THE UNIVERSITY OF MANITOBA

DESIGN AND TESTING OF ZERO - TILLAGE PLANTING EQUIPMENT

by

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ABSTRACT

Zero-tillage studies have indicated advantages over conventional methods in reducing costs and labor requirements for seedbed preparation, in reducing soil erosion and in increasing soil moisture for plant growth and crop yield. There has been no machinery specifically designed for zero-tillage planting.

In this study a zero-tillage planting machine was designed by attaching cutting disks ahead of the furrow-openers on a standard pressdrill. The function of the cutting disks was to cut heavy trash and to open a small furrow for furrow openers to put seed into the soil. The additional horsepower required for the cutting disks was measured. The relationship between the additional horsepower required for the cutting disks and the depth of penetration was determined.

The machine operated very well and the cutting disks had good penetration on very fine sandy loam soils at Carman. Good penetration of the cutting disks was not initially obtained on Red River clay soil at Glenlea. Adequate penetration was obtained after the drill was ballasted with additional weight.

The additional horsepower required for the cutting disks depended on the depth of penetration, the type of soil and the speed of operation. The machine could also be used for both conventional and zero-tillage purposes.

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CHAPTER I

INTRODUCTION

Tillage has been considered one of the most important operations in crop growing. A lot of time, money and effort are spent in tillage to prepare a suitable seedbed. Tillage methods, to provide a good seedbed for plant growth, have been extensively studied. However, there is no tillage method which is suitable for all soil conditions. An unsuitable tillage method can destroy soil physical properties and soil organic matter.

In recent years, zero-tillage methods have been introduced in the United States of America in order to reduce soil disturbance, soil erosion and time and cost of seedbed preparation. Zero-tillage corn acreages in Ohio were less than 1500 acres in 1964 but they were about 50000 acres in 1969. These figures show a good acceptance of zero-tillage methods. Studies of zero-tillage in the US, Europe and Canada have indicated that zero-tillage as well as other tillage methods were not suitable for all soil conditions. Zero-tillage methods appear to have advantages on silt-loam textured soils. However, there still are two main problems with zero-tillage methods. These problems are:

1. Present herbicides have not been able to control all weeds (4).

2. There is no machine specifically designed for zero-tillage purposes.

This study was designed to overcome this second problem. The two main objectives were:

1. To fit a standard double disk pressdrill with cutting disks so that the drill would be able to plant cereal grains and oil seeds in zero-tillage conditions with heavy trash cover.

2. To determine additional horsepower requirements for operation with the cutting disks.

CHAPTER II

REVIEW OF LITERATURE

2.1. Definition of Tillage

Tillage may be defined as the mechanical manipulation of soil for any purpose. In agriculture some of the objectives of tillage are (8):

1. To develop a desirable soil structure for a seedbed or a rootbed.

2. To control weeds or to remove unwanted plants.

3. To manage plant residues.

4. To minimize soil erosion by following such practices as contour tillage, listing and proper placement of trash.

5. To establish specific surface configuations for planting, irrigating, drainage, harvesting operations, etc.

6. To incorporate and mix fertilizers, pesticides or soil amendments into the soil.

7. To accomplish segregation. This may involve moving soil from one layer to another, removal of rocks and other foreign objects, or root harvesting.

2.2. Tillage Systems

Most tillage can be classified into three different systems:

1. Conventional tillage includes primary and secondary tillage for seedbed preparation. A primary tillage

operation constitutes the initial, major soil-working operations; it is normally designed to reduce soil strength, cover plant materials and rearrange aggregates. Secondary tillage operations are intended to create refined soil conditions following primary tillage (8).

2. Minimum tillage provides the minimum of soil manipulation necessary for crop production under existing soil and climatic conditions. Minimum tillage does not define a system of tillage, but generally refers to a system with fewer tillage operations than some conventional tillage systems. This implies the employment of substitute techniques for weed control and/or seedbed preparation (15). The major objectives of minimum tillage are (8):

(a) To reduce mechanical energy and labor requirements.

(b) To conserve moisture and reduce soil erosion.

(c) To perform only the operations necessary to optimize the soil conditions for each type of soil within a field.

(d) To minimize the number of trips over the field.

3. Zero-tillage or no-tillage has the same purposes as minimum tillage but in this system there is no soil preparation. In other words, zero-tillage is a system in which a crop is planted directly into a seedbed which is untilled since the harvest of the previous crop.

2.3. Characteristics of zero-tillage

A comprehensive discussion of zero-tillage can be found in reference number one. The main characteristics of zero-tillage as discussed in the above reference are summarized in the following.

2.3.1. Historical background In 1927, Garber successfully introduced a legume into an unproductive grass sod without tillage. He used simple techniques such as close grazing or burning and heavy seeding rates to manipulate the competition between the old grass and the surface-sown forage. This idea was believed to be the first introduction of a zero-tillage system. Zero-tillage systems became more feasible in the 1950's when selective herbicides were improved.

2.3.2. Operation of zero-tillage systems Zero-tillage machinery should perform three tasks in one operation. The three tasks are to open the soil for seed insertion, to place the seed properly and to cover the seed adequately. Before planting, nonselective herbicides with short residual effects must be applied to completely destroy the initial vegetation. Selective herbicides are also needed during subsequent growth phases.

2.3.3. Comparison of environmental conditions in tilled and untilled soils Research has shown that untilled soil surfaces were relatively smooth, even and more dense than tilled soil surfaces. Thus soil aeration under untilled soil was reduced (1). Differences in soil moisture content were relatively small between tilled and untilled soil. With a similar soil moisture content, untilled soil generally had less resistance to water uptake by plants.

Mulch cover on untilled soil acted as an insulator. Thus soil temperature on the surface of untilled soil was lower than in tilled soil. In the subsoil the reverse was found. Resistance to soil erosion by both water and wind was larger for untilled soil due to mulch cover and dense soil surface conditions.

Higher decomposition rates and lower concentration of available nitrogen were also observed in untilled soils.

2.3.4. Effects of zero-tillage on plant growth

Higher numbers of emerged plants were observed under zero-tillage on light to medium textured soils with sod cover and friable soil surfaces. But thick mulches may smother emerging plants. Zero-tillage crops were observed to grow faster due to an increase in available water and suitable root zone environment.

Root growth was lower for zero-tillage due to high resistance to root growth in undisturbed soil especially during early vegetative phases (1). Annual and perennial weeds increased in zero-tillage systems due to faulty weed control by chemical means.

2.3.5. Crop yields Crop yields under zero-tillage systems have depended largely on the type of soil. On soils that range from clay to clay loam, zero-tillage crops produced less than conventional tillage crops. On medium textured soils, zero-tillage crops generally produced equal or higher yields.

2.4. Advantages of Zero-Tillage System

2.4.1. Soil moisture content under zero-tillage systems was increased due to killed sod cover. Soil moisture in the top 0 to 8-cm soil layer under zero-tillage was significantly higher than under conventional tillage throughout the entire growing season (3).

2.4.2. Soil aeration was improved since excessive tillage produces small pore spaces which tend to retard seed germination and early growth (2). Repeated tillage operations can result in soil compaction.

2.4.3. Zero-tillage practices resulted in less soil resistance to root penetration throughout the growing season and lower bulk density as compared to conventional tillage (10).

2.4.4. An experiment in the western corn belt showed that seed zone temperature in zero-tillage systems was lower than in conventional systems (9). Soil temperature under zero-tillage systems was slightly lower than under conventional systems early in the growing season (10). Reduced soil temperature may have advantages in hot regions but may be detrimental in warm or cold regions.

2.4.5. Soil erosion at the rate of 0.06 tons per acre was found with zero-tillage while it was 2.8 tons per acre with conventional tillage (7). The resistance to erosion was due to the mulch cover. Soil loss of 0.4 tons per acre was found with soil which had mulch cover whereas soil loss with no cover was 2.8 tons per acre (11).

2.4.6. Zero-tillage systems generally produced higher corn yields during years of either poor or favorable rainfall distribution (3). Crop yields with zero-tillage have generally equalled or exceeded those obtained with conventional tillage (14).

2.4.7. Zero-tillage reduces the number of field operations, labor, machinery requirements and also saves fuel (5).

2.5. Disadvantages of Zero-Tillage System

Zero-tillage methods cannot be applied to all types of soil. The most suitable soil types for zero-tillage have been light to medium textured soils. Zero-tillage methods have been most successful with crops having small seeds (1).

CHAPTER III

DESIGN OF ATTACHMENT

A zero-tillage planting machine was developed from a standard press drill by attaching coulters or cutting disks ahead of each double disk furrow opener to cut crop residues or trash. The coulters were designed to raise and lower to the desired depth of penetration into the soil independently from the double disk furrow openers.

A press drill which had double drawbars for each double disk furrow opener was more convenient for adaption because the coulters could be placed between the drawbars of the double disk furrow openers. This made it easy to line up the double disk furrow openers with the coulters.

An International 620 press drill was selected to be adapted in this study. All attachment parts are shown in Drawing No. 1 (see back cover).

3.1. Adaption of Drawbar of Double Disk Furrow Openers

Originally, the arrangement of the drawbars of the double disk furrows openers were staggered (Fig. 3.1). The shorter drawbars of the double disk furrow openers were lengthened to be the same length as the longer drawbars (Fig. 3.2). This provided space for the coulters ahead of each double disk opener.

3.2. Coulter Gang

Disks of 17 in. diameter were selected as coulters

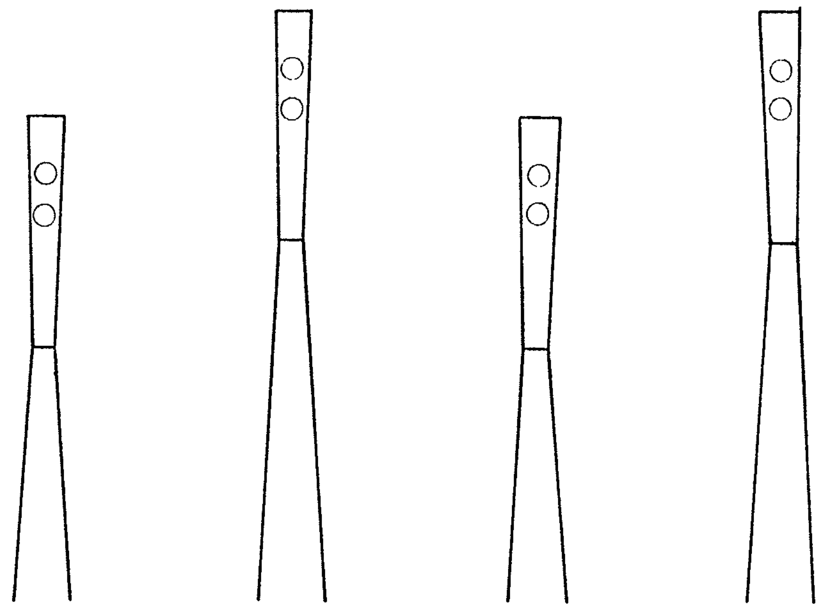

Figure 3.1. Drawbars of the double disk furrow openers before adaptation (top view).

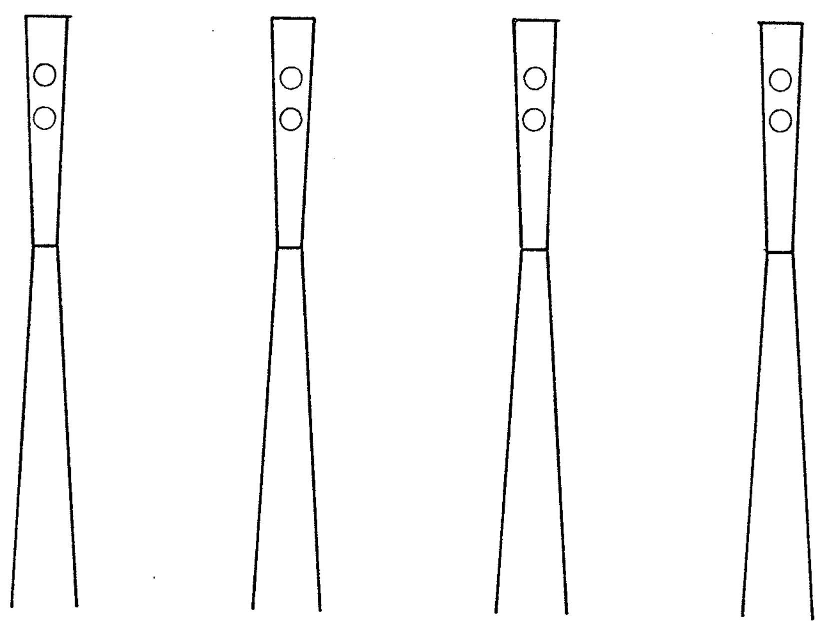

Figure 3.2. Drawbars of the double disk furrow openers after adaptation (top view).

which were mounted on four gangs of four. Each gang consisted of four cutting disks 6 in. apart. Each disk was sandwiched by two collars (3/8 in.x4 in. diameter). The collars were welded to a spacer (1 in. XS pipe). Each gang was held together by a 3/4 in. diameter bolt running through the spacers. Centering washers were used at both ends of the gangs to hold the bolts on center. Coulter gang drawbars were made of 3/8 in. x 3 in. mild steel with one end connected to the spacers by means of a bearing mount and the other end welded to a bearing pipe (2 in. XS pipe).

3.3. Shaft Support

The shaft supports consisted of 1/2 in. x 3 in. x 3/6 in. channels 12 in. long and two pieces of 2 in. XS pipe 4 in. long for upper and lower shaft support bushings. The upper shaft support bushing was welded on the front of the channel and the lower shaft support bushing was welded on the back of the channel.

3.4. Rotation Linkage

The rotation linkage consisted of double upper lift arms, a connecting link and double lower lift arms. All of these were made of 5/16 in. x 2 in. mild steel and were connected by pins.

The maximum depth of penetration of the coulters into the soil was designed to be three inches. The coulter gang drawbars were 10.90 in. long. This length was graphically determined so that there was no interference between the coulters and the drawbars of double disk furrow openers when they were raised and lowered. The maximum clearance of the coulters above the ground level was 3.4 in. This meant that the coulter rotated 42 degrees from transport position to maximum operating depth.

The upper and lower lift arms were arbitrary designed to be 6.35 in. and 6.875 in. long (center to center) respectively. The lower lift arms were positioned in a horizontal position when the coulters were at the maximum depth position. Graphical methods were used to size the connecting link at 8.25 in. long (center to center). The angle between the lower lift arm and the coulter gang drawbar was 124 degrees. The upper lift arm rotated 47 degrees for the 42 degrees rotation of the lower lift arm.

3.5. Hydraulic Cylinder Support

A hydraulic cylinder was used to raise and lower the coulters. An eight inch stroke double acting hydraulic cylinder was selected. The hydraulic cylinder attachment was determined by two conditions:

(1) The hydraulic cylinder had to be fully retracted when the coulters were at the maximum depth position.

(2) The hydraulic cylinder had to be fully extended when the coulters were in transport position.

With these two conditions, the coulters stopped automatically at the maximum depth position and at the transport position without any interference with the drawbars of the double disk furrow openers. Depth of penetration of the coulters into the soil could be controlled at any depth up to the maximum depth.

The hydraulic cylinder supports consisted mainly of a column and a hydraulic control lever. The lengths of the column and the hydraulic control lever were determined graphically to be 24.25 in. and 11.5 in., respectively. The angle between the hydraulic control lever and the upper lift arms was 86 degrees.

3.6. Assembly

The shaft supports were attached to the drill frame at spacings shown in Drawing No. 1. The coulter gangs were placed beneath the drawbars of the double disk furrow openers supported by the lower shaft (1 1/2 in. XS pipe). The lower shaft had a free running fit with the bearing pipes and the lower shaft support bushings. To prevent the coulter gangs from moving from side to side, two locking collars were used at each end of the bearing pipes. Before the locking collars were locked, each coulter gang was lined up with the double disk furrow openers. The upper shaft (1 1/2 in. XS pipe) was run through the upper shaft support bushings. The rotation linkages were fixed to the upper shaft and to the bearing pipes at the spacing shown in Drawing No. 1. The hydraulic control lever was fixed to the upper shaft and the hydraulic cylinder support was mounted on the drill frame.

3.7. Analysis of Design

The design of the machine elements was determined largely by the kinematic requirements. The actual sizing of the members was based on materials that were in stock or were readily available. The design was further complicated by the fact that the loading of the various members was unknown.

The loading of the attachment parts would be determined by the loads transferred to the coulter gangs by the penetration resistance force of the soil. The maximum loading, neglecting impact loading, would occur if the sixteen coulters were to support the total loaded weight of the press drill in a situation where there was no penetration of the coulters due to extremely hard soil conditions. Under these severe conditions the vertical load on each coulter would be approximately 300 lb (full load capacity of the drill plus total weight of the drill). The positions of the machine elements are shown in Figure 3.3.

The stresses were estimated for the machine elements that were considered critical. The allowable design stresses are listed in Table 3.1. These values were calculated with a factor of safety of three (N = 3) based on ultimate stress and using allowable bearing stress equal to the allowable tensile stress. The torsional deflection allowed was one degree per foot.

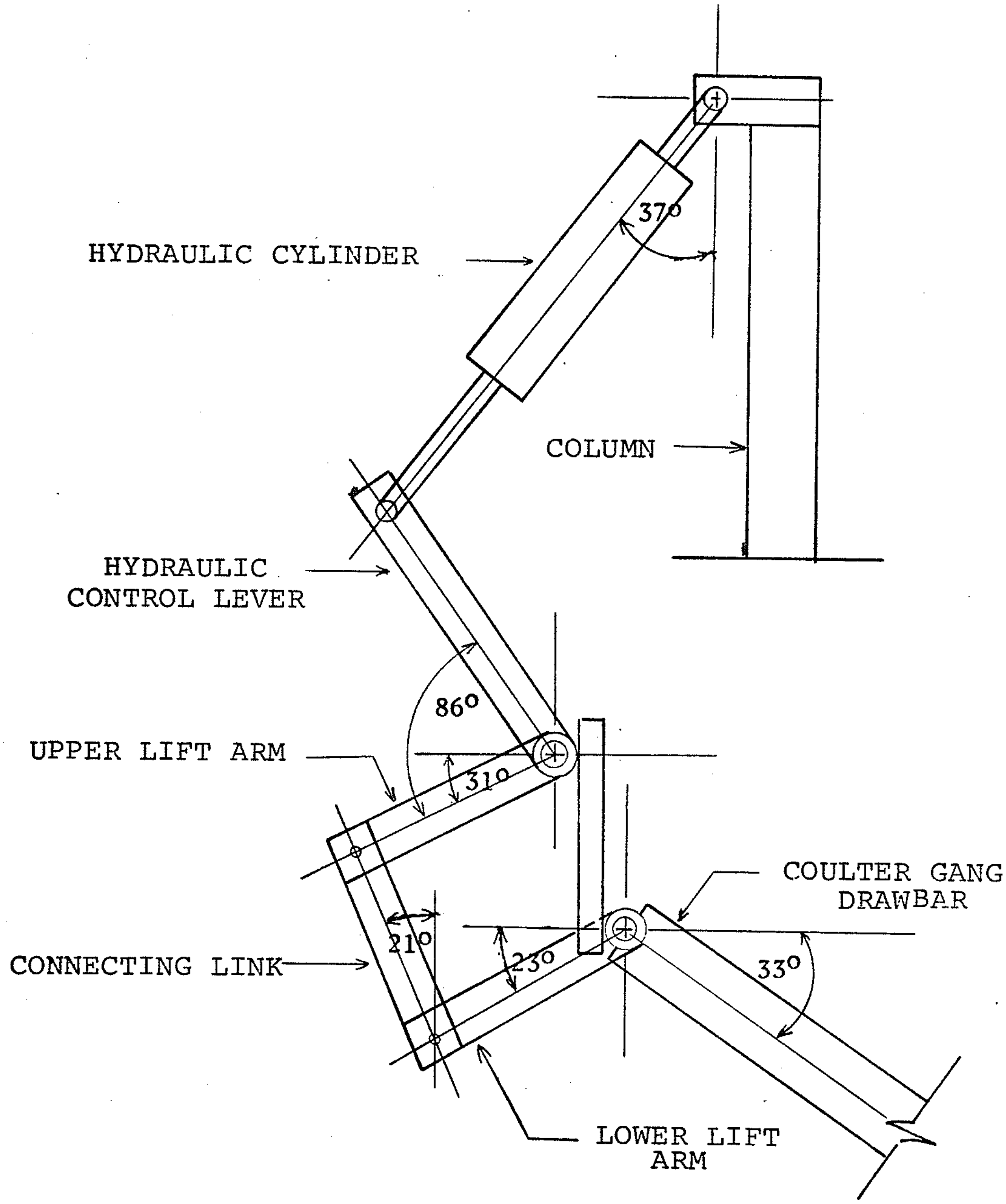

Figure 3.3. Machine elements in maximum load position.

The AISI specification for the materials used and the properties of the pipes used are shown in Table 3.2 and Table 3.3, respectively.

TABLE 3.1. Allowable Design Stresses (6)

Materials (AISI No.)	S_u (ksi)	S_s (ksi)
C 1020 steel, as rolled	22	16
C 1035 steel, as rolled	28	21
C 1045 steel, as rolled	32	24

S_u = allowable design stress in tension
S_s = allowable design stress in shear

TABLE 3.2. AISI Specifications for Materials Used

Name	Size	AISI No.
Spacer	1" XS pipe	C 1020
Coulter gang drawbar	3/8" x 3"	C 1035
Lower lift arm	5/16" x 2"	C 1045
Connecting link	5/16" x 2"	C 1045
Pin	1/2" diameter	C 1020
Upper lift arm	5/16" x 2"	C 1045
Upper shaft	1-1/2" XS pipe	C 1020
Hydraulic control lever	3-5/16" x 2"	C 1045
Column	2-3/8" x 3"	C 1035
Bracing	3/4" x 3/4"	C 1035

TABLE 3.3. Properties of XS Pipes Used

Nominal diam. in.	Outside diam. in.	Inside diam. in.	Thickness in.	I in.^{4}	A in.^{2}	r in.
1	1.315	0.957	0.179	0.106	0.639	0.41
1-1/2	1.900	1.500	0.200	0.391	1.068	0.61
2	2.375	1.939	0.218	0.868	1.477	0.77

3.7.1. Stress analysis of the coulter shaft assembly It was assumed that the coulter shaft spacers carry the vertical loads. The coulter gang bolt was assumed to carry no load. The loading on the coulter gang was assumed as illustrated in Figure 3.4.

The coulter shaft assembly was assumed to act as a continuous beam. The resisting force on each coulter gang drawbar was R_D = 600 lb and acted as shown in Figure 3.4.

$$\text{Maximum moment} = 300 \times 9 - 600 \times 7 + 300 \times 3$$

$$= 600 \text{ in.-lb}$$

$$s_{max} = \frac{Mc}{I}$$

$$= \frac{600}{0.106*} \times \frac{1.315*}{2}$$

$$= 3721.70 \text{ psi}$$

$$= 3.72 \text{ ksi}$$

$$Q = \frac{2}{3}(r_o^3 - r_i^3)$$

$$= \frac{2}{3}\left[\left(\frac{1.315}{2}\right)^3 - \left(\frac{0.957}{2}\right)^3\right]$$

$$= 0.116 \text{ in.}^3$$

$$S_s = \frac{VQ}{Ib}$$

*Value taken from Table 3.3.

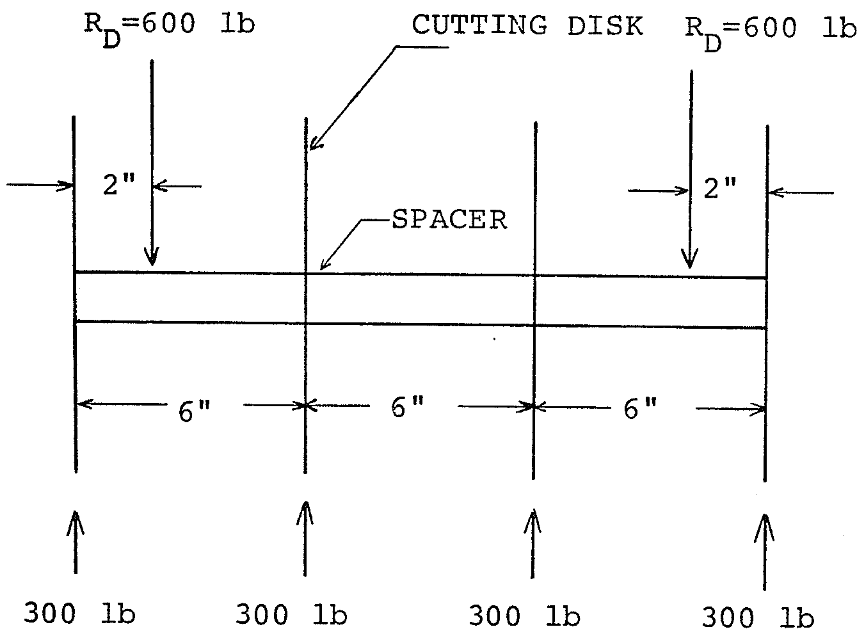

Figure 3.4. Free body diagram of the coulter shaft assembly

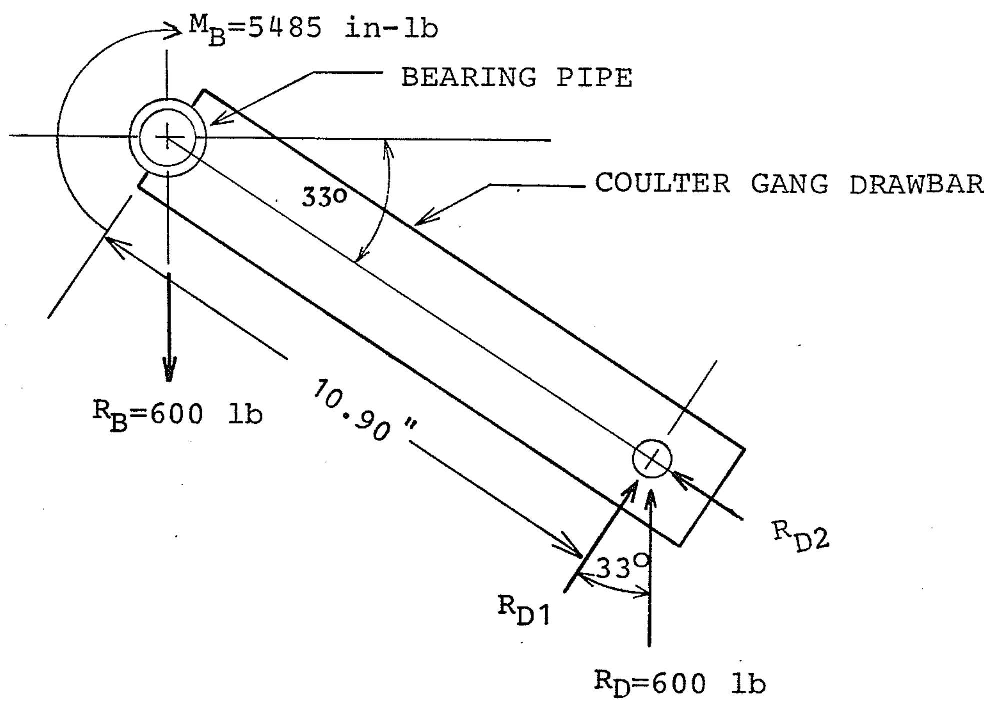

Figure 3.5. Free body diagram of the coulter gang drawbar.

$$= \frac{300 \times 0.116}{0.106^* \times (0.179^* \times 2)}$$

$$= 917 \text{ psi}$$

$$= 0.92 \text{ ksi}$$

According to Table 3.1, the coulter shafts were adequate.

3.7.2. Stress analysis for the coulter gang drawbars The free body diagram for a single coulter gang drawbar is shown in Figure 3.5.

Maximum force applied on the drawbar, $R_D = 600$ lb

$$R_{D1} = 600 \cos 33$$

$$= 503.20 \text{ lb}$$

$$R_{D2} = 600 \sin 33$$

$$= 326.78 \text{ lb}$$

Maximum moment, $M_B = 503.20 \times 10.90$

$$= 5485 \text{ in.-lb}$$

$$I = \frac{1}{12} bd^3$$

$$= \frac{1}{12} \times \frac{3}{8} \times 3^3$$

$$= 0.84 \text{ in.}^4$$

$$r = (I/A)^{1/2}$$

$$= (0.84/(\frac{3}{8} \times 3))^{1/2}$$

$$= 0.864 \text{ in.}$$

Slenderness ratio; $\frac{l}{r} = \frac{10.90}{0.864}$

$= 12.62$ (Column action can be neglected)

*Value taken from Table 3.3.

$$S_{max} = \frac{Mc}{I} + \frac{P}{A}$$
$$= \frac{5485 \times 1.5}{0.84} + \frac{326.78}{3/8 \times 3}$$
$$= 10084.93 \text{ psi}$$
$$= 10.08 \text{ ksi}$$

Comparing with Table 3.1 the coulter gang drawbars were adequate.

3.7.3. Stress analysis for the lower lift arms

The lower lift arms were made of two pieces of 5/16 in. x 2 in. C 1045 steel. Each lower lift arm transferred the loads to each coulter gang drawbar. A free body diagram of the lower lift arm is shown in Figure 3.6.

Maximum moment for each piece of the lower lift arm, M_B = 5485 in.-lb, $\Sigma M_B = 0$;

$$F_{c1} \times 6.875 = 5485$$
$$F_{c1} = 797.82 \text{ lb}$$
$$F_c = \frac{F_{c1}}{\cos 2}$$
$$= 798.29 \text{ lb}$$
$$F_{c2} = F_c \sin 2$$
$$= 27.86 \text{ lb}$$
$$I = \frac{1}{12} bd^3$$
$$= \frac{1}{12} \times \frac{5}{16} \times 2^3$$
$$= 0.21 \text{ in.}^4$$
$$S_{max} = \frac{Mc}{I} + \frac{P}{A}$$
$$= \frac{5485 \times 1}{0.21} + \frac{27.86}{5/16 \times 2}$$
$$= 26163.15 \text{ psi}$$
$$= 26.16 \text{ ksi}$$

The lower lift arms were considered adequate.

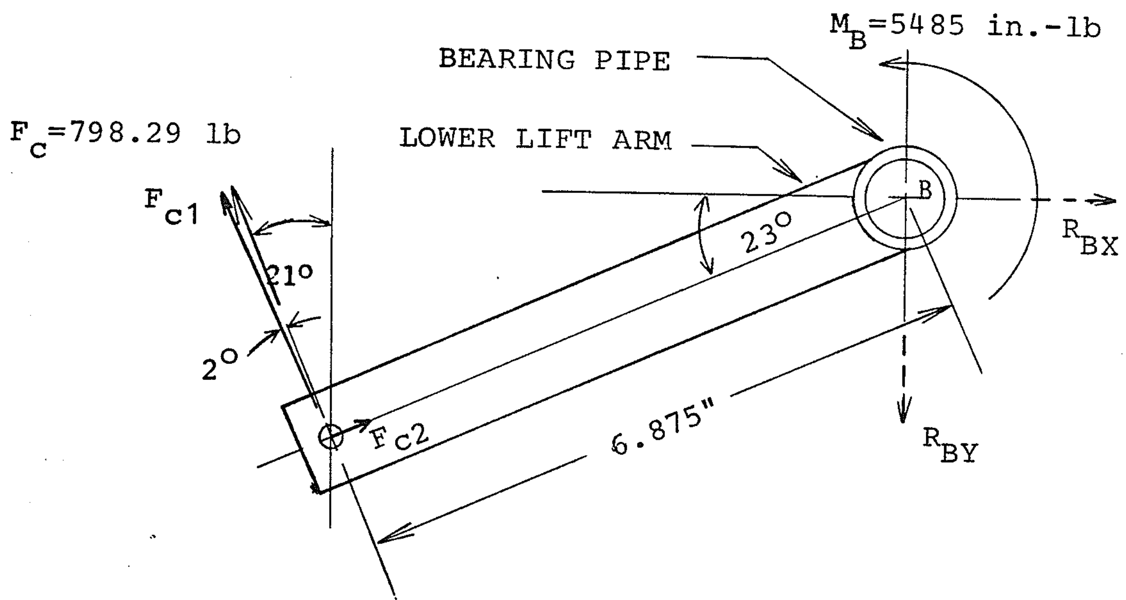

Figure 3.6 Free body diagram of the lower lift arm

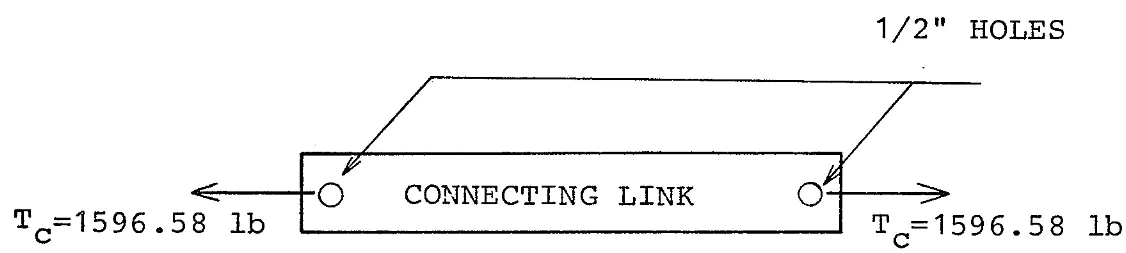

Figure 3.7 Free body diagram of the connecting link.

3.7.4. <u>Stress analysis for the connecting links</u>

The free body diagram of the connecting link is shown in Figure 3.7. The total axial force applied to the connecting link was

$$T_c = 2F_c$$

$$= 2 \times 798.29$$

$$= 1596.58 \text{ lb}$$

$$\text{Maximum tensile stress} = \frac{1596.58 \times 16}{(2-0.5) \times 5}$$

$$= 3406.04 \text{ psi}$$

$$= 3.41 \text{ ksi}$$

$$\text{Bearing stress} = \frac{1596.58 \times 16}{0.5 \times 5}$$

$$= 10218.11 \text{ psi}$$

$$= 10.22 \text{ ksi}$$

According to Table 3.1, the connecting links were adequate.

3.7.5. <u>Stress analysis for the pins</u> The pins were in a double shear condition. Total force acting on the pin, T_c = 1596.58 lb.

$$\text{Diameter of pin} = 1/2 \text{ in.}$$

$$\text{Shear area} = \frac{\pi}{4} \left(\frac{1}{2}\right)^2$$

$$= 0.196 \text{ in.}^2$$

$$S_s = \frac{V}{A}$$

$$= \frac{1596.58}{2 \times 0.196}$$

$$= 4072.91 \text{ psi}$$

$$= 4.07 \text{ ksi}$$

According to Table 3.1, the pins were adequate.

3.7.6. <u>Stress analysis for the upper lift arms</u>

Each upper lift arm was considered separately. A free body diagram of the upper lift arm is shown in Figure 3.8.

The tensile force in the connecting link for a single upper lift arm is $F_c = 798.29$ lb. Then, the bending moment in each upper lift arm is,

$$M_L = F_c \cos 10 \times 6.35$$
$$= 798.29 \cos 10 \times 6.35$$
$$= 4992.13 \text{ in.-lb}$$
$$F_{c4} = F_c \sin 10$$
$$= 138.62 \text{ lb}$$
$$I = \frac{1}{12} bd^3$$
$$= \frac{1}{12} \times \frac{5}{16} \times 2^3$$
$$= 0.21 \text{ in.}^4$$
$$S_{max} = \frac{Mc}{I} + \frac{P}{A}$$
$$= \frac{4992.13 \times 1}{0.21} + \frac{138.62}{5/16 \times 2}$$
$$= 23993.84 \text{ psi}$$
$$= 23.99 \text{ ksi}$$

According to Table 3.1, the upper lift arms were adequate.

3.7.7. <u>Stress analysis for upper shaft</u> A free body diagram of the upper shaft is shown in Figure 3.9. The torque transferred by each double upper lift arm was,

$$T = 2M_L$$
$$= 2 \times 4992.13$$
$$= 9984.26 \text{ in.-lb}$$

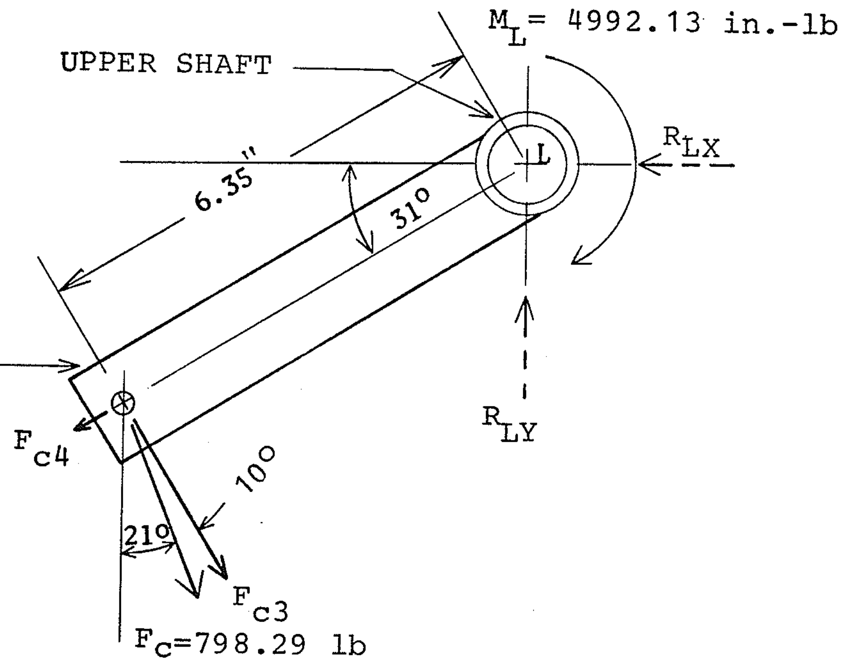

Figure 3.8 Free body diagram of the upper lift arm.

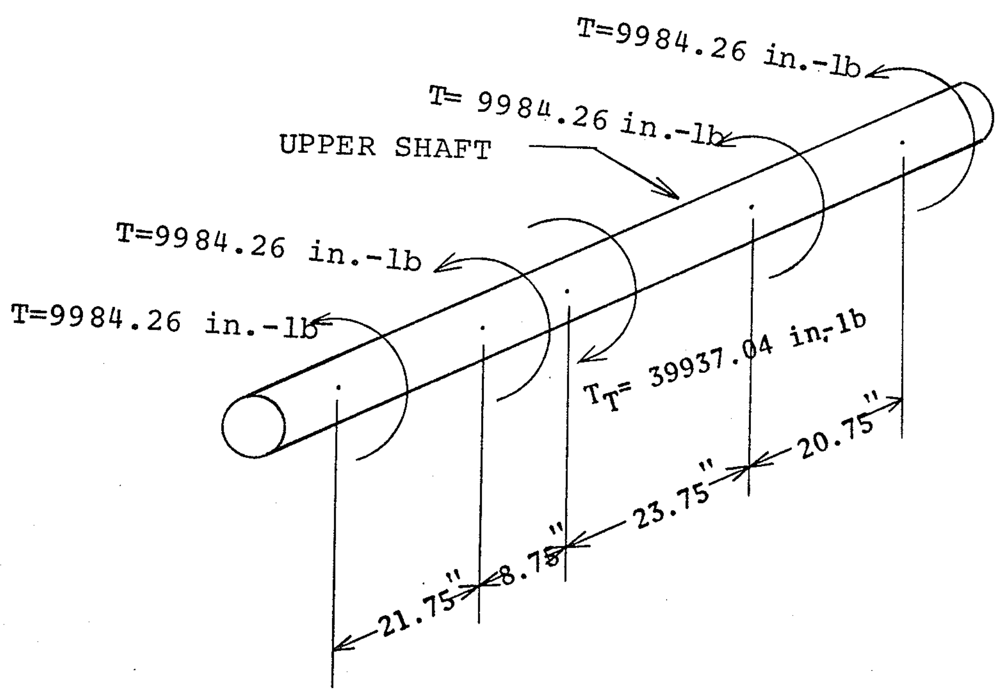

Figure 3.9 Free body diagram for torques on upper shaft.

Then the total torque required, $T_T = 4 \times 9984.26$

$= 39937.04$ in.-lb

Consider point H as a fixed support.

Maximum torque on the shaft = 2×9984.26

$= 19968.52$ in.-lb

$$S_s = \frac{Tc}{J}$$
$$= \frac{19968.52 \times 0.95^*}{2 \times 0.391^*}$$
$$= 24258.43 \text{ psi}$$
$$= 24.26 \text{ ksi}$$

The torques that would produce the maximum torsional deflection of the shaft were the torques at point C and D.

$$\theta = \frac{TL}{GJ}$$
$$= \frac{9984.26 \times 2 \times 12 \times 57.3}{11.5 \times 10^6 \times 2 \times 0.391^*}$$
$$= 1.53 \text{ deg/ft}$$

The maximum shear stress in the upper shaft exceeded the allowable limit. The factor of safety N was actually only 2.02. The deflection of the shaft exceeded the allowable deflection. In the field tests in severe conditions the shaft did not show excessive deflection nor did it fail. The actual loading was probably less than the assumed design loading. Impact loading might cause failure.

3.7.8. <u>Stress analysis of the hydraulic control lever</u> A free body diagram of the hydraulic control lever is shown in Figure 3.10.

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*Value taken from Table 3.3.

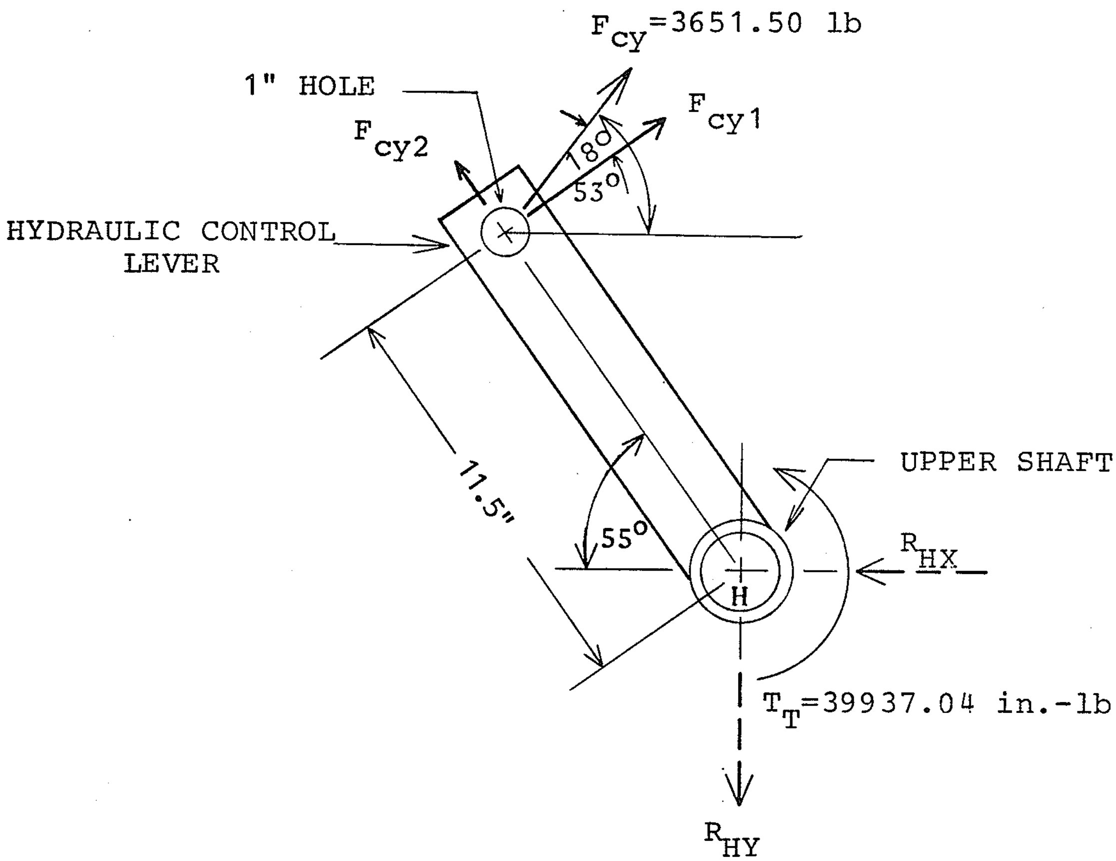


Figure 3.10 Free body diagram of the hydraulic control lever.

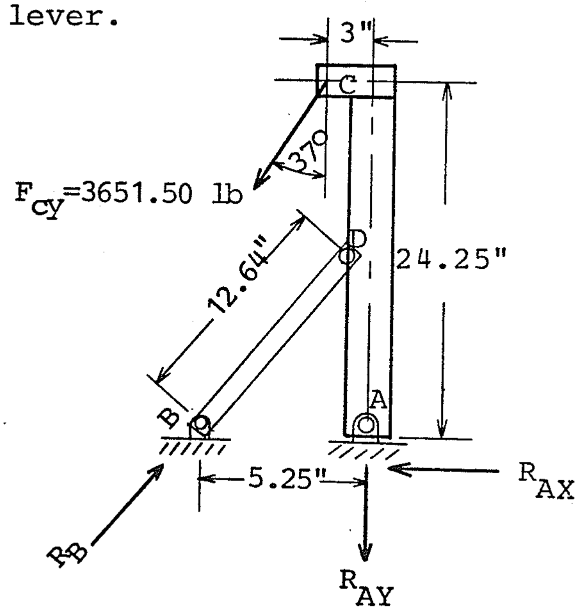


Figure 3.11 Free body diagram of the column.

The maximum moment for the hydraulic control lever, T_T = 39937.04 in.-lb

$$\Sigma M_H = 0 \;;$$

$$F_{cy} \cos 18 \times 11.5 = 39937.04$$

$$F_{cy} = 3651.50 \text{ lb}$$

$$F_{cy2} = F_{cy} \sin 18$$

$$= 1128.38 \text{ lb}$$

$$I = \frac{1}{12} bd^3$$

$$= \frac{1}{12} \times \frac{15}{16} \times 2^3$$

$$= 0.625 \text{ in.}^4$$

$$S_{max} = \frac{Mc}{I} + \frac{P}{A}$$

$$= \frac{39937.04 \times 1}{0.625} + \frac{1128.38}{15/16 \times 2}$$

$$= 64501.06 \text{ psi}$$

$$= 64.50 \text{ ksi}$$

$$\text{Bearing stress} = \frac{3651.50}{1 \times 15/16}$$

$$= 3894.93 \text{ psi}$$

$$= 3.89 \text{ ksi}$$

The bearing stress for the hydraulic control lever was below the allowable limit. The maximum bending stress was 64.5 ksi which exceeded the allowable stress (32 ksi). Thus, for the hydraulic control lever the design factor was reduced to 1.5. In the field tests in severe conditions, the hydraulic control lever did not fail nor did it show any signs of failing. In use either the press wheels or the front wheel carried some of the total weight so that the coulters never actually carried the total weight of the drill.

3.7.9. Stress analysis for the column and column bracing A simplified free body diagram for the column is shown in Figure 3.11. Pin connections have been assumed for the analysis. The forces at the pin connections were determined analytically. The forces were:

$$R_B = 10102.03 \text{ lb}$$

$$R_{AY} = 6274.71 \text{ lb}$$

$$R_{AX} = 1998.33 \text{ lb}$$

For the bracing

$$I = \frac{1}{12} bd^3$$

$$= \frac{1}{12} \times 0.75 \times 0.75^3$$

$$= 0.026 \text{ in.}^4$$

$$A = 0.75 \times 0.75$$

$$= 0.5625 \text{ in.}^2$$

$$r = (\frac{I}{A})^{1/2}$$

$$= (0.026/0.5625)^{1/2}$$

$$= 0.215 \text{ in.}$$

Slenderness ratio; $\frac{l}{r} = \frac{12.64}{0.215}$

$= 59$ (Column action must be considered)

$$\frac{F}{A} = S_e \left[1 - \frac{S_Y (l/r)^2}{4\pi^2 E}\right]$$

$$\frac{10102.03}{0.75 \times 0.75} = S_e \left[1 - \frac{55^{\dagger} (59)^2}{4\pi^2 \times 30 \times 10^6}\right]$$

$$S_e = 17958.46 \text{ psi}$$

$= 17.96$ ksi (allowable was 28 ksi)

---

† yield stress in tension.

For the column;

Consider portion CD, moment at D is

$$M_D = 2916.22 \times 3 + 2197.53 \times 12.75$$
$$= 36767.17 \text{ in.-lb}$$
$$I = \frac{1}{12} bd^3$$
$$= \frac{1}{12} \times \frac{3}{4} \times 3^3$$
$$= 1.6875 \text{ in.}^4$$
$$S_{max} = \frac{Mc}{I}$$
$$= \frac{36767.17 \times 1.5}{1.6875}$$
$$= 32681.93 \text{ psi}$$
$$= 32.68 \text{ ksi}$$

The bracing was considered adequate. The design factor for the column was reduced to 2.6.

3.8. Estimation of Cost of Materials and Quantity of Material The materials required and their costs were estimated for the drill modification. A list of materials used is shown in Drawing No. 1 and the estimated costs are shown in Table 3.4. The prices are for 1975 and no labor charge is included. A very rough estimate of the man-hours used for the construction and attachment of the modification would be 90 man-hours. Semi-skilled laborers with a knowledge of welding should be capable of assembling the attachment.

TABLE 3.4. Estimated Costs and Material Quantities

Total Amount	Materials	Estimated Cost ($)
7.40 ft	3 in. x 1-1/2 in. channel	9.41
8.50 ft	2 in. XS pipe	13.74
16.50 ft	1-1/2 in. XS pipe	19.42
5.84 ft	1 in. XS pipe	4.13
0.33 ft	2 in. diameter steel rod	3.00
7.33 ft	3/4 in. diameter steel rod	9.16
0.75 ft	1/2 in. diameter steel rod	0.75
11.42 ft	3/8 in. x 4 in. mild steel	14.80
13.35 ft	3/8 in. x 3 in. mild steel	12.89
17.70 ft	5/16 in. x 2 in. mild steel	9.53
5.33 ft	1/4 in. x 1 in. mild steel	1.15
16 only	17 in. diameter disks	304.16
8 only	bearings	131.70
20 only	1/2 in. x 1 in. cap screws	1.45
48 only	5/16 in. x 1-1/4 in. cap screws	3.05
8 only	3/8 in. set screws	0.80
8 only	3/4 in. nuts	1.57
36 only	1/2 in. nuts	1.88
56 only	3/8 in. nuts	1.25
8 only	3/4 in. locking washers	0.31
16 only	1/2 in. locking washers	0.52
16 only	1/2 in. flat washers	0.50
8 only	3/16 in. cotter pins	0.30
16 only	1/2 in. x 1-1/2 in. cap screws	1.44
	Paint	3.00
	Welding rods	5.00
1 only	Hydraulic cylinder	54.00
2 only	Hydraulic hoses	38.72
	Gas for cutting	10.00
	TOTAL: (Based on 1975 costs)	$657.62

CHAPTER IV

SOIL RESISTANCE, HORSEPOWER AND THEIR MEASUREMENT

The additional horsepower required to pull the modified press drill should be known in order to ensure adequate tractor power. Horsepower depends on the drawbar pull and the speed of operation. The drawbar pull depends on the soil resistance and the contact area between the implement and the soil. The contact area depends on the shape of the implement and the depth of penetration.

4.1. Soil Resistance Measurements

Soil resistance may be determined by measuring the penetration resistance of the soil. The penetrating element may be a circular, rectangular, flat or cone-shaped tip. In agricultural soil studies a cone penetrometer is frequently used and the best known type of soil penetrometer is the Cornell Soil Penetrometer.

The Cornell Soil Penetrometer is a self-recording device which is quite accurate, requires little adjusting, is light in weight and is simple to build. The construction of the device is such that the recording pointer is positioned by the depth of penetration of the cone and the downward force required to overcome resistance to penetration. Thus, as the point is pushed into the ground, a curve of force versus penetration is drawn. The mechanism consists of two parts:

i) a depth measuring element, and

ii) a force measuring element.

To measure the depth of penetration, the chart board is supported on a foot which rests on top of the ground, while a pointer, attached to the probe, moves down a distance equal to the depth of penetration. The force measurement is based on the fact that deflection of a spring is directly proportional to the force applied (12). Several samples can be quickly taken and recorded on the same chart without confusing the individual traces (by using different colors of pens) (12). However, accuracy of the penetrometer depends on soil moisture content. Maximum accuracy will be obtained when soil moisture content is about 20 percent (13).

The American Society of Agricultural Engineers recommends for field use a 30-degree circular cone penetrometer driven through the soil at a rate of 72 inches per minute (10).

4.2. Horsepower Measurements

The horsepower required for the drill was determined using a recording hydraulic drawbar dynamometer. Power is the product of drawbar pull and operating speed. A recording hydraulic drawbar dynamometer was available to measure and record these two variables. It consists of a hydraulic cylinder which converts drawbar pull into pressure which is recorded on a pressure recorder. The recording chart moves at a speed proportional to the ground speed. Thus, the recording chart records both pressure (for drawbar pull) and distance travelled in a known time (for speed) at the same time. In the field tests the drawbar dynamometer cart was connected to the tractor and the drill was connected to the rear of the drawbar dynamometer cart.

Figure 4.1. The arrangement of the tractor, the drawbar dynamometer and the machine in the field tests.

Horsepower can be determined by the following equation:

$$hp = \frac{K_{cy} \; P \; K_{ch} \; d}{K \; t} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4.1)$$

where:

hp = horsepower

K_{cy} = hydraulic cylinder constant (lb/psi), determined by static calibration

P = pressure reading from pressure recorder (psi)

K_{ch} = chart constant (ft/in.) determined by calibration before and after testing

d = chart distance (in.)

K = constant, 550 ft-lb/(sec/hp)

t = time (sec)

The additional horsepower required for the coulters was determined by subtracting the horsepower required for the double disk furrow openers from the total horsepower required when both the cutting coulters and the double disk furrow openers were used.

$$hp_a = hp_t - hp_f \quad \ldots\ldots\ldots\ldots\ldots \quad (4.2)$$

where:

hp_a = additional horsepower required for the coulters

hp_t = total horsepower (coulters and furrow openers)

hp_f = horsepower required for the double disks furrow openers

The uncertainty of the horsepower calculations was determined as follows:

$$W_{hp} = [(AW_{cy})^2 + (BW_p)^2 + (CW_{ch})^2 + (DW_d)^2 + (EW_t)^2]^{1/2} \quad \ldots\ldots\ldots\ldots\ldots \quad (4.3)$$

where:

W_{hp} = uncertainty of horsepower, hp

W_{cy} = uncertainty of hydraulic cylinder constant, lb/psi

W_p = uncertainty of pressure, psi

W_{ch} = uncertainty of chart constant, ft/in.

W_d = uncertainty of chart distance, in.

W_t = uncertainty of time, sec

$$A = \frac{\partial hp}{\partial K_{cy}} = \frac{PK_{ch}d}{K\ t} \text{ (hp-psi/lb)}$$

$$B = \frac{\partial hp}{\partial P} = \frac{K_{cy}\ K_{ch}\ d}{K\ t} \text{ (hp/psi)}$$

$$C = \frac{\partial hp}{\partial K_{ch}} = \frac{K_{cy}\ P\ d}{K\ t} \text{ (hp-in./ft)}$$

$$D = \frac{\partial hp}{\partial d} = \frac{K_{cy}\ P\ K_{ch}}{K\ t} \text{ (hp/in.)}$$

$$E = \frac{\partial hp}{\partial t} = -\frac{K_{cy}\ P\ K_{ch}\ d}{K\ t^2} \text{ (hp/sec)}$$

The uncertainty of the additional horsepower can be determined as follows:

$$W_a = (W_t^2 + W_f^2)^{1/2} \quad \text{.........................} \quad (4.4)$$

where:

W_a = uncertainty of additional horsepower required for the coulters

W_t = uncertainty of total horsepower (coulters and furrow openers)

W_f = uncertainty of horsepower required for double disk furrow openers.

CHAPTER V

TESTING PROCEDURE

5.1. General Approach

The experiments to determine the additional horsepower required for the cutting disks were repeated three times in two different soil conditions. The soil conditions were very fine sandy loam at Carman and Red River clay at Glenlea. Two tests were done at Glenlea with and without extra weight on the drill. The purpose of adding the extra weight on the drill was to achieve penetration of both cutting disks and double disks furrow openers.

In the field tests the drill was loaded with fertilizer to simulate one half of the full load capacity of the drill. The depth of penetration was measured by mounting an indicator on the drill. The scale reading was determined by a static calibration (Figure 5.1).

5.2. Measurement of Soil Resistance

Soil resistance was measured by a Cornell soil penetrometer. Measurements were taken randomly at both locations to give a representative soil resistance for each field. The soil penetration tests were done on the same day as the horsepower measurements, except for the first test at Glenlea, where the soil penetration tests were done one day later. Average soil resistances at depths of 0.5, 1.0, 1.5, 2.0, 2.5 and 3.0 inches were evaluated. The averages of

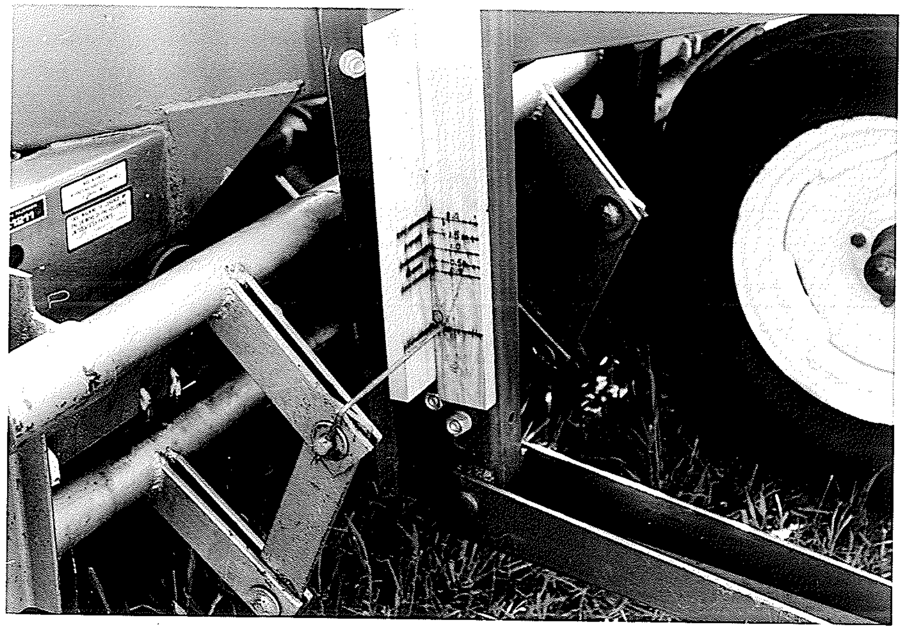


Figure 5.1. Penetration depth indicator.

soil resistance were expressed in terms of Cone Index (C.I., psi).

Prior to the field measurements, the penetrometer was calibrated by applying a force to the cone bearing on a platform scale. The relationship between the force on the cone and the deflection of the pen was determined using a linear regression analysis. The relation was:

$$F = 35.73\ X + 7.90 \quad \dots\dots\dots\dots\dots\dots\dots \quad (5.1)$$

where:

F = force required to deflect the pen (lb)

X = deflection of the pen due to force F (in.)

Cone areas of 0.2 and 0.5 in.^{2} were used. The soil resistance can be expressed as:

Soil resistance (C.I., psi) = (35.73 X + 790)/0.2 ...... (5.2)

When the 0.2 $in.^2$ cone was used or

Soil resistance (C.I., psi) = (35.73 X + 7.90)/0.5 ..... (5.3)

when the 0.5 $in.^2$ cone was used.

5.3. Horsepower Measurement

The horsepower was measured using a hydraulic recording drawbar dynamometer. The horsepower required for the cutting disks was determined by subtracting the horsepower to operate with the double disks from the horsepower required to operate both the double disks and the cutting disks. The penetration depth of the cutting disks was varied from zero to maximum depth to determine a relationship between the horsepower required and the depth of penetration.

The dynamometer was calibrated before testing. The hydraulic cylinder constant K_{cy} was 4.65 lb/psi. In each test a chart calibration was done with a field distance of 200 feet. The chart calibration was the average of four runs before the horsepower testing and four runs after the horsepower testing.

CHAPTER VI

RESULTS AND DISCUSSION

6.1. Machine Performance

The machine was constructed in the spring of 1975. The added parts are shown in blue in Figure 6.1. As soon as the modification were finished, the drill was used for zero-tillage planting. The performance of the machine was observed on the Glenlea research farm.

The coulters were able to cut through heavy trash and open a track for the furrow openers to place the seed into the soil. The machine performed adequately the functions which were required for a zero-tillage planting machine. Very little surface soil disturbance was observed. For sharp turns of the machine the coulters and furrow openers had to be lifted from the soil to prevent damage.

The depth of penetration of each coulter was the same (from a level surface) and fixed during operation; if the machine was operated on a rough soil surface the coulters would not penetrate uniformly into the soil. In an extremely rough surfaced field, some coulters may not be able to touch the soil surface even if the coulters were in the maximum depth position. However, this problem seems to be not too serious a problem for zero-tillage because zero-tilled soil surfaces are relatively smooth and even.

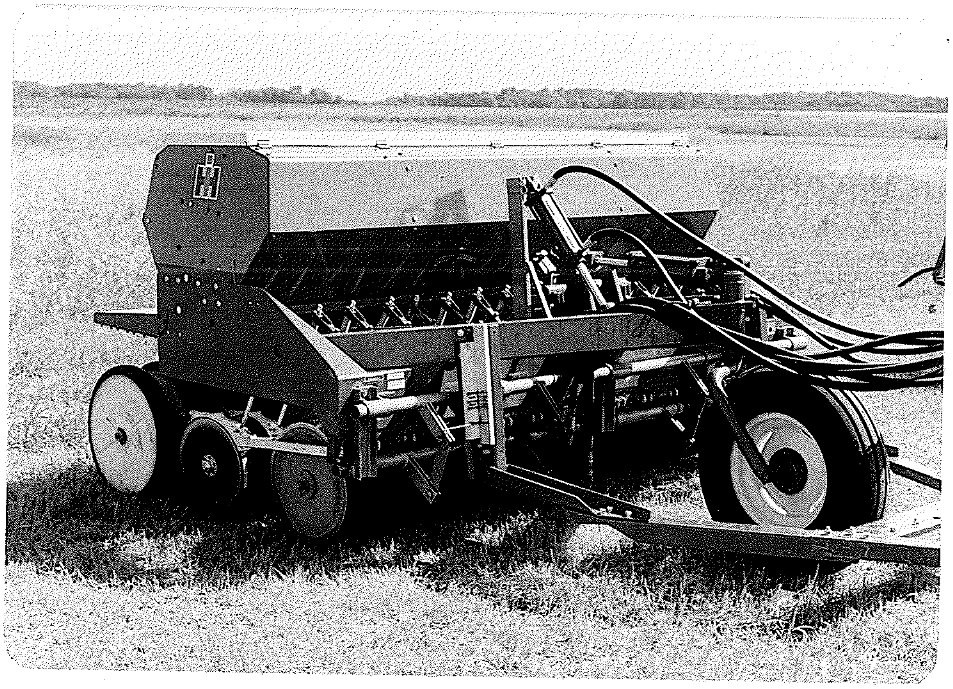


Figure 6.1 Complete constructed machine.

6.2. Additional Horsepower Required

The soil at Carman, Manitoba was very fine sandy loam with the soil resistance at 0.5, 1.0, 1.5, 2.0, 2.5 and 3.0 inches deep, as shown in Table 6.1. The tests were done at average speeds of 3.58 mph and 5.14 mph. During the tests the cutting coulters penetrated easily into the soil. The results of the horsepower testing are shown in Table A.4.1, Table A.4.2 and Figure 6.2. The relationship between the additional horsepower required for the cutting disks and the depth of penetration is shown in Figure 6.2. At the operating speeds of 3.58 mph and 5.14 mph, the relationship between the additional horsepower required for the cutting disks and depth of penetration were hp = 1.36 x Depth and hp = 1.63 x Depth, respectively. The standard errors for the regression equations were ±0.25 and ±0.31 horsepower per inch, respectively.

For the Red River clay soil at Glenlea, the first test failed to achieve good penetration because the soil was too hard (Table 6.1). The coulters were lowered but the hard soil prevented penetration. The whole frame of the drill was lifted so that the front wheel, the furrow openers and the press wheels were almost lifted from the ground.

The depth indicator attached to the drill frame did not indicate the actual depth of penetration. Nevertheless testing was completed as shown in Table A.4.3 and the relationship between additional horsepower and indicated depth was hp = 1.84 x Depth for an average speed of 3.32 mph (Figure 6.3). The standard error for the regression equation was ±0.22 horsepower per inch.

To achieve adequate penetration the drill was ballasted with 960 pounds of additional weight and the tests were repeated. The soil resistance is shown in Table 6.1 and the results are shown in Table A.4.4. Good penetration of the coulters was obtained and the relationship between the additional horsepower and depth of penetration was hp = 2.04 x Depth for an average speed of 3.67 mph (Figure 6.4). The standard error for the regression equation was ±0.20 horsepower per inch.

From the results above it was clear that the additional horsepower required for the cutting disks varied as the depth of penetration, the soil type and the ground speed. This information could be very useful for tractor selection for zero-tillage use.

TABLE 6.1. Soil Resistance at Carman and Glenlea

	Average Soil Resistance (psi)*					
	Depth from Surface (in.)					
Location	0.5	1.0	1.5	2.0	2.5	3.0
Carman	142	226	273	304	334	363
Glenlea #1	366	550	656	736	788	827
Glenlea #2	334	559	649	674	697	709

*Measured by Cornell Soil Penetrometer (Cone Index, psi).

Figure 6.2 Relationship between depth of penetration and additional horsepower required for coulters at Carman (very fine sandy loam soil).

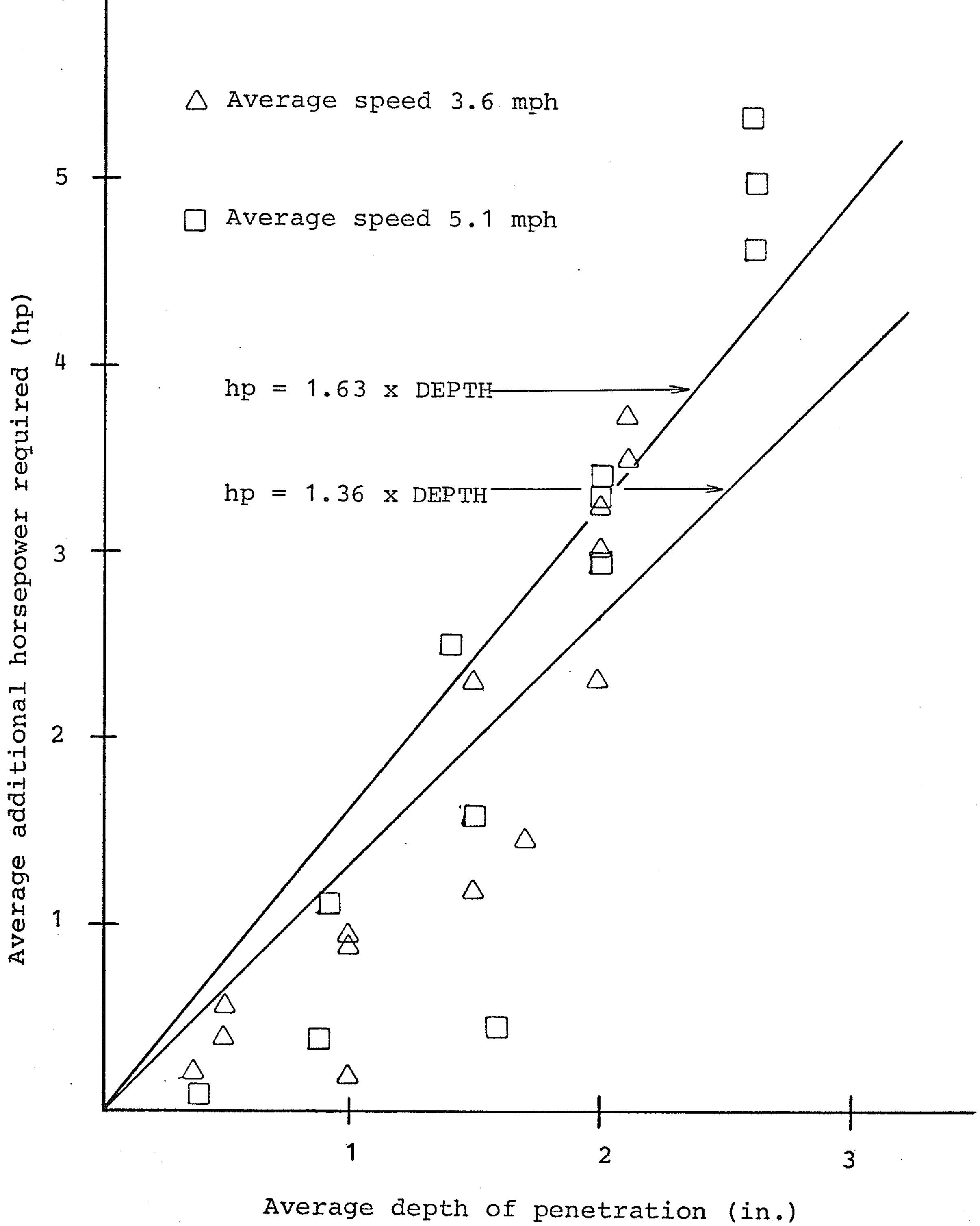


Figure 6.3 Relationship between indicated depth of penetration and additional horsepower required for coulters at Glenlea (Red River clay soil) without additional weight on the drill.

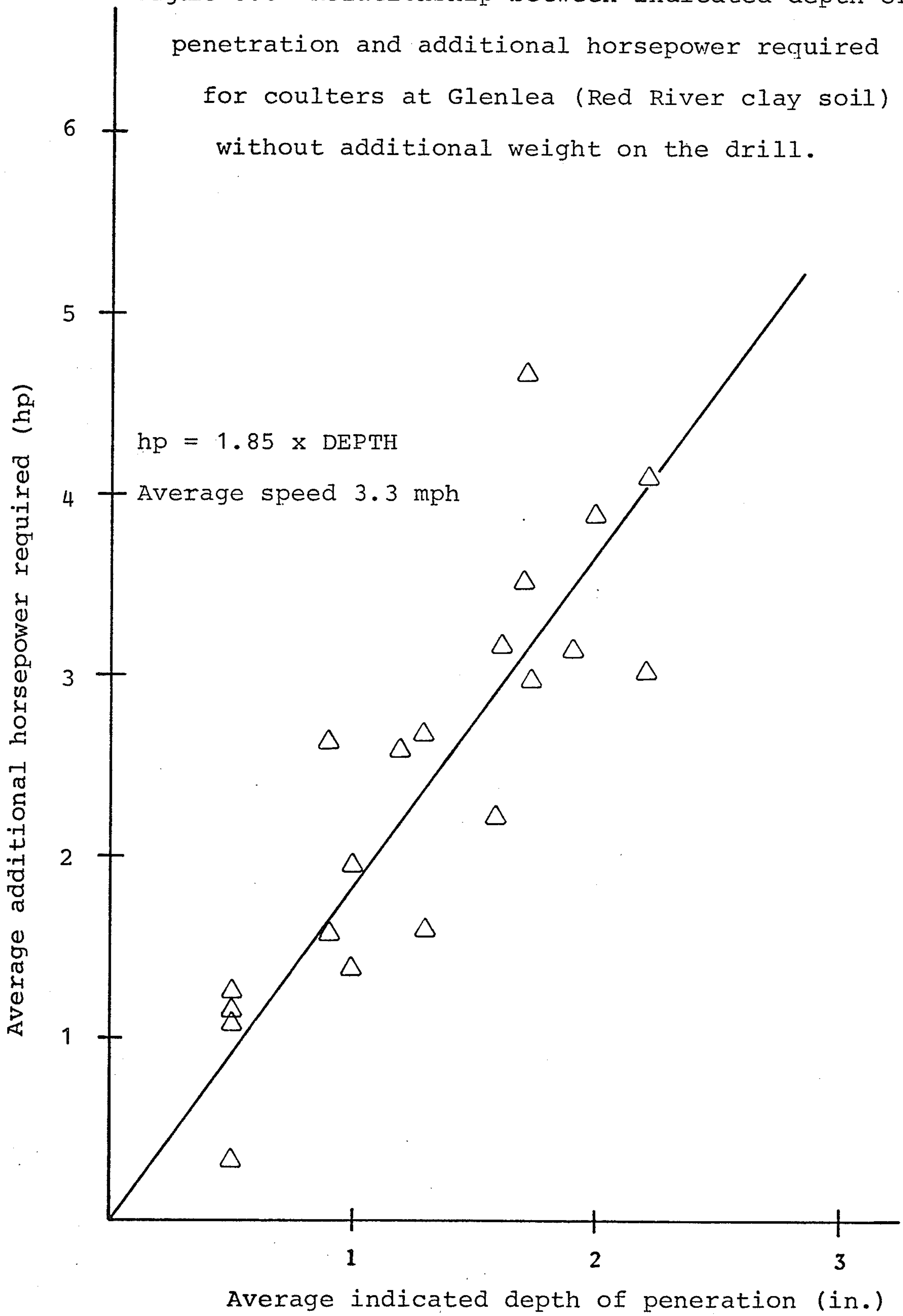
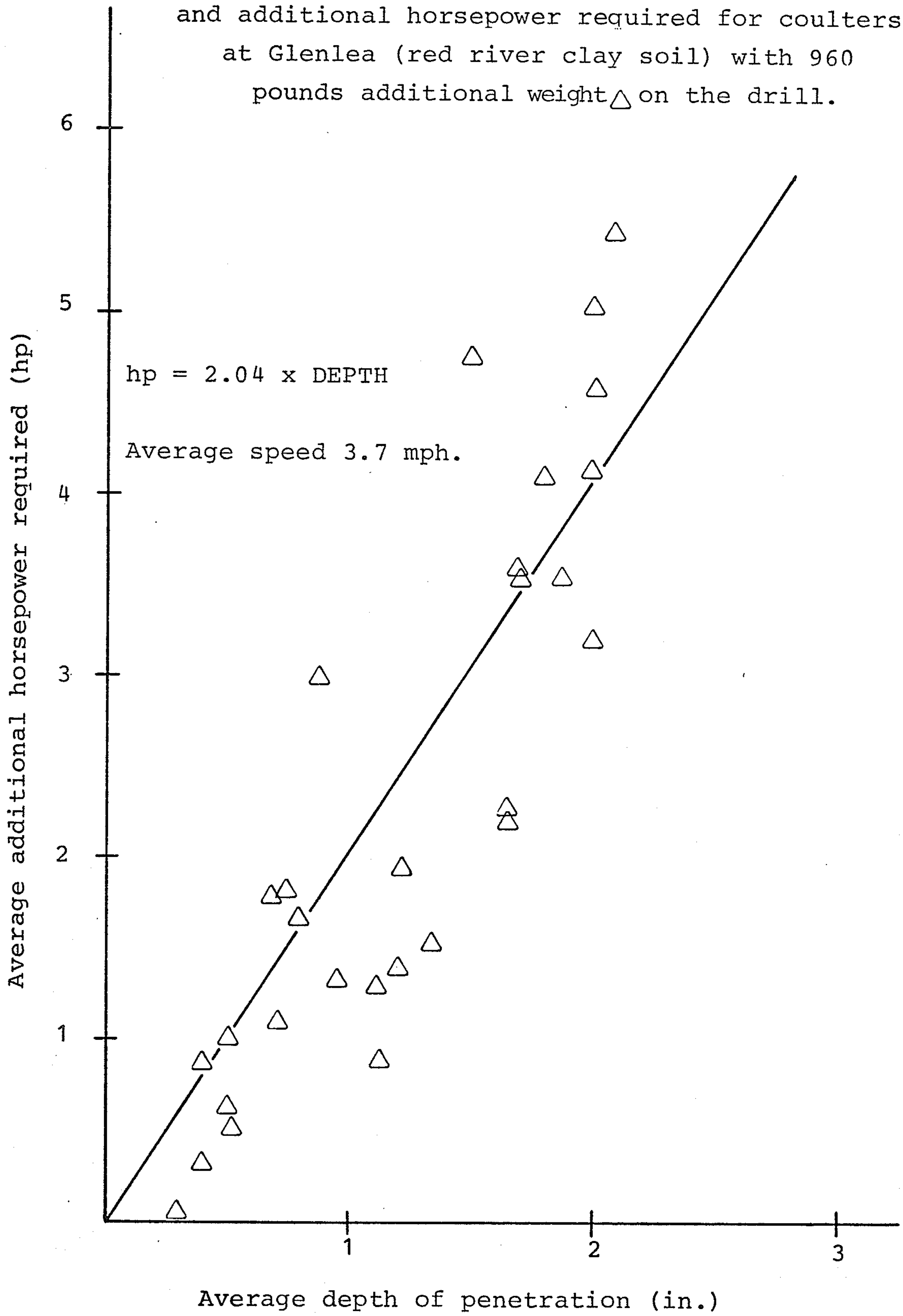


Figure 6.4 Relationship between depth of penetration and additional horsepower required for coulters at Glenlea (red river clay soil) with 960 pounds additional weight △ on the drill.



CHAPTER VII

CONCLUSIONS

The zero-tillage planting machine designed was found to be practical for zero-tillage operations. Machines similar to this machine could be made available for zero-tillage operations either by commercial manufacturers or by enterprising farmers. The adaptation of the basic machine was simple and was completed in about three weeks by two men. The cost of the adaptation was comparatively low ($660.00 not including labor cost). All of the purchased components were readily available at local suppliers.

The additional horsepower requirements for the cutting disks were not high even for the very hard soil (below 10 hp). Any available tractor that would normally be used with an unmodified press drill could be used for zero-tillage operations.

Another advantage of the zero-tillage planting machine was that it could be used for both conventional and zero-tillage purposes. For a conventional tillage purpose, the cutting disks would be lifted into transport position permitting the machine to work as an ordinary drill. However, the machine may not be suitable for all types of soil. It functioned efficiently with light to medium textured soil but had difficulty in penetrating heavy soil.

CHAPTER VIII

RECOMMENDATIONS FOR FUTURE STUDY

The main purpose of this study was to design a zero-tillage planting attachment for a standard press drill. Further develoments and studies are needed for zero-tillage cultural methods.

Recommendations for future study are as follows:

1. Automatic depth control for each cutting disk would be essential for fields with very rough surface conditions.

2. Limitations of the machine on different types of soil should be studied more thoroughly.

3. Comparison of costs and benefits for conventional methods versus zero-tillage methods should be done.

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APPENDIX A

TABLE A.1 Calibration constants

Location	K_{cy} (lb drawbar pull/ psi cylinder pressure)	K_{ch} (ft ground travel/ in. chart travel)
Carman	4.65	16.61
Glenlea #1	4.65	16.67
Glenlea #2	4.65	16.63

TABLE A.2.1 Raw data for horsepower measurements at Carman for 3.58 mph average speed of operation

Run No.	Average Depth of Penetration of Coulter (in.)	Time (sec)	Chart Distance (in.)	Average Pressure (psi)
1	-	14.95	5.03	65
2	0.5	23.79	8.03	75
3	1.0	25.78	8.41	85
4	1.5	21.30	6.63	120
5	2.0	21.39	6.69	120
6	2.0	24.15	7.13	150
7	-	19.34	6.13	70
8	0.5	19.72	6.26	75
9	1.0	20.24	6.50	85
10	1.5	20.30	6.31	95
11	2.0	21.39	6.47	140
12	2.1	20.11	6.21	155
13	-	14.57	4.66	60
14	2.1	15.59	4.71	150
15	1.7	15.11	4.75	100
16	1.0	16.72	5.31	70
17	0.4	20.05	6.50	70
18	-	17.50	5.50	65

TABLE A.2.2 Raw data for horsepower measurements at Carman for 5.14 mph average speed of operation

Run No.	Average Depth of Penetration of Coulter (in.)	Time (sec)	Distance (in.)	Average Pressure (psi)
1	-	14.70	6.91	90
2	0.38	15.60	7.30	80
3	0.88	16.23	7.63	95
4	1.38	17.72	8.03	120
5	2.00	18.41	8.22	135
6	2.56	19.10	7.93	180
7	2.56	17.20	7.08	170
8	2.08	16.15	7.16	130
9	1.56	18.06	8.54	85
10	0.78	14.61	7.00	75
11	-	13.90	6.72	70
12	0.88	13.59	6.32	85
13	1.48	12.00	5.47	105
14	2.08	10.71	4.84	135
15	2.58	14.49	6.16	170
16	-	9.53	4.34	75

TABLE A.2.3 Raw data for horsepower measurements at Glenlea without additional weight on the drill

(average speed = 3.32 mph)

Run No.	Average Indicated Depth of Penetration of coulter (in.)	Time (sec)	Chart Distance (in.)	Average Pressure (psi)
1	-	22.75	6.79	80
2	-	16.62	4.97	65
3	0.5	17.16	5.00	95
4	0.9	20.01	5.50	160
5	1.3	19.29	5.63	125
6	1.6	16.68	5.00	160
7	1.7	21.72	6.25	175
8	1.7	19.28	5.64	200
9	1.3	14.67	4.09	160
10	1.0	14.25	4.22	120
11	0.5	17.17	5.05	115
12	-	15.39	4.63	85
13	-	16.11	4.93	90
14	0.5	16.43	4.85	100
15	0.9	15.40	4.52	125
16	1.2	14.17	4.13	150
17	1.7	14.82	4.28	160
18	2.0	13.24	4.00	175
19	2.2	15.82	4.38	170
20	2.2	14.80	4.06	200
21	1.9	14.63	4.10	170
22	1.6	15.80	4.63	140
23	1.0	14.22	4.13	135
24	0.5	14.11	4.31	110
25	-	15.90	4.81	100

TABLE A.2.4. Raw data for horsepower measurements at Glenlea with 960 pounds additional weight on the drill

(average speed = 3.67 mph)

Run No.	Average Depth of Penetration of Coulter (in.)	Time (sec)	Chart Distance (in.)	Average Pressure (psi)
1	-	17.52	6.72	70
2	0.40	18.88	6.03	105
3	0.78	18.49	6.04	120
4	1.15	16.22	5.38	110
5	1.65	16.65	5.25	130
6	2.00	16.47	5.35	175
7	2.00	15.19	5.06	150
8	1.65	15.50	5.06	135
9	1.15	14.00	4.44	105
10	0.75	15.38	4.75	130
11	0.32	14.62	4.69	85
12	-	13.57	4.52	85
13	0.43	14.38	4.69	90
14	0.73	14.82	4.75	110
15	1.18	14.72	4.56	120
16	1.73	12.74	3.94	170
17	2.08	13.05	4.10	200
18	2.08	14.13	4.44	190
19	1.73	13.02	4.27	160
20	1.23	12.85	4.22	125
21	0.73	12.78	4.26	120
22	0.48	14.48	4.63	100
23	-	13.90	4.56	85
24	0.53	11.85	3.85	95
25	0.95	12.18	3.88	115
26	1.51	13.28	4.16	195
27	1.83	12.80	4.13	175
28	2.13	14.90	4.69	210
29	2.13	17.00	5.52	220
30	1.83	12.80	4.19	160
31	1.33	12.00	4.00	115
32	0.88	11.00	3.68	145
33	0.53	11.48	3.81	105
34	-	13.12	4.40	90

A.3 Sample Calculation of Horsepower

The horsepower equation was given by eq. 4.1

$$hp = \frac{K_{cy} \; P \; K_{ch} \; d}{K \; t}$$

From Table A.2.4 for run number five;

$t = 15.65$ sec

$d = 5.25$ in.

$P = 130$ psi

From Table A.1;

$K_{cy} = 4.65$ lb/psi

$K_{ch} = 16.63$ ft/in.

and K = Constant = 550 ft-lb/(sec/hp)

Thus;

$$hp = \frac{4.65 \times 130 \times 16.63 \times 5.25}{550 \times 15.65}$$

$hp = 6.13$ hp (tabulated in Table A.4.4)

The calculated horsepower are tabulated in Table A.4.1, Table A.4.2, Table A.4.3 and Table A.4.4. The additional horsepower required for the cutting disks is:

$$hp_a = hp_t - hp_f$$
$$= 6.13 - 3.82$$
$$= 2.31 \text{ hp (tabulated in Table A.4.4)}$$

TABLE A.4.1 Horsepower measurement at Carman for 3.58 mph average speed of operation

Run No.	Average Depth of Penetration of Coulter (in.)	Average Ground Speed (mph)	Total hp	Additional hp
1	-	3.81	3.07*	
2	0.5	3.83	3.51	0.57
3	1.0	3.70	3.90	0.96
4	1.5	3.53	5.25	2.31
5	2.0	3.54	5.25	2.31
6	2.0	3.34	6.21	3.27
7	-	3.59	3.11*	
8	0.5	3.59	3.34	0.40
9	1.0	3.63	3.83	0.89
10	1.5	3.52	4.14	1.20
11	2.0	3.42	5.94	3.00
12	2.1	3.44	6.62	3.68
13	-	3.62	2.69*	
14	2.1	3.42	6.37	3.43
15	1.7	3.56	4.41	1.47
16	1.0	3.60	3.12	0.18
17	0.4	3.67	3.18	0.24
18	-	3.56	2.87*	

*Average horsepower required for furrow openers = 2.94 hp

TABLE A.4.2 Horsepower measurements at Carman for 5.14 mph average speed of operation

Run No.	Average Depth of Penetration of Coulter (in.)	Average Ground Speed (mph)	Total hp	Additional hp
1	-	5.33	5.94*	
2	0.38	5.30	5.26	0.10
3	0.88	5.33	6.27	1.11
4	1.38	5.13	7.64	2.48
5	2.00	5.06	8.47	3.31
6	2.56	4.70	10.50	5.34
7	2.56	4.66	9.83	4.67
8	2.08	5.02	8.09	2.93
9	1.56	5.35	5.64	0.48
10	0.78	5.43	5.05	-0.11
11	-	5.48	4.75*	
12	0.88	5.26	5.55	0.39
13	1.48	5.16	6.72	1.56
14	2.08	5.12	8.57	3.41
15	2.58	4.81	10.15	4.99
16	-	5.15	4.79*	

*Average horsepower required for furrow openers = 5.16 hp

TABLE A.4.3 Horsepower measurement at Glenlea without additional weight on the drill

(average speed = 3.32 mph)

Run No.	Average Indicated Depth of Penetration of Coulter (in.)	Average Ground Speed (mph)	Total hp	Additional hp
1	-	3.39	3.37*	
2	-	3.40	2.74*	
3	0.5	3.31	3.90	0.33
4	0.9	3.12	6.20	2.63
5	1.3	3.32	5.14	1.57
6	1.6	3.41	6.76	3.19
7	1.7	3.27	7.10	3.53
8	1.7	3.32	8.25	4.68
9	1.3	3.17	6.29	2.72
10	1.0	3.37	5.01	1.44
11	0.5	3.34	4.77	1.20
12	-	3.42	3.60*	
13	-	3.48	3.88*	
14	0.5	3.36	4.16	0.59
15	0.9	3.34	5.17	1.60
16	1.2	3.31	6.16	2.59
17	1.7	3.28	6.51	2.94
18	2.0	3.43	7.45	3.88
19	2.2	3.15	6.63	3.06
20	2.2	3.12	7.73	4.16
21	1.9	3.19	6.71	3.14
22	1.6	3.33	6.78	2.21
23	1.0	3.30	5.53	1.96
24	0.5	3.47	4.74	1.17
25	-	3.44	4.26*	

*Average horsepower required for furrow openers = 3.57 hp

TABLE A.4.4 Horsepower measurements at Glenlea with 960 pounds additional weight on the drill

(average speed = 3.67 mph)

Run No.	Average Depth of Penetration of Coulter (in.)	Average Ground Speed (mph)	Total hp	Additional hp
1	-	3.70	3.12*	
2	0.40	3.62	4.72	0.90
3	0.78	3.70	5.51	1.69
4	1.15	3.76	5.13	1.31
5	1.65	3.80	6.13	2.31
6	2.00	3.68	7.99	4.17
7	2.00	3.78	7.03	3.21
8	1.65	3.70	6.02	2.20
9	1.15	3.60	4.68	0.86
10	0.75	3.50	5.64	1.82
11	0.32	3.64	3.83	0.01
12	-	3.78	3.98*	
13	0.43	3.70	4.13	0.31
14	0.73	3.64	4.96	1.14
15	1.18	3.51	5.23	1.41
16	1.73	3.51	7.39	3.57
17	2.08	3.56	8.83	5.01
18	2.08	3.56	8.39	4.57
19	1.73	3.72	7.38	3.56
20	1.23	3.72	5.77	1.95
21	0.73	3.78	5.62	1.80
22	0.48	3.62	4.50	0.68
23	-	3.72	3.92*	
24	0.53	3.68	4.34	0.52
25	0.95	3.61	5.15	1.33
26	1.51	3.55	8.59	4.77
27	1.83	3.66	7.94	4.12
28	2.13	3.57	9.29	5.47
29	2.13	3.68	10.04	6.22
30	1.83	3.71	7.36	3.54
31	1.33	3.78	5.39	1.57
32	0.88	3.79	6.82	3.00
33	0.53	3.76	4.90	1.08
34	-	3.80	4.24*	

*Average horsepower required for furrow openers = 3.82 hp

APPENDIX B

ERROR ANALYSIS

The uncertainty of the hydraulic cylinder constant (W_{cy}) was the standard error of the hydraulic cylinder constant (K_{cy}). The uncertainties of the chart constants (W_{ch}) were the standard errors of the chart constants (K_{ch}). The uncertainties of the pressure readings (W_p), chart distances (W_d) and times (W_t) were estimated from the scale readings. They are listed in Table B.1.

TABLE B.1. The uncertainties of the hydraulic cylinder constant (W_{cy}), the chart constants (W_{ch}), the pressure readings (W_p), the chart distances (W_d) and the times (W_t)

Location	W_{cy} (lb/psi)	W_{ch} (ft/in.)	W_p (psi)	W_d (in.)	W_t (sec)
Carman	±0.009	±0.051	±10	±0.03125	±0.1
Glenlea #1	±0.009	±0.020	±10	±0.03125	±0.1
Glenlea #2	±0.009	±0.018	±10	±0.03125	±0.1

B.2 Sample Calculation of the Uncertainty of the Horsepower

The equation for the uncertainty of the horsepower calculation was given by eq. 4.3.

$$W_{hp} = [(AW_{cy})^2 + (B\,W_p)^2 + (CW_{ch})^2 + (DW_d)^2 + (EW_t)^2]^{1/2}$$

$$A = PK_{ch}\ d/K\ t$$

$$B = K_{cy}\ K_{ch}\ d/K\ t$$

$$C = K_{cy}\ P\ d/K\ t$$

$$D = K_{cy}\ P\ K_{ch}/K\ t$$

$$E = -K_{cy}\ PK_{ch}\ d/K\ t^2$$

From Table A.2.4 for run number five (sample calculation of hp from appendix A.3) -

$$A = \frac{130 \times 16.63 \times 5.25}{550 \times 15.65}$$

$$= 1.319$$

$$B = \frac{4.65 \times 16.63 \times 5.25}{550 \times 15.65}$$

$$= 0.047$$

$$C = \frac{4.65 \times 130 \times 5.25}{550 \times 15.65}$$

$$= 0.369$$

$$D = \frac{4.65 \times 130 \times 16.63}{550 \times 15.65}$$

$$= 1.168$$

$$E = \frac{-4.65 \times 130 \times 16.63 \times 5.25}{550 \times (15.65)^2}$$

$$= -0.392$$

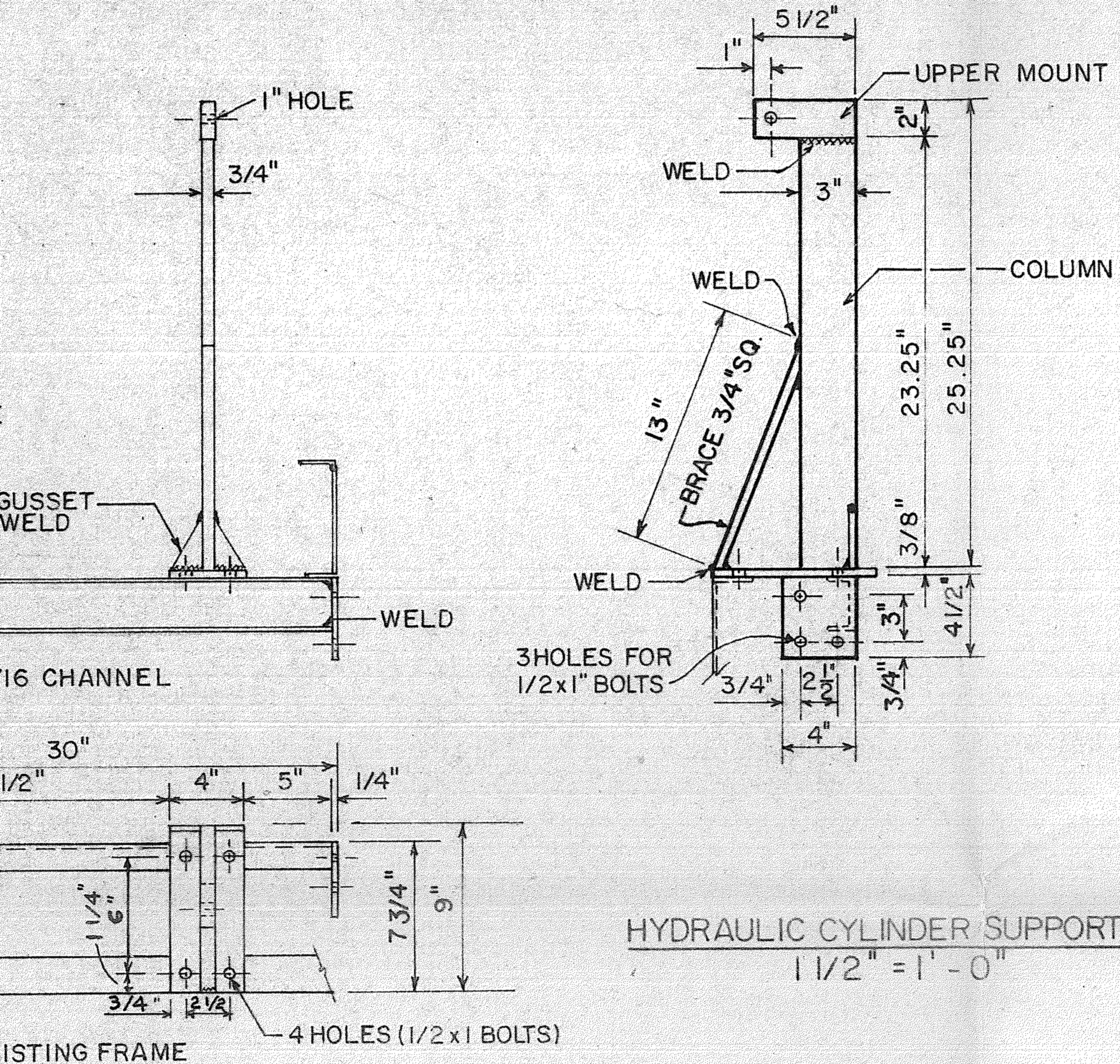
$$W_{hp} = [(1.319 \times 0.009)^2 + (0.047 \times 10)^2 + (0.369 \times 0.018)^2 + (1.168 \times 0.03125)^2 + (-0.392 \times 0.1)^2]^{1/2}$$

The uncertainty of the additional horsepower can be determined by eq. 4.4.

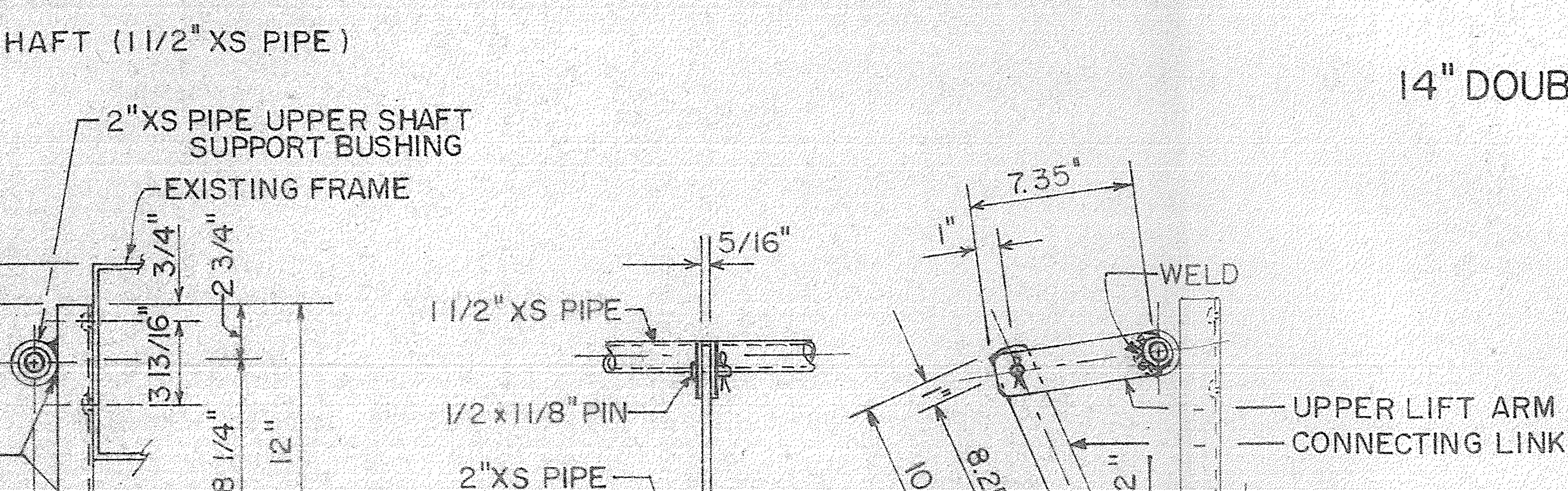
The uncertainties of both the total horsepower and the additional horsepower were relatively high and uniform over the range of horsepowers calculated. This was so because of the high uncertainty of the pressure readings (±10 psi). The average of the uncertainties of the total horsepower and the additional horsepower are shown in Table B.2.

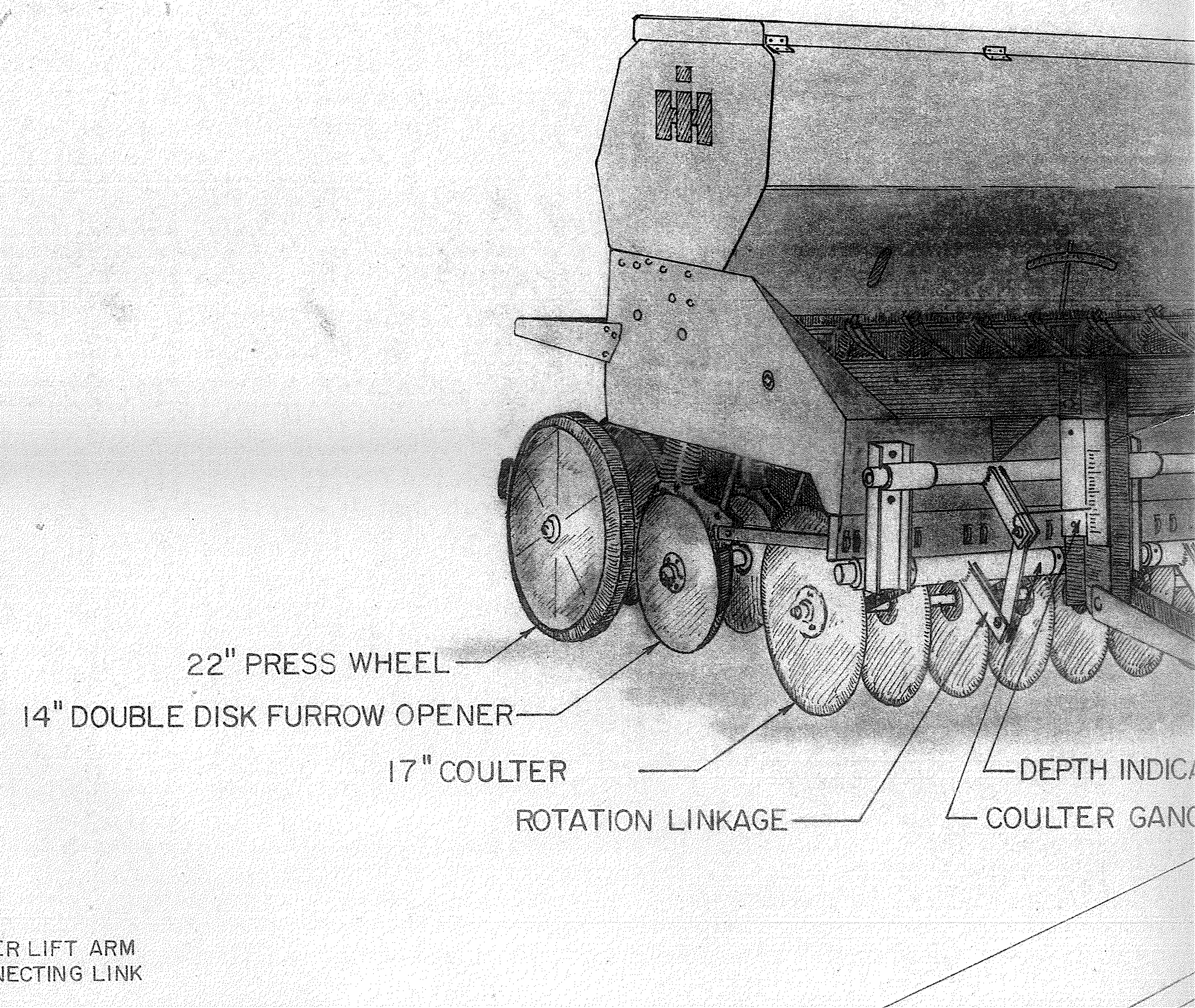
TABLE B.2 Average uncertainties of the total horsepower and the additional horsepower.

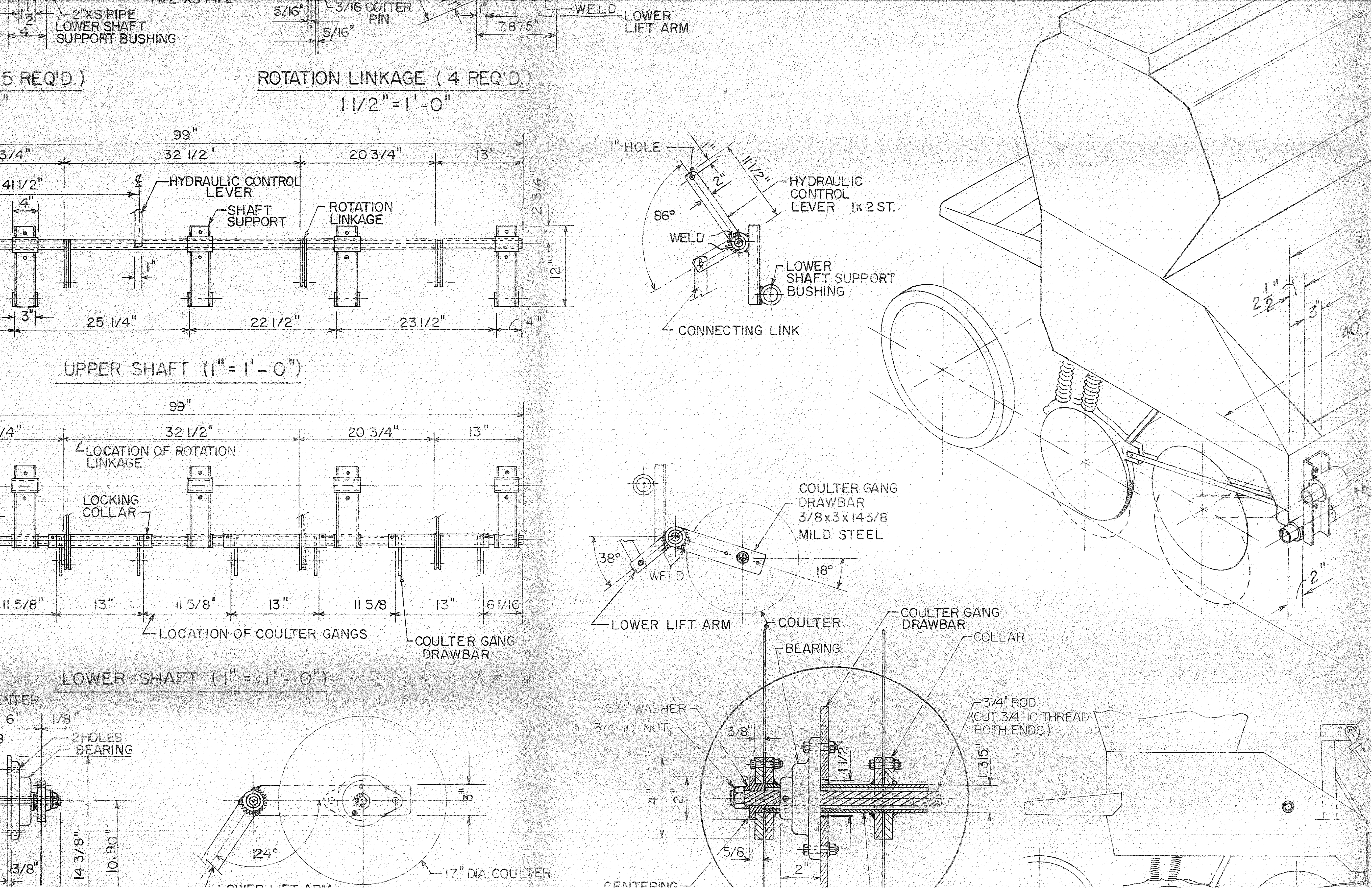
Location	Average W_{hp} hp	Average W_{hp_a} hp
Carman (3.58 mph)	0.45	0.63
Carman (5.14 mph)	0.64	0.91
Glenlea #1	0.42	0.59
Glenlea #2	0.46	0.65

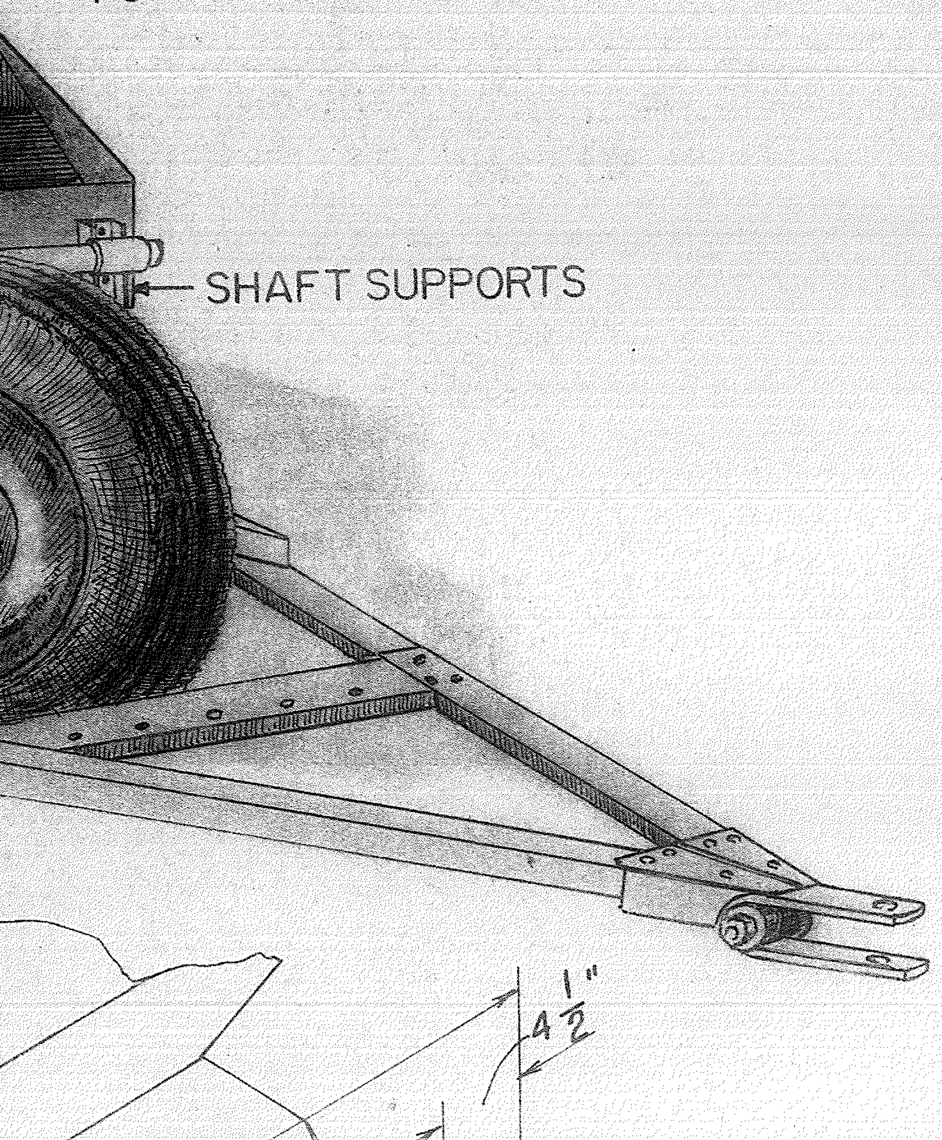


HYDRAULIC CYLINDER SUPPORT
1 1/2" = 1' - 0"



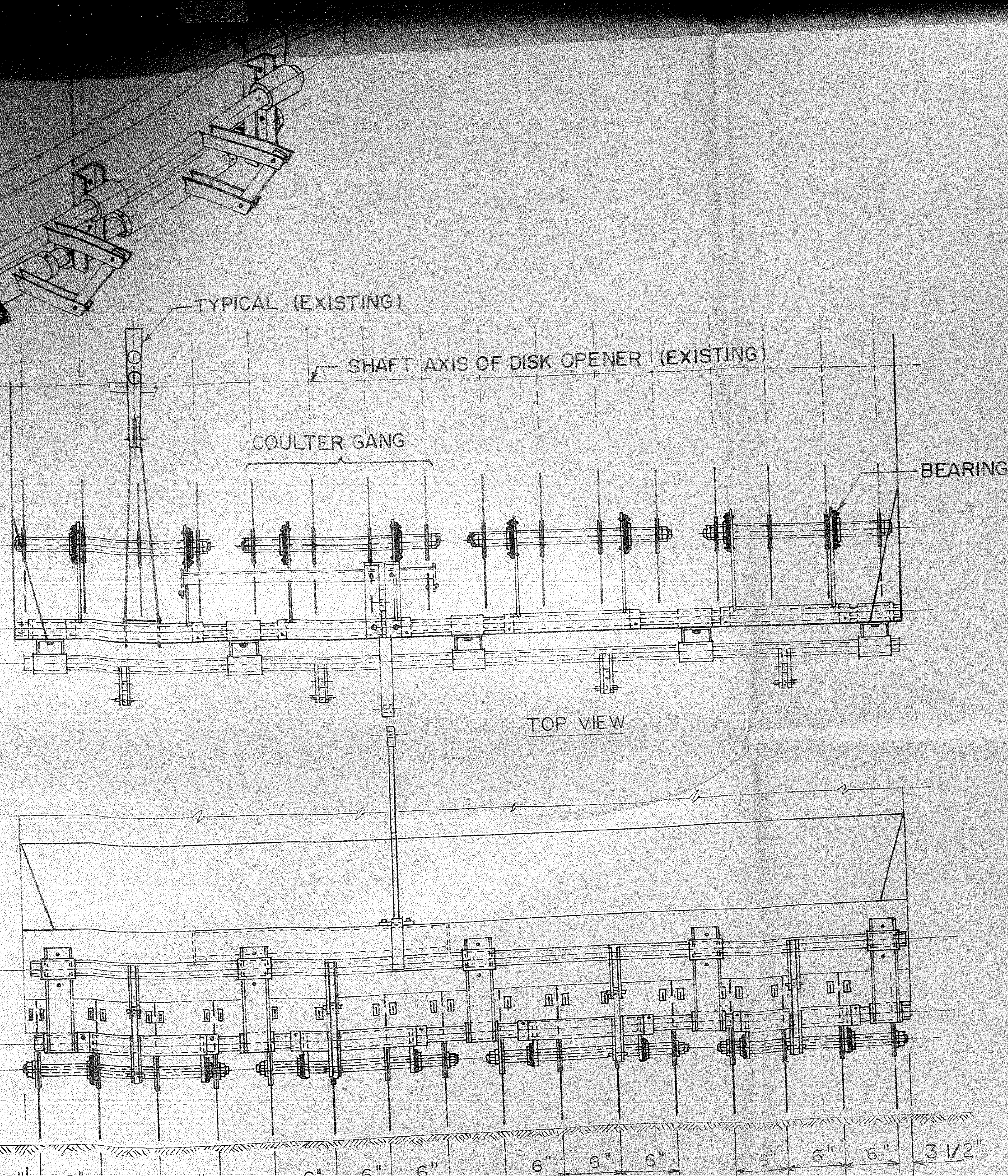



2"XS PIPE
LOWER SHAFT
SUPPORT BUSHING
3/16 COTTER PIN
5/16"
7.875"
WELD
LOWER LIFT ARM
5 REQ'D.)
ROTATION LINKAGE (4 REQ'D.)
1 1/2" = 1'-0"
99"
32 1/2"
20 3/4"
13"
41 1/2"
HYDRAULIC CONTROL LEVER
SHAFT SUPPORT
ROTATION LINKAGE
2 3/4"
12"
25 1/4"
22 1/2"
23 1/2"
UPPER SHAFT (1" = 1'-0")
1" HOLE
11 1/2"
HYDRAULIC CONTROL LEVER 1x2 ST.
86°
WELD
LOWER SHAFT SUPPORT BUSHING
CONNECTING LINK
LOCATION OF ROTATION LINKAGE
LOCKING COLLAR
11 5/8"
LOCATION OF COULTER GANGS
COULTER GANG DRAWBAR
6 1/16
LOWER SHAFT (1" = 1'-0")
COULTER GANG DRAWBAR 3/8x3x14 3/8 MILD STEEL
38°
18°
LOWER LIFT ARM
COULTER
BEARING
COLLAR
3/4" WASHER
3/4-10 NUT
3/4" ROD (CUT 3/4-10 THREAD BOTH ENDS)
1.315"
2 HOLES
BEARING
14 3/8"
10.90"
124°
17" DIA. COULTER
40"
2 1/2"



GENERAL VIEW OF ATTACHMENT LOCATION
(Added Mechanism shown unshaded) 1" = 1'
FOR INTERNATIONAL 620 PRESS DRILL

LIST OF MATERIAL		
QUANTITY	NAME	MATERIAL USED
1	HYDRAULIC CYLINDER SUPPORT	1 - 1 1/2x3x3/16 Channel 28 3/4" long
		1-3/8x4x9 Mild Steel
	GUSSETS	2-1/4x1 1/2x4 Mild Steel
		2-1/4x4x4 1/2 Mild Steel
	COLUMN	1-3/4x3x23.25 Mild Steel
	UPPER MOUNT	1 — 1x2x5 1/2 Mild Steel
	HYDRAULIC CONTROL LEVER	1 - 1x2x11 1/2 Mild Steel
		10-1/2x1 Cap Screws+Nuts, Washers
5	SHAFT SUPPORTS	
		5 - 1 1/2x3x3/16 Channel 12" long
	UPPER and LOWER SHAFT	10 - 2"XS Pipe 4" long
	SUPPORT BUSHINGS	10 - 1/2x1 Cap Screws + Nut, Washers
4	ROTATION LINKAGE	
	UPPER LIFT ARMS	8-5/16x2 Mild Steel 7.35" long
	LOWER LIFT ARMS	8- " " " 7.275" "
	CONNECTING LINKS	4- " " " 10.25" "
	PINS	8- 1/2"ϕ ST. Rod 1 1/8" long
		16- 1/2" Flat Washers
		8 - 3/16x1" Cotter Pins
2	SHAFTS	
	UPPER and LOWER SHAFT	2 - 1 1/2"XS Pipe 99" long
4	COULTER GANGS	
	COULTER DISKS	16 - 17ϕ
	BOLTS	4-3/4 ϕ ST. Rods 22" long
	COLLARS	32-3/8x4 ϕ Mild Steel
	SPACERS	12 - 1"XS Pipe 5.84" long
	BEARINGS	8 - 1-5/16" SC 2-Bolt Flange Bearings
	CENTERING WASHERS	8 - 2"ϕ ST. Rod 1/2" long
		48 - 5/16x1 1/4 Cap Screws + Nuts
		8 - 3/4-10 Nuts + Washers
	COULTER GANG DRAWBARS	8 - 3/8x3 Mild Steel 14 3/8" long
	BEARING PIPE	4 - 2"XS Pipe 13" long
8	LOCKING COLLARS	8 - 2"XS Pipe 1 1/4" long
		8 - 3/8-16 Nuts



1	HYDRAULIC CYLINDER.	8"Stroke, 2-1/2" DIA.
	HYDRAULIC HOSE	

NOTE:

The shorter drawbars on the double disk furrow openers were lengthened to be the same length as the longer drawbars.

In order to increase the depth of penetration of the coulters one of the following steps can be taken:

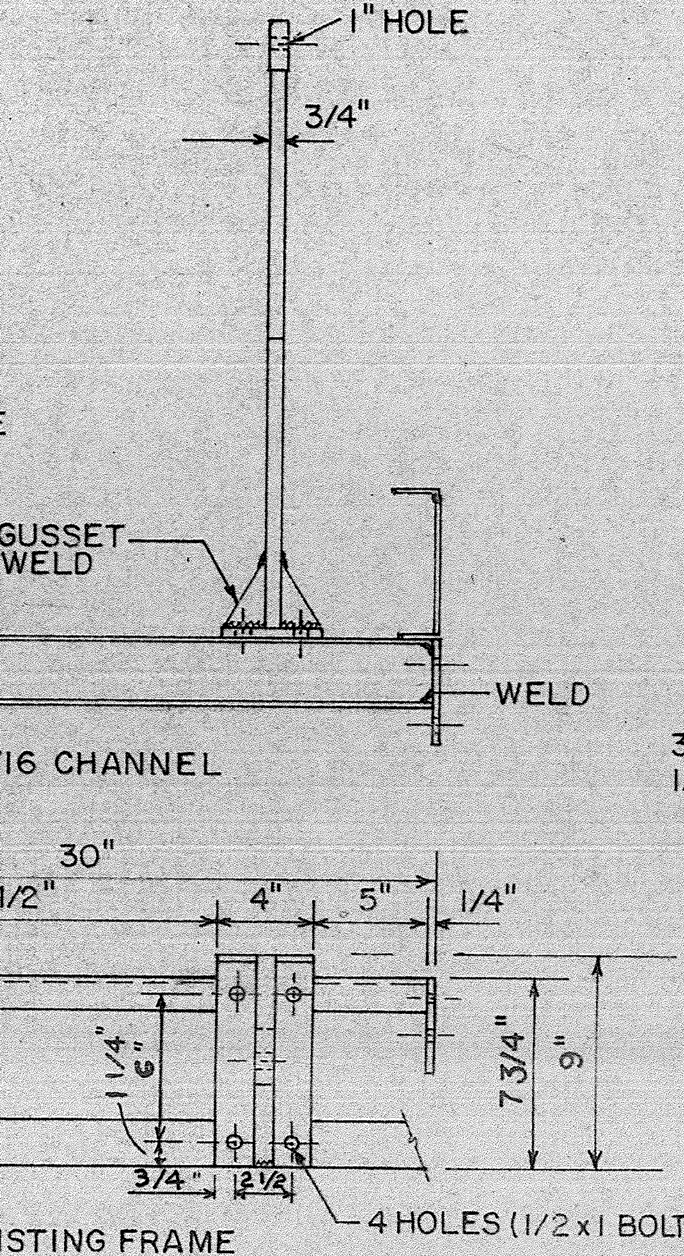
1. Length of coulter drawbars to be increased,
2. Position of lower shaft to be moved further down.

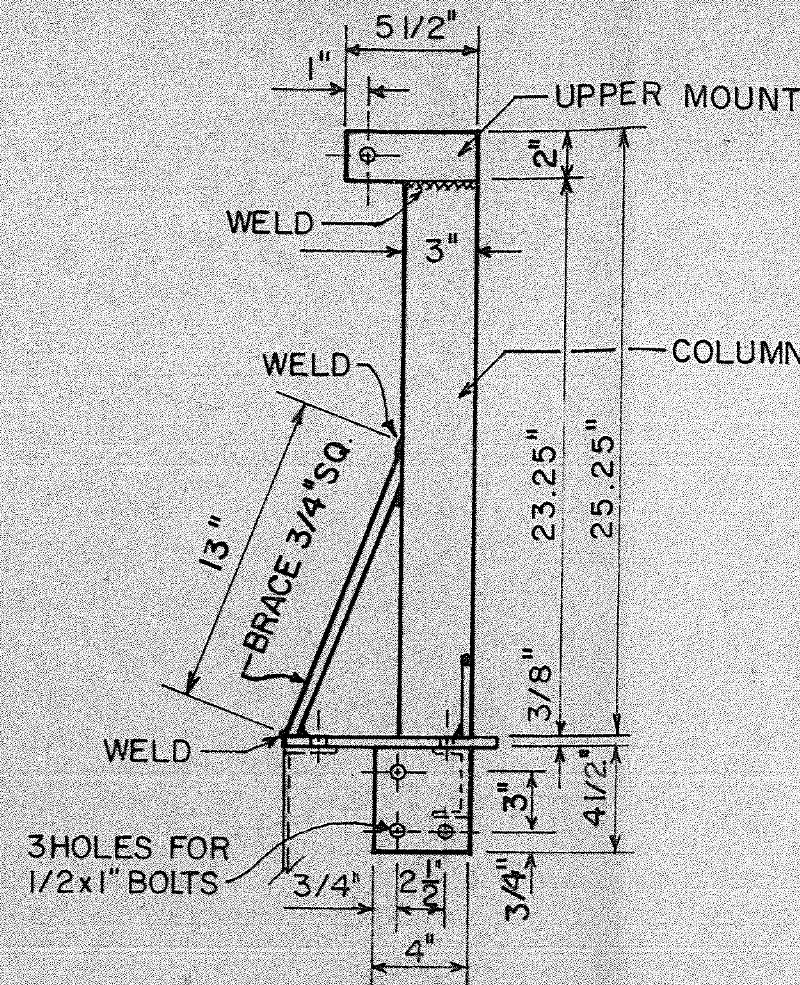
Connecting link can be replaced by heavy duty tension spring in order to prevent damage of the coulters if they hit stones.

Date of Revision	Particulars	by

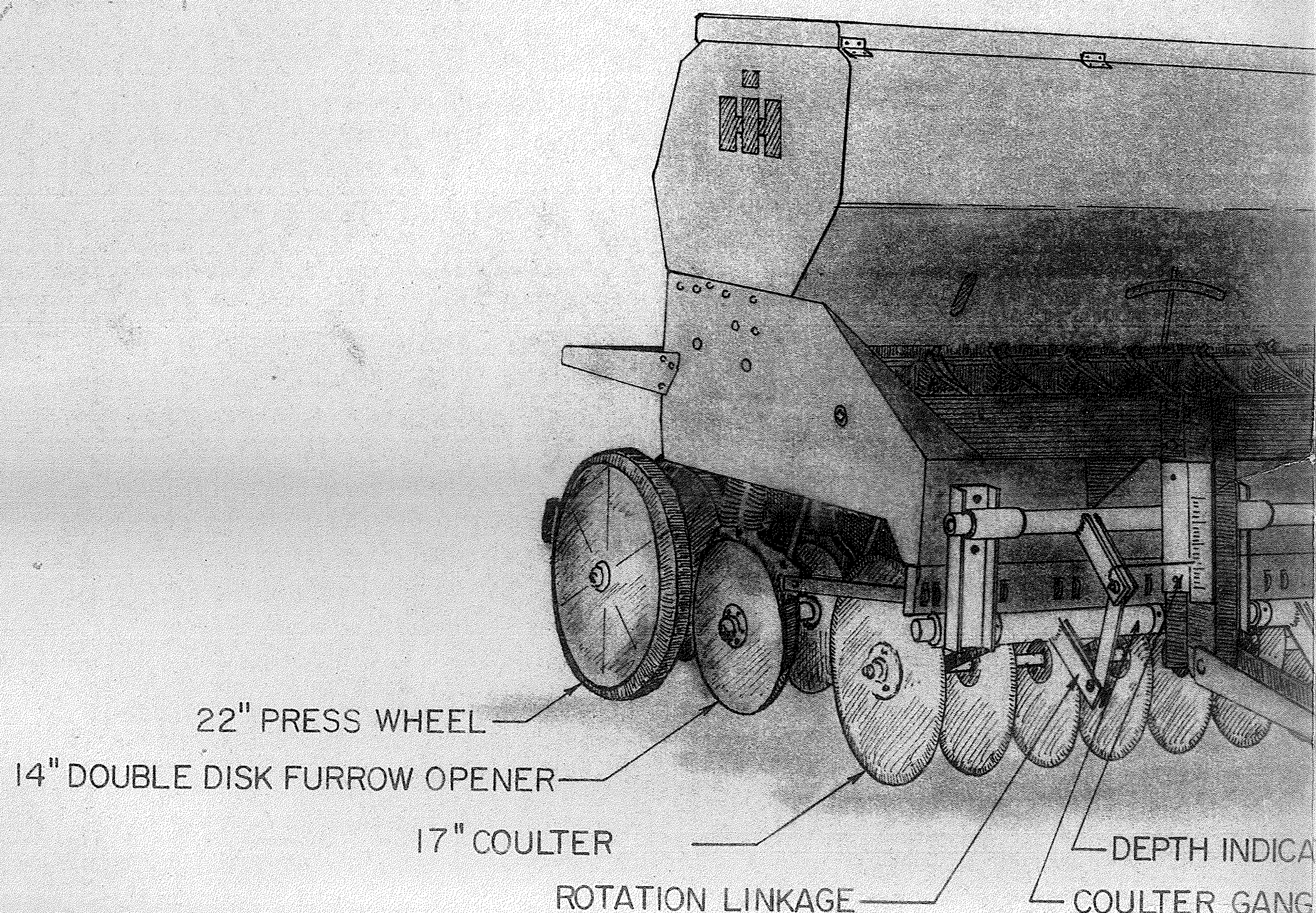
Department of
AGRICULTURAL ENGINEERING
UNIVERSITY OF MANITOBA

TILLAGE ATTACHMENT

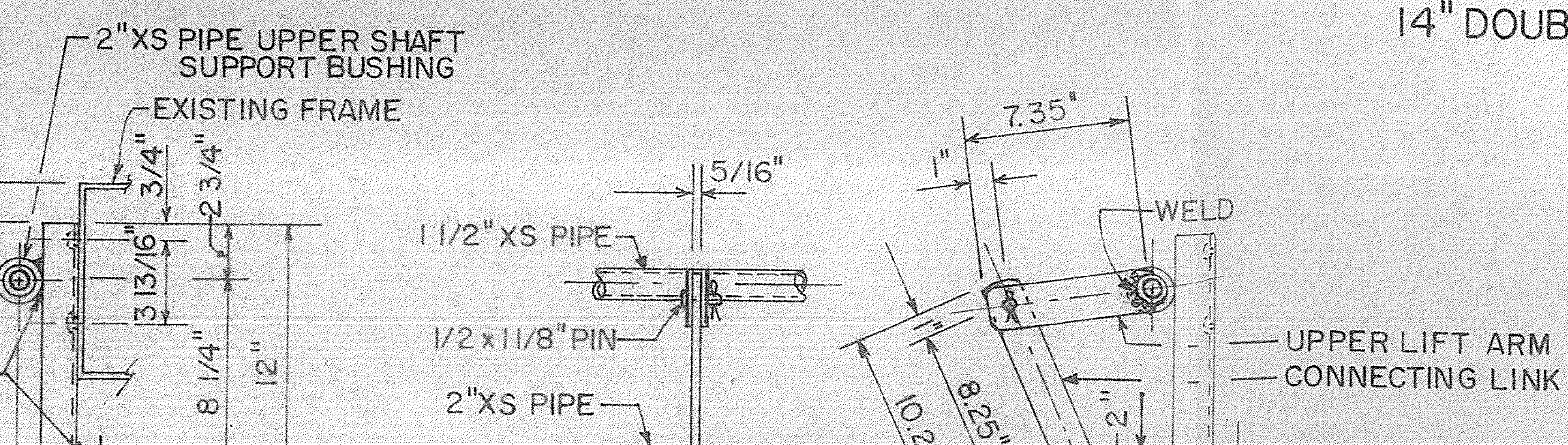


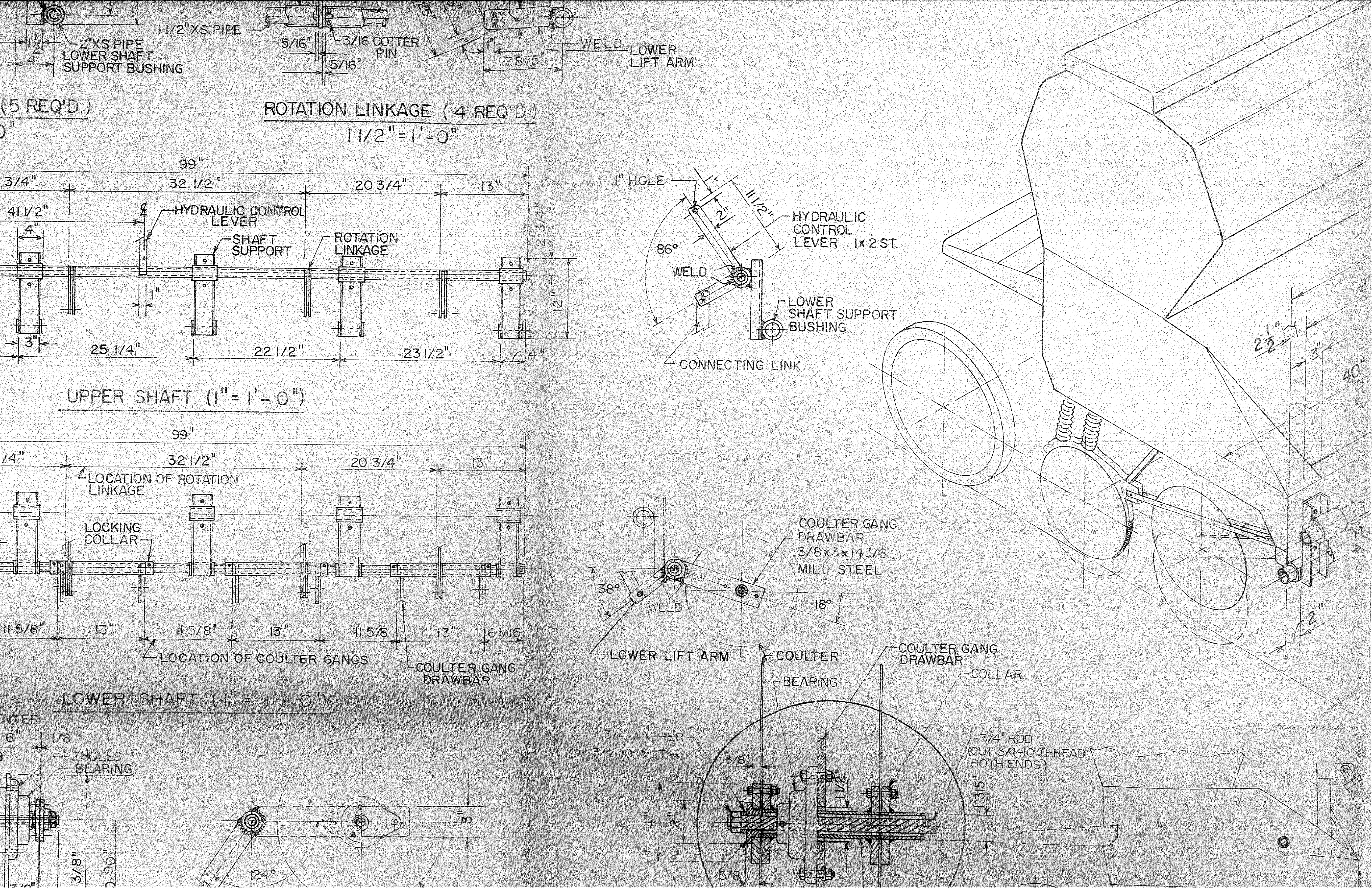


HYDRAULIC CYLINDER SUPPORT
1 1/2" = 1' - 0"



HAFT (1 1/2" XS PIPE)




1 1/2"XS PIPE
2"XS PIPE
LOWER SHAFT
SUPPORT BUSHING
5/16"
3/16 COTTER PIN
5/16"
7.875"
WELD
LOWER LIFT ARM
(5 REQ'D.)
ROTATION LINKAGE (4 REQ'D.)
1 1/2"=1'-0"
99"
3/4"
32 1/2"
20 3/4"
13"
41 1/2"
HYDRAULIC CONTROL LEVER
4"
SHAFT SUPPORT
ROTATION LINKAGE
2 3/4"
12"
1"
3"
25 1/4"
22 1/2"
23 1/2"
4"
UPPER SHAFT (1"=1'-0")
1" HOLE
1"
2"
11 1/2"
HYDRAULIC CONTROL LEVER 1x2 ST.
86°
WELD
LOWER SHAFT SUPPORT BUSHING
CONNECTING LINK
99"
/4"
32 1/2"
20 3/4"
13"
LOCATION OF ROTATION LINKAGE
LOCKING COLLAR
11 5/8"
13"
11 5/8"
13"
11 5/8
13"
6 1/16
LOCATION OF COULTER GANGS
COULTER GANG DRAWBAR
LOWER SHAFT (1" = 1' - 0")
COULTER GANG DRAWBAR 3/8x3x14 3/8 MILD STEEL
38°
WELD
18°
LOWER LIFT ARM
COULTER
ENTER
6"
1/8"
2 HOLES BEARING
3"
3/8"
0.90"
124°
BEARING
COULTER GANG DRAWBAR
COLLAR
3/4" WASHER
3/4-10 NUT
3/8"
1 1/2"
3/4" ROD (CUT 3/4-10 THREAD BOTH ENDS)
1.315"
4"
2"
5/8
2 1/2"
3"
40"
2"