisted growth especially in the 3.125 in. and 4.125 in. diameter samples.

The physical properties of wood samples were determined in the Agricultural Engineering Physical Properties Laboratory. The methods and results are given below.

6.1.1 Moisture Content

Small pieces of wood were taken from each sample. These sample pieces, after being weighed, were placed into an oven at 100°C. for 72 hours and the dry weight recorded. The moisture content w.b. was determined by using the following equation (23) and results are shown in Table 1.

$$\text{M.C.} = \frac{\text{Wet Weight - Dry Weight}}{\text{Wet Weight}} \times 100 \qquad (23)$$

TABLE 1.--Moisture content of samples.

Stem diameter in.	Moisture Content % w.b.
3.125	36.2
4.125	35.1
5.500	34.0

6.1.2 Maximum Bending Stress

The maximum bending stress is also called failure stress, and is the max stress used in equation (7).

To determine the bending stress, small beams 8 in. long and .375 in. x .250 in. cross section were prepared. Each beam was tested as a cantilever. The load was applied gradually until rupture, and the stress computed by using the bending equation (24).

$$\text{Max. Stress} = \frac{\text{Maximum bending movement}}{\text{Beam Section Modules}} \qquad (24)$$

Most of these small beams showed a great plastic deformation before breaking. The results are given in Table 2.

For comparison it might be important to remark that Kempe (1963) used Mckenzie's equation No. (8) with a

TABLE 2.--Maximum bending stress.

Stem diameter in.	Cross Section sq. in. *b	**h	Stress lb./sq. in
3.125	.375	.250	7,700
4.125	.275	.250	8,500
5.500	.375	.250	9,700

*b is width of the beam

**h is the depth of the beam.

maximum stress of 1,100 psi. Also Scofield (1963) reported that wooden beam failure stress is not constant; it decreases with beam depth. Therefore the great difference between values showed in Table 2 and Kempe's value could be due to beam size, mositure content and tree variety.

6.1.3 Stem Weight Density

Weight density was determined for each sample by applying Mohsenin's method. This method consists of weighing the sample, in air and water, on a platform scale.

In order to prevent the absorption of moisture by the wood sample, a very thin layer of grease was applied to the sample before weighing in water for density determination. The weight of the grease applied was considered insignificant compared to the weight of the sample.

The results are given in Table 3. The densities were determined by applying the following equation.

$$\text{Weight Density} = \frac{\text{Weight in air x Weight density of water}}{\text{Weight of displaced water}} \quad \frac{\text{lb}}{\text{in.}^3} \qquad (25)$$

TABLE 3.--Weight density of sample.

Stem diameter in.	Weight Density lb./cu. in.
3.125	.0232
4.125	.0262
5.500	.0259

The weight density values in Table 3 show that there is not too much difference among samples. Therefore we may consider that for a tree diameter range of 1.125-5.500 in., weight density is constant. Moisture content can also be assumed constant because of the small variation shown in Table 1.

6.2 Model Test

Stems from the same specimen as those used for blades were obtained for testing the model. From these stems, samples with diameters of 3.5, 4.0, and 4.5 in. in diameter were prepared. Each sample was 26 in. long and was cut twice at 8 in. intervals.

Here, again the samples appeared to be heterogeneous; irregular distribution of knots at the level of cut was observed, but twisted growth was much smaller.

Since the samples were prepared from the same specimen as those for the blade test, moisture content and weight density were not determined and were assumed to be similar to those given in Tables 1 and 2. The maximum stress was determined from the equation (24) for the model test. The results are given in Table 4.

TABLE 4.--Maximum bending stress.

Stem diameter in.	Cross Section in. x in. *b	**h	Stress lb./sq. in.
3.250	.375	.250	8,900
4.000	.375	.312	5,700
4.500	.375	.312	5,300

*letter b stands for width and

**h for depth.

It is evident from Tables 2 and 4 that the failure stress is not constant. This is in agreement with the results obtained by Scofield. Therefore, selecting a failure stress value as a model system parameter is a compromise.

6.3 Blade Speed

The capacity of the model hydraulic system was a limiting factor because the no-load speed of the equipment was limited to 2.0-2.5 i.p.s.

CHAPTER VII

DATA FROM THE TEST

7.1 Blade

Twenty-nine experiments were performed to test the three blades on the model frame. The force, P, required for each cut was obtained from the oscillograph chart.

The blades were made of steel plate. The blade of .500 in. thickness, after working with a flat shape, was changed to a grooved one. Figure 22 shows its characteristics.

The force, P, required for 0.5 in. blade with edge angles of 30, 45, and 60 degrees are tabulated in Table 5.

The force, P, required for the 0.25 in. blade is tabulated in Table 6 and for the 0.187 in. blade in Table 7. These forces are recorded for edge angles of 30, 45 and 60 degrees.

In Table 7 each sample was completely cut.

The values of P in Tables 5, 6 and 7, show that the performance of the .187 in. blade with a 30 degree edge angle was best. Therefore this blade was selected for the model test. However edge wear was observed especially

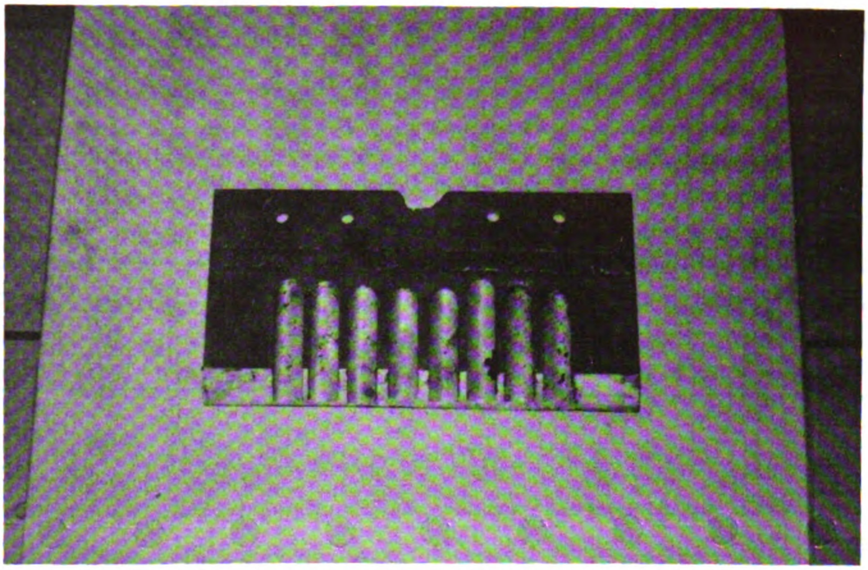


Figure 22.--0.5 in. grooved blade.

TABLE 5.--Force, P, required for the .5 in. blade.

	Edge angle degrees	Stem diameter in.	Force lb.
Normal Blade	60	3.125	7,800
	60	4.125	8,100
	60	5.500	9,300
	45	3.125	6,000*
	45	4.125	7,200
	45	5.500	7,200
	30	3.125	7,500
	30	4.125	7,800
	30	5.500	7,500
Grooved Blade	30	4.125	6,000*
	30	5.500	7,800

*The cut was complete.

TABLE 6.--Force required (P) for .250 in. blade.

Edge angle degrees	Stem diameter in.	Force lb.
60	3.125	5,100*
60	4.125	5,400*
60	5.500	7,800**
45	3.125	5,100*
45	4,125	7,200*
45	5.500	7,500**
30	3.125	5,100*
30	4.125	6,900*
30	5.500	7,500*

*The cut was complete.

**The cut was not complete.

TABLE 7.--Force, P, required for .187 in. blade.

Edge angle degrees	Stem diameter in.	Force lb.
60	3.125	5,400
60	4.125	6,000
60	5.500	7,800
45	3.125	4,200
45	4.125	5,100
45	5.500	7,800
30	3.125	4,500
30	4.125	5,100
30	5.500	7,500

when great knots were present; but this defect could be avoided by using heat-tempered steel.

As indicated in the tables the hydraulic system had a limiting working capacity of 7,500-7,800 lbs. Since this force was not great enough to handle the selected diameter, smaller diameter samples had to be used.

The model tests were done with .187 in. blade and a max. stem diameter of 4.500 in. The effect of this change on the accuracy of the Prediction Equation will be discussed in the next chapter.

7.2 Model

Twenty-two tests were run on the test model to get the necessary values to compute the Pi term.

The data for dimensionless terms, P/Qv and t/d, which represent π_1 and π_2 respectively, were obtained by varying d and Qv, and keeping the other dimensionless terms constant, i.e., π_3 which equals $\sigma d^2/Qv$ and π_4 which equals $\alpha°$.

The value of P/Qv and t/d are tabulated in Table 8.

TABLE 8.--P/Qv and t/d values, corresponding to $\pi_3 = 670$ and $\pi_4 = 30$.

Stem diameter in.	P lb.	Qv lb.	Dimensionless values t/d	P/Qv
3.250	4,800	85	.058	56.5
3.250	4,500	85	.058	53.0
4.000	7,500	212	.047	35.4
4.000	5,700	212	.047	27.0
4.500	5,700	172	.042	33.2
4.500	5,100	172	.042	30.0

A comparison of values of P in Table 4 for 4 inch diameter stem indicate that the mechanical properties of wood were not constant.

The data for dimensionless terms P/Qv and $\sigma d^2/Qv$, i.e., π_1 and π_3 were obtained by varying Qv while keeping t/d and α constant. The results are tabulated in Table 9.

TABLE 9.--P/Qv and $\sigma d^2/Qv$ values corresponding to π_4 = 30 and π_2 = .042.

Stem diameter in.	lb.	lb.	Dimensionless values P/Qv	$\sigma d^2/Qv$
4.500	7,500	110	68.0	1150
4.500	7,800	110	71.0	1150
4.500	6,900	115	60.0	1000
4.500	6,750	115	59.0	1000
4.500	5.750	145	39.0	800
4.500	5,900	145	41.0	800
4.500	7,200	220	32.75	520
4.500	6,300	330	19.10	350
4.500	6,900	330	21.00	350

The data for dimensionless terms P/Qv and α was obtained by changing α and keeping t/d and $\sigma d^2/Qv$ constant. These values were tabulated in Table 10.

Table 10 shows that the 30 degrees edge angle is the best for the model because the blade with such an angle requires the lowest P.

The performance of normal and grooved blades were plotted in Figure 23. A look at these curves motivates one to make a comparison between a normal and a grooved blade. Taking for instance a 4 in. diameter and assuming that the normal blade had made a complete cut, we arrive

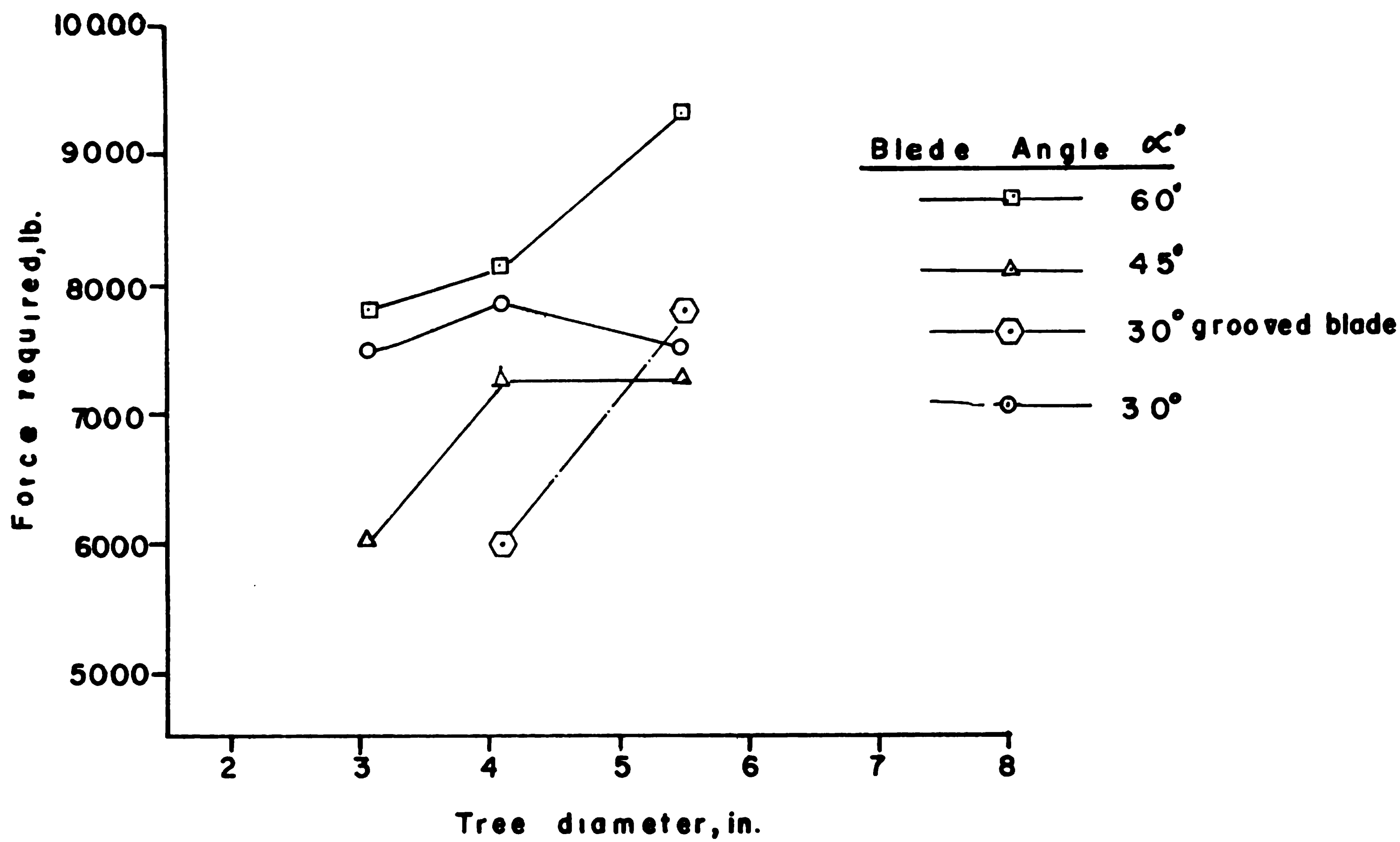


Figure 23.--Tree diameters vs. Force required for 0.5 in. blade.

TABLE 10.--P/Qv and α values corresponding to π_3 = 670 and π_2 = .042.

Stem diameter in.	P lb.	Qv lb.	Dimensionless values P/Qv	α
4.500	5,600	172	32.7	30
4.500	4,700	172	27.5	30
4.500	5.700	172	33.2	45
4.500	7,500	172	43.6	45
4.500	6,900	172	40.0	60
4.500	7,500	172	43.6	60

at the conclusion that grooves had reduced the force by 30 per cent. These curves also show the maximum force for 5.5 in. dia. which the normal blade in no case had cut completely, whereas the grooved blade did.

The .25 in. blade showed better performance than the 0.5 in. blade. It cut the samples completely except in two cases, that was at 45 and 60 degrees edge angles with a 5.5 in. tree diameter. This was attributed to sample knots. The values of P versus d for this blade are plotted in Figure 24. The unusual behavior of the required force for a 4.125 diameter sample can only be attributed to the variation of the mechanical properties of wood.

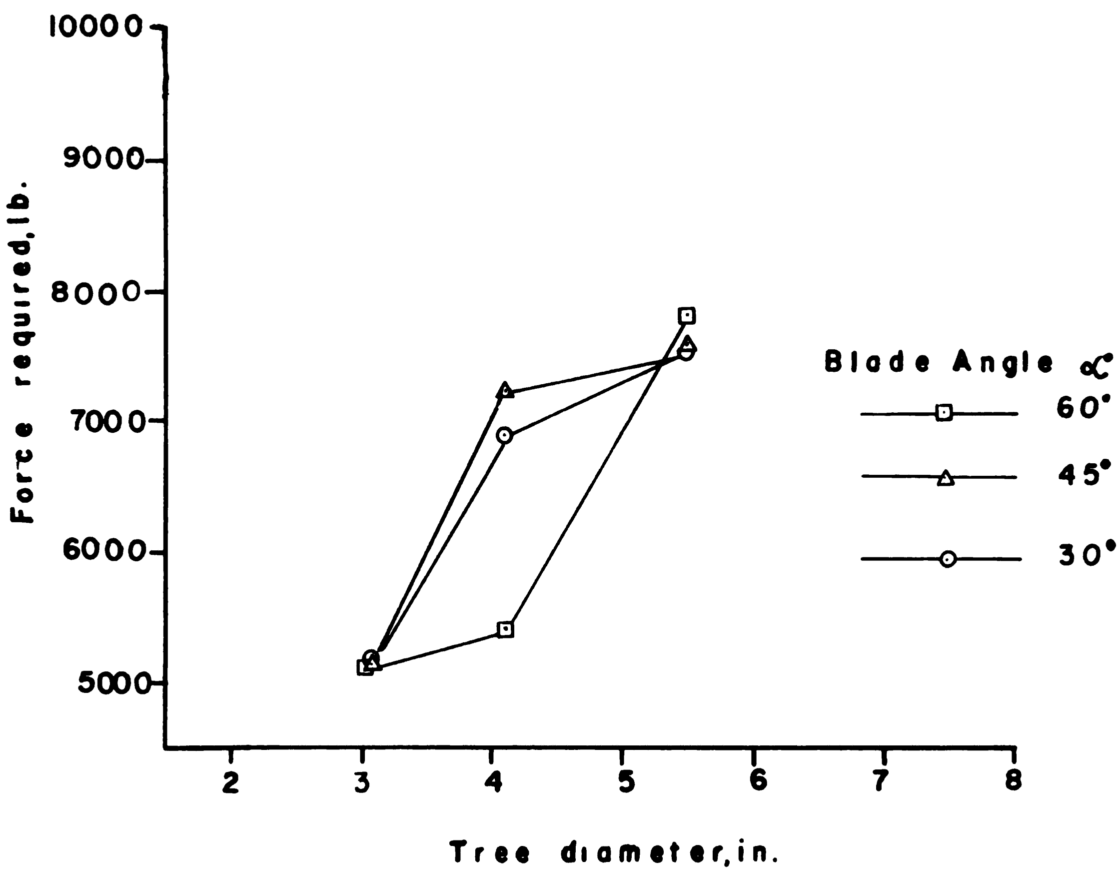


Figure 24.--Tree diameters vs. Force required for 0.250 in. blade.

The 0.187 in. blade showed the best performance. It cut each sample neatly and completely with all edge angles used. The values P versus d for this blade were plotted in Figure 25; these curves show that for a 30 degree edge angle, less force was required, especially with the 5.5 in. tree diameter. This blade was selected for the model due to its performance.

Finally the curves in Figures 24 and 25 show that the capacity of the hydraulic testing system was 7,500-7,800 lb. which was not enough for working with a 9 in. tree diameter as was planned.

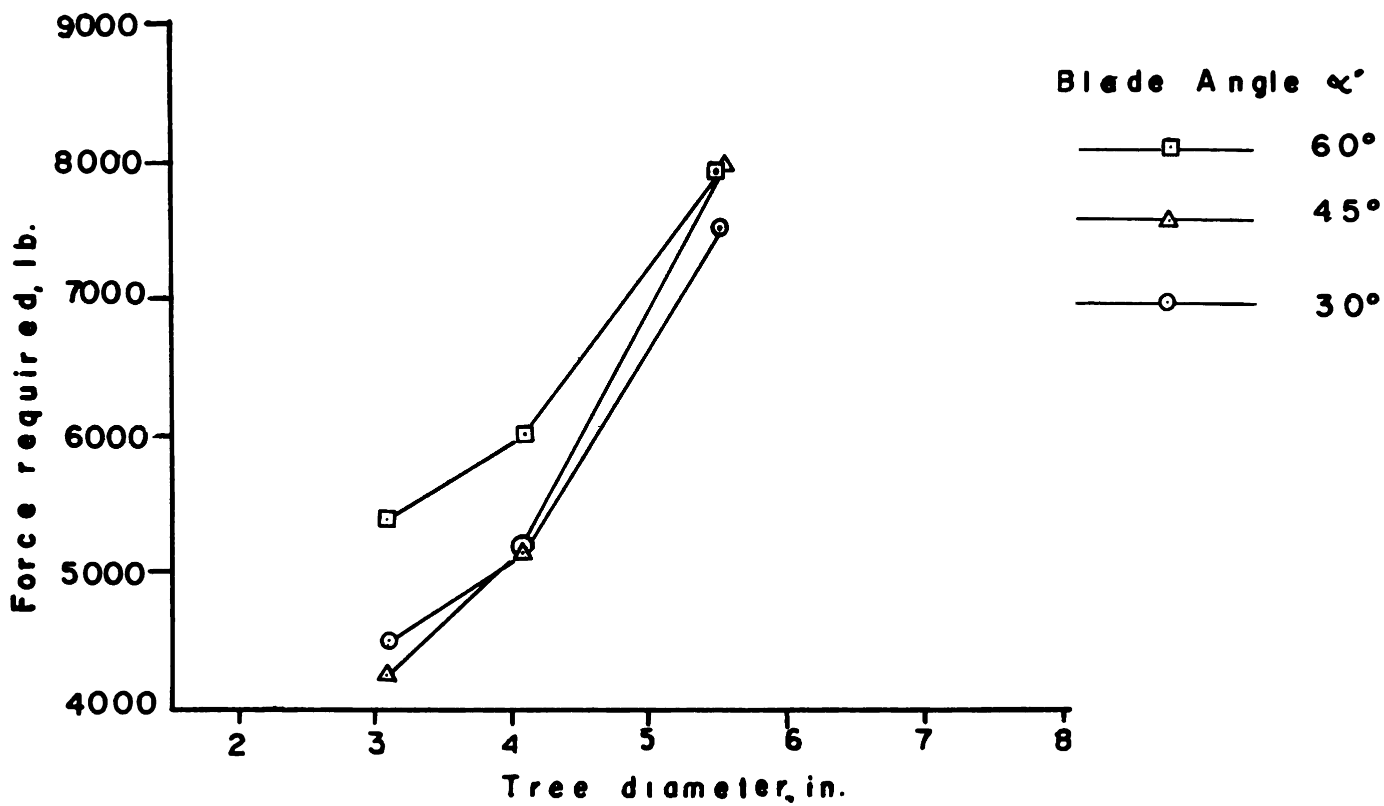


Figure 25.--Tree diameters vs. Force required for 0.187 in. blade.

CHAPTER VIII

RESULTS AND DISCUSSION

8.1 Blades

Part of the objective of this work, as it was mentioned before, was to determine the best blade.

The normal .5 in. blade gave a poor performance. It did not cut the sample completely and in some cases split the wood longitudinally. In one case this blade did make a neat cut.

The poor performance of the normal blade was attributed to the combination of thickness and edge angle. Therefore it was decided to change the shape of the blade in such a way that its stiffness was not affected. The 30 degrees edge angle was used because with these the acting edge remained sharp.

The blade shape was changed by making grooves parallel to the cutting direction, on both sides of the blade. The grooves dimensions were: 0.75 in. width, 4.0 in. long and 0.125 in. depth.

8.2 Model Test

The aim of these tests was to obtain the required data to compute the dimensionless terms in the Prediction

Equation. Given the Prediction Equation, the prototype behavior will be easily predicted.

According to Larson (1968), both model and prototype systems have to have geometric and dynamic similitude; geometric similitude means that the ratio of any two lengths of one system should be equal to the ratio in the second system; and dynamic similitude means that the ratio of any two forces in one system should be numerically equal to the corresponding force ratio in the second system.

However, when testing the model, the requirements mentioned above were not completely satisfied. The geometric condition was not satisfied because of the low capacity of the hydraulic system of the model. The tests were done with 4.5 in. diameter instead of 9.0 in. This caused a distortion in the Prediction Equation. The dynamic condition also was not satisfied because the sample, being a biological material behaved as a heterogeneous and anisotropic system. Mechanical properties of wood are not constant even in the same diameter sample. Therefore the measured force (P) was somewhat greater when knots were encountered. Since Qv is not affected by knots, the ratio P/Qv for the model was somewhat larger; i.e. distorted.

Distortion is common when working with models, especially when parts of the system do not have constant properties. However, distortion can be evaluated in order to correct the Prediction Equation by running similar models of two or more scales.

The Pi terms for the model were plotted in Figures 26, 27 and 28. Because of the distortion explained above, the relationship between π_1 and π_{1p} will be:

$$\pi_{1p} = \delta\pi_1$$

where the subscript p stands for the prototype and δ is the factor required to correct for distortion. The value of δ could be equal to, less, or greater than one. Since it is not within the scope of this work to evaluate δ, it will be assumed to be equal to 1 and then:

$$\pi_{1p} = \pi_1$$

From Figure 26, the value of $\pi_1 = \underline{P}p/Qv = 30$, then the force required for the prototype will be:

$$\underline{P}_p = 60{,}000 \text{ lb.}$$

when:

Qv = 2000 lbs.

tp = .75 in. blade thickness

dp = 18 in. tree diameter

α = 30 degrees edge angle

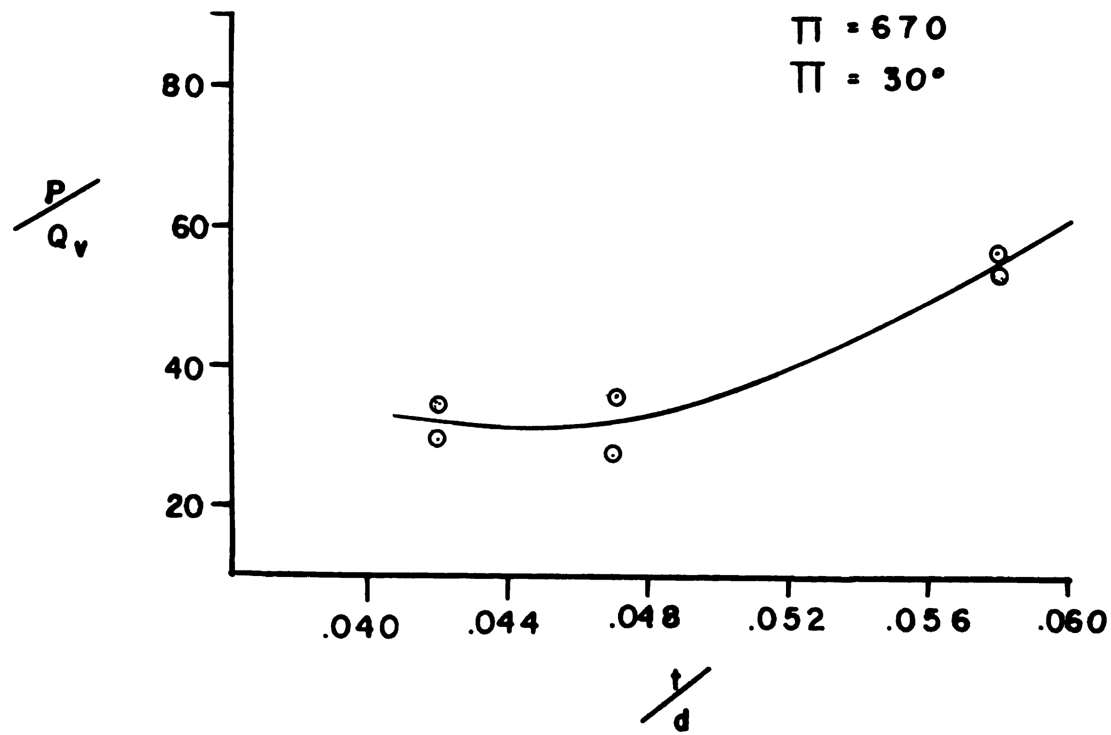


Figure 26.--Dimensionless value P/Q_v vs. t/d.

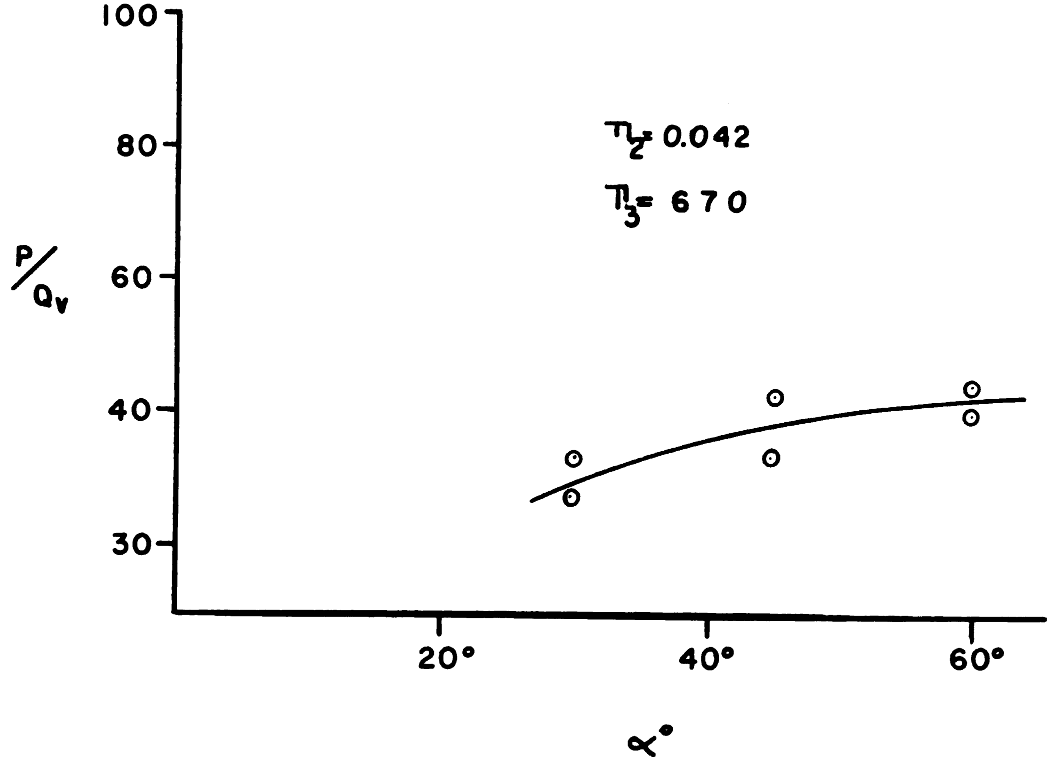


Figure 27.--Dimensionless value P/Q_v vs. α.

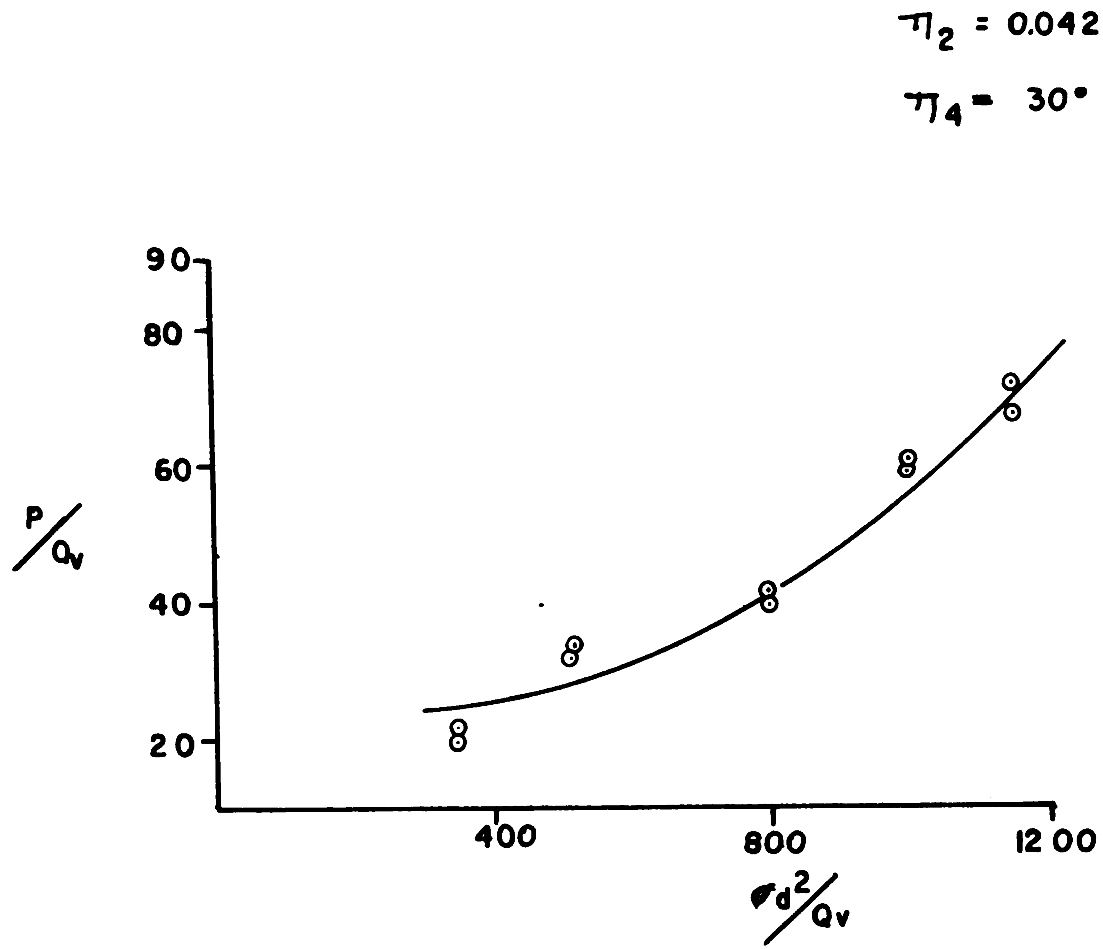


Figure 28.--Dimensionless value P/Q_v vs. $\sigma d^2/Q_v$.

The calculated value of Pp is for a normal blade; however during the discussion on blade tests it was shown that for the grooved blade, a 30% force reduction could be obtained. Therefore by using grooved blade in the prototype the actual force should be:

$$P = 42,000 \text{ lb.}$$

when:

t = 0.75 in. grooved blade thickness

d = 18 in. tree diameter

α = 30 degrees edge angle

CHAPTER IX

SUMMARY

A model for shearing rod wood, used in forest mechanization, with two blades traveling parallel and toward each other without overlapping was built and tested in the Research Laboratory of the Agricultural Engineering Department at Michigan State University.

Three blade thicknesses, 0.5, 0.25, and 0.187 in., were tested with blade angles of 30, 45 and 60 degrees. Wood samples of 3.125, 4.125 and 5.5 in. diameters were used in the test.

Twenty-nine tests were made in order to determine the best blade for the model. The 0.187 in. blade with 30 degrees edge angle, proved to be the best; therefore it was selected to be used in the model. Although the 0.5 in. blade had the poorest performance, by changing its shape it was possible to improve its performance.

Twenty-two tests were made with the model to obtain the data for computing the Pi terms, which appear in the Prediction Equation.

In solving the prediction equation for a prototype shearing tree 18 in. in diameter the required force was found to be 42,000 lbs. which is considered less than values reported in literature. The prototype blade must be 0.75 in. thick and of the grooved design.

CHAPTER X

CONCLUSIONS

1. Hardness as blade material property is very important when designing shears for felling rod wood. Hardness is related to the time that a blade can hold its sharpness. Therefore this property should have the highest possible value when designing.

2. Blade thickness, edge angle and shape have a great influence in the force required for shearing. Of the edge angles tested (30, 45 and 60 degrees) the 30 degree edge angle gave the best results.

3. The blade thickness effect can be reduced by 30 per cent by grooving both blade sides. This treatment did not seem to affect the blade stiffness for the blades tested.

4. Wood knots increase the force required for shearing and reduced the blade sharpness.

5. The terms in the design condition equation could not be precisely satisfied because of the varying properties of the biological materials used. These properties include such parameters as weight density, failure stress and so on.

CHAPTER XI

SUGGESTIONS FOR FUTURE STUDY

1. Further tests should be made with blade thicknesses and edge angles. Blade materials with high stiffness and an improved hardness must be used.

2. Model frames should be designed in such a way that material deformation can be kept at a minimum. This is particularly important for devices in which the blades do not overlap.

3. Further works should be conducted with models of different scales to determine the distortion factor and thereby enable correction of the prediction equation.

4. Finally, based on the corrected prediction equation, a prototype should be built and tested in actual field conditions.

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