1950

Design and performance of a vibrating cultivator

Syed Aejaz Ali
Iowa State College

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DESIGN AND PERFORMANCE OF A VIBRATING CULTIVATOR

by

Syed Aejaz Ali

A Thesis Submitted to the Graduate Faculty in Partial Fulfillment of The Requirements for the Degree of

MASTER OF SCIENCE

Major Subject: Agricultural Engineering

Approved:

Signatures have been redacted for privacy

Iowa State College
1950
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INTRODUCTION

Cultivation and land tillage are regarded as the fundamental practices influencing crop growth and yield. In the past these practices were not very efficient due to primitive types of plows and other cultivating implements. With the dawn of the machine age, application of modern methods of power to agriculture has led to the invention and development of several convenient and efficient tillage implements and cultivators that are used for the operations of seed bed preparation and weed control.

These modern trends in the realm of farm mechanization are bringing changes in the farm practices together with a reduction of drudgery required in vegetable farming. Truck crop farming has been the progressive and one of the important farm enterprises in the Eastern and North Central United States. According to the latest statistics, 3,511,500 acres of all the cultivated land in the United States is under commercial truck crop production (38).

On an acreage basis truck crops are among the highest cash value crops compared to other general farm crops produced (27). However, the disadvantage of this system of farming is the decrease in net profit due to increased labor and time consumed for hand weeding and cultivation.
After an extensive study of the economic situation facing New Jersey vegetable growers, Carncross and Walter (6) have found that on the average 114-247 man hours and 40-74 horse hours labor is required for the production of an acre of tomatoes. Similar results have been obtained by Robertson (24) of Indiana, who indicated that cultivation in onions is usually done five to twelve times by a wheel hoe, and in addition hand weeding is done twice.

Much work has been done on the development of a vegetable farm cultivator to reduce the excessive man labor required. A variety of such cultivators are available at low initial cost and low cost of operation, but these are still limited in their applications.

The development of an efficient and flexible type of cultivator will have greater demand in the truck crop farming areas. In the words of H. C. Thompson (35):

If cultivation is given often enough to control weeds, the other factors might be ignored. From the standpoint of weed control a cultivator that cuts off the weeds just below the surface of the ground seems to be the best type.

The aim and purpose of this study was to design and construct a cultivator for cutting weeds at the ground level or just below the surface. One of the features of this machine is its ability to get in through vines without injuring the plants and controlling weed growth. Its flexibility also enables it to be used as a tillage tool and a mower. To have
the minimum operating cost, simplicity of the design was given particular consideration.
TOMATO AS A CROP FOR STUDY

Tomatoes are selected as a crop for study due to the following reasons:

1. Tomatoes are typical representatives of the thick-row vegetables where the performance of cultivators becomes a complicated job.

2. Tomatoes form a great portion of truck crop farming. Figure 1 will further emphasize this point.

3. Tomatoes are among those vegetables that consume the highest men hours and tractor or horse hours for cultivation and hand weeding.

For crops like tomatoes, celery, egg plant, lettuce, and water melons, etc., a good job of cultivation is mostly desired requiring the maximum power. The shovels are set quite deep for the first cultivation. As the plants get larger and have wider spread, shovels have to be spread wider and set to run more shallow. "This is where we begin to get into trouble with most of our cultivators," says Stair (29) of Indiana. Down in front where the cultivator beams are attached the frame is too low and not fixed so it can be made wider. If this is done the result is that plants are knocked down.

Close and clean cultivation and killing weeds without injuring the plants is a very important problem in 646,530
Fig. 1. Comparison of tomatoes with all truck crops.
acres of tomatoes grown in the United States. J. M. Huffington (13) in discussing the beneficial practices for tomato production mentions that about six to seven cultivations are desirable for keeping crops free of weeds. Work done by Carncross (6) also supports the above statement.
REVIEW OF LITERATURE

All the experiments conducted on cultivation indicate that its primary purpose is to control weeds. Next to soil erosion the greatest damage caused to the American farms is due to weeds.

Some of the problems caused by the weed population are:
1. Heavy competitions with the plants for moisture, food nutrients, light and air.
2. Increase of labor for a successful crop production.
3. Loss of soil fertility and quality.
4. Weeds act as host to other harmful insects and pests and also cause and promote plant diseases.
5. Some weeds are poisonous and fatal to human and animal life.

It is estimated that weeds levy an annual tax of $3,000,000,000 on American agriculture and industry (15).

In Lachman's opinion, cultivation is still the cheapest means of controlling weeds (15). As in other farm practices, it is the time element which serves as a decisive factor whether or not cultivation will be advantageous.

With better machinery and proper time for cultivation, a greater saving can be expected by weed control. Delayed or infrequent cultivation very often results in unnecessary extra costs, abundant weed growth, additional hand labor for hand
weeding, late maturity and poor yield.

For advantageous results, cultivation should be performed at a rapid rate so as to get a thin soil layer thrown over the uncultivated strip to kill weeds in the crop row.

It is more important to cultivate when there is need for it, and not to cultivate when nothing desirable could be accomplished.

There is no such thing as optimum number of cultivations, since it depends on the soil, weather conditions, weed growth, and perhaps other factors. (35)

This is the opinion of H. C. Thompson, professor of vegetable crops, Cornell University, Ithaca, New York.

Number of cultivations required and the amount of power and labor needed depends to a great extent on the kind of crops grown and also the type of soil on which the crop is growing.

Generally cultivation is given just enough to control weeds, ignoring other factors. A cultivator that cuts off the weeds just below the surface of the ground is about the best type.

When weed growth is heavy late cultivation might do good but when there is light weed growth late cultivation might damage the standing crops.

Experiments conducted on Long Island, New York have yielded the following comparative results for the four practices performed, i.e. cultivation, cultivation half season, scraped, and weed allowed to grow (34).
Table 1
Yields of Crops in Cultivation Experiment on Long Island

<table>
<thead>
<tr>
<th>Crop</th>
<th>No. of Years</th>
<th>Average Yield lb./plot</th>
<th>Weed allowed to grow</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Cult.</td>
<td>Cult. half season</td>
</tr>
<tr>
<td>Tomatoes</td>
<td>7</td>
<td>164.03</td>
<td>166.57</td>
</tr>
</tbody>
</table>

One of the benefits claimed for cultivation over mere scraping is the mulch formation to conserve moisture. This mulch formation phenomenon is explained as one that causes reduction or stops the capillary flow and slows down the surface movement of moisture which then makes its way by diffusion.

Comparative study on cultivation vs. scraping of soil surface was carried out on six crops at Ithaca, New York, for six years on a sandy-gravelly loam soil (33). The treatments were given three times in each case.

Table 2
Average Yield in Pounds Per Plot of Marketable Portion of Crop

<table>
<thead>
<tr>
<th>Crop</th>
<th>Cultivated</th>
<th>Scraped</th>
<th>Per cent increase from cultivation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tomatoes (untrained)</td>
<td>187.96</td>
<td>185.92</td>
<td>1.10</td>
</tr>
<tr>
<td>Tomatoes (trained)</td>
<td>158.61</td>
<td>156.04</td>
<td>1.65</td>
</tr>
</tbody>
</table>
A table comparing the per cent soil moisture calculated on dry-weight basis in cultivated and scraped plots is added to further emphasize the importance of cultivation (30).

Table 3
Average Percentage of Soil Moisture During Growing Season, Calculated on Dry-Weight Basis

<table>
<thead>
<tr>
<th>Crop</th>
<th>1921 Cult</th>
<th>1922 Cult</th>
<th>1923 Cult</th>
<th>1924 Cult</th>
<th>1925 Cult</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tomatoes</td>
<td>7.9</td>
<td>7.4</td>
<td>14.0</td>
<td>12.9</td>
<td>8.7</td>
</tr>
<tr>
<td></td>
<td>9.3</td>
<td>9.8</td>
<td>11.7</td>
<td>11.6</td>
<td>15.1</td>
</tr>
<tr>
<td></td>
<td>12.9</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Cult. = Cultivated Scra. = Scrapped

In the above table (where other crops are omitted), it is indicated that cultivation as compared to scraping resulted in the conservation of moisture in 39 out of 39 comparisons, and in loss in seven comparisons, while in the others the two treatments were equal.

The Two Extremes in Soils for Vegetable Production

Robert D. Sweet (30) after several experiments on cultivation studies of certain vegetables grown on peat soils, has given the following specifications for the two extremes of soil types.

Most desirable for vegetable production

0-10 inches: dark brown, moderately to well decomposed,
woody peat from maple, ash, and elm trees.

10-60 inches: a fibrous peat containing reeds and sedges, and occasionally interspersed with partially decomposed tamarack and other coniferous timber.

60-72 inches: olive-green, partly fibrous, sedimentary peat.

76-81 inches: chara and shell marl.

Lowest in merit for vegetable production

0-4 inches: dark brown, well decomposed, granular, woody peat from maple, ash and elm trees.

4-7 inches: black, nut-like, amorphous, compact and woody peat (because of cultural practices, these two layers are now thoroughly mixed).

7-26 inches: this layer extended to an average depth of 26 inches, but varied from 24-29 inches and was composed of black, compact, columnar, amorphous peat which broke with colloidal fracture and contained decaying twigs and coniferous timber.

26 inches: a layer of blue, sticky calcareous sand of undetermined depth.

Various Factors Affected by Depth of Cultivation

Sweet (30) has done extensive work on the vegetables grown on peat soils and has determined the relative effects
of shallow and deep cultivation on various physical characteristics of vegetable soils.

**Soil temperature**

Shallowness or depth of cultivation had no significant effect on soil temperature at a depth of 2 to 3 inches as revealed by the tests conducted in 1938 and 1939. A few readings taken at a depth of 5 to 6 inches did not show any significant difference.

**Soil moisture**

In all but one case deep cultivation has shown reduction of soil moisture in the upper-most 3 inches of soil.

**Soil nitrates**

Lyon (1922) (30) and Sachs (1926) (26) found that when fallow plots are cultivated they frequently contain more soil nitrates than when scraped. The results of tests conducted by R. D. Sweet show that on the average more soil nitrate is found under deep cultivation.

**Root distribution**

From the studies made on lettuce, onions, carrots and celery on moist and dry peat it was brought out that cultivation had but little effect on these root systems. Because of
the angle of their penetration by the time they had grown out to the place where the cultivator could cut them, the principal lateral roots were below the cultivated zone. Of course, some secondary roots were disturbed but this did not modify the pattern of the root system as a whole. Moreover, cultivation had little effect on roots in the poor soil because roots developed at a greater angle there than they did on the relatively moist or good peat.

Labor Requirements for Truck Crop Production

In an effort to estimate the labor required for various truck crops, an exhaustive search was made for the available information in the college main library. In addition to this, requests were sent to several colleges and universities in the truck crop production regions, asking to furnish with any experimental results on determination of labor required.

After studying and comparing the information obtained from both sources, the data are arranged in tabular form as indicated in Tables 4 and 5. These tables deal with man hours of labor only, so as to emphasize the element of human labor involved which forms the basis of every consideration towards mechanization. In these tables only those crops that have cultivation problems in common are included. A further distinction is made by bringing out the difference in labor consumed in irrigated region and dry farming region, thus showing the
<table>
<thead>
<tr>
<th>Type of labor</th>
<th>Carrots (market)</th>
<th>Celery (market)</th>
<th>Cucumber (market)</th>
<th>Lettuce (market)</th>
<th>Onions EM</th>
<th>Onions LM</th>
<th>Potatoes EM</th>
<th>Potatoes LM</th>
<th>Spinach (State average market)</th>
<th>Tomatoes (State average market)</th>
<th>Watermelon (State average market)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Cultivation, thinning, weeding and hoeing, man hrs.</td>
<td>55</td>
<td>68</td>
<td>32</td>
<td>42</td>
<td>32</td>
<td>68</td>
<td>6</td>
<td>8</td>
<td>25</td>
<td>16</td>
<td>26</td>
</tr>
<tr>
<td>2. Per cent of cultural labor consumed in cultivation operations</td>
<td>62.5</td>
<td>31.9</td>
<td>50.8</td>
<td>62.7</td>
<td>30.5</td>
<td>72.4</td>
<td>13.6</td>
<td>25.8</td>
<td>53.2</td>
<td>15.5</td>
<td>38.8</td>
</tr>
<tr>
<td>3. Per cent of total labor (including harvesting) consumed in cultivation operations</td>
<td>21.7</td>
<td>22.0</td>
<td>17.5</td>
<td>42.0</td>
<td>15.3</td>
<td>46.0</td>
<td>5.7</td>
<td>9.5</td>
<td>13.6</td>
<td>9.6</td>
<td>24.3</td>
</tr>
</tbody>
</table>

EM - Early market

LM - Late market
Table 5

Man Hours of Labor Per Acre
Dry Farming Region

<table>
<thead>
<tr>
<th>Type of Labor</th>
<th>Cucumber</th>
<th>Egg Plant</th>
<th>Green Pepper</th>
<th>Potatoes</th>
<th>Tomatoes</th>
<th>Water Melon</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Cultivating, weeding and hoeing, man hours</td>
<td>56.6</td>
<td>169.2</td>
<td>235.6</td>
<td>2.7</td>
<td>41.6</td>
<td>12.9</td>
</tr>
<tr>
<td>2. Per cent of cultural labor consumed in cultivation operations</td>
<td>28.6</td>
<td>75.9</td>
<td>62.6</td>
<td>9.6</td>
<td>55.5</td>
<td>45.7</td>
</tr>
<tr>
<td>3. Per cent of total labor (including harvesting) consumed in cultivation operations</td>
<td>16.2</td>
<td>35.2</td>
<td>37.5</td>
<td>4.2</td>
<td>21.1</td>
<td>24.2</td>
</tr>
</tbody>
</table>
relative importance of cultivation and weeding operations in both types of farming.

The term "cultural labor" under the column "Type of Labor" refers to all the labor needed for seed bed preparation, preparing rows, fertilizing, planting or setting, pest and disease control, and in case of irrigated region it also included labor for irrigation. The word "market" under the name of the crop indicates the crop produced for commercial markets. Similarly early market and late market point out the seasonal variations and State average market covers all the commercial vegetable production averaged on the State basis (13).

Types of Implements Used for Cultivation and Weed Control in Vegetable Crop Production

Primitive types of wheel-hoe cultivators are still a common sight in some of the commercial vegetable growing areas where labor is relatively cheap. Horse-drawn cultivators are used by some large acreage farmers ranging from one to three horse implements and cultivating two rows at a time. Larger ones are commonly used for potatoes, tomatoes, etc. Tractor drawn cultivators are gradually moving into this picture with the progress of mechanization. These vary in size from small garden tractor cultivators to large-sized farm tractor cultivators similar to that shown in Figure 2 which
Fig. 2. The Tillivator. Ariens Company, Brillion, Wis.
cultivate from seven to nine rows in one operation using an Oliver Cletrac tractor.

Present types of cultivators available in the market can be classified under two different groups:

1. Rotary cultivators
2. Sweep or shovel type cultivators.

Rotary cultivators are very similar to the tillage implements designed to till from 2 to 3 inches. Rotary hoe is the first step towards rotary cultivation. This merely consists of a bar on which two rimless wheels called hoes are mounted. The spokes on the wheels are curved at the ends and penetrate the soil while the wheels roll on the soil. The fundamental principle involved in rotary cultivation is that plants are more resistant to the hoe than the weeds, therefore it is used disregarding the usual pattern.

Harry Seaman developed a double row cultivator in 1926 which operated at a depth of 2 to 3 inches (11). The Tillivator (Fig. 2) is one of the common implements among the rotary cultivators now available. This was designed and built by Ariens Company in 1941 (2). It is used for multi-row cultivation and is primarily meant for track-type tractors. Power is generally supplied by an auxiliary engine, but in later models, power can be used indirectly from the power take-off. Power is transmitted by V-belts to the rotors revolving at 600 rpm. Tillivator tines are riveted to the hubs having widths of one or two inches. Width of cut is increased
or decreased as desired by adding or removing the hubs from the section.

Recently two smaller rotary cultivators have been put out. One of these is the Model T Roto-ette made by Roto-Tiller Inc., Troy, New York. The other is Rotary Tiller built by Milwaukee Equipment Manufacturing Company, South Milwaukee, Wisconsin (14). Both claim better performance for narrow-spaced rows only.

E. N. Scarborough (27) has designed a rotary cultivator unit around a two-inch square steel framing bar as shown in Figure 5. The rotors were mounted on this bar, and clamped to it were the lifting device, the depth control wheels, the bearing supports for the top cross drive shaft and the steering rods. The power was obtained from the power take-off of the tractor. The machine did not work satisfactorily during the 1945-46 trials. After some modifications it was tried again in 1946. It did a good cultivation job but some mechanical defects showed up in the machine itself. Further changes and revamping of the machine enabled better performance in winter and spring of 1947.

Among the rotary cultivators built abroad, the Howard Rotavator (25) manufactured by the Rotary Hoes Limited of England has become very popular and several of these were recently exported to the United States. A hand-controlled 6 horsepower tractor is used with an alternative working width of 20 inches or 24 inches. The rotor is equipped with "L"
Fig. 3. Howard Rotavator. Rotary Hoes Ltd. Essex, England.

Fig. 4. Howard Rotavator. Rotary Hoes Ltd. Essex, England.
Fig. 5. Rotary cultivator. Designed by E. N. Scarborough.

Fig. 6. Garden cultivator. Kruger Mfg. Co., Marshalltown, Iowa.
shaped hoe blades which are capable of cultivating up to a depth of about 8 inches to 10 inches under good conditions. The drive to the rotor is by chain. A Howard Rotavator is shown in operation in Figure 4.

Various types of sweeps and shovels used for cultivation are shown in Figure 7. Among the sweep type cultivators the Power Garden cultivator built by Will-Burt Company, Orrville, Ohio and the Power Cultivator by Le Claire Company, Le Claire, Iowa, are being used for cultivation of narrow-spaced rows.

A small experimental garden cultivator made by Kruger Manufacturing Company of Marshalltown, Iowa, is shown in Figure 6. A 1½ horsepower gasoline engine is used for providing traction power. Sweeps are mounted behind the traction reel and act as a drag, causing the rotor to slip. The machine built for two-row cultivation proved to be unsatisfactory.

In addition to these, many other cultivating machines are manufactured in the United States and abroad. After studying the literature available, the author has attempted to put them in a tabular form, classifying the American and British makes separately, giving the information about each type of cultivator obtained either through private communication or from various farm machinery and implement magazines. (See Tables 6 and 7).
Fig. 7. Types of sweep and shovels.
Table 6

**Truck Crop Cultivators (American)**

<table>
<thead>
<tr>
<th>Trade Name</th>
<th>Type</th>
<th>Size</th>
<th>Effective Width</th>
<th>Engine</th>
<th>Drive</th>
<th>Outstanding Features</th>
<th>Approx. Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>Garden Cultivator (LeClair Mfg. Co)</td>
<td>Shovel wide</td>
<td>25 in.</td>
<td>25 in.</td>
<td>Briggs and Stratton 1 - 1½ hp or 2½hp 4 cycle drive</td>
<td>Positive drive clutch, Adjustable depth control</td>
<td>90 lbs</td>
<td></td>
</tr>
<tr>
<td>Gravely Motor Cultivator (Gravely Company)</td>
<td>Shovel W = 13½ in. (inter-row spacing)</td>
<td>15 in.</td>
<td>Gravely 5 hp. 4 cycle</td>
<td>Gear drive automatic type clutch</td>
<td>Double acting cone type clutch</td>
<td>210 lbs</td>
<td></td>
</tr>
<tr>
<td>Powered Cultivator (Will-Burt Co.)</td>
<td>L = 63 in. W = 28 in. H = 38 in.</td>
<td>14 in.</td>
<td>Briggs and Stratton, 1 hp. 4 cycle</td>
<td>Roller chain</td>
<td>Adjustable handle height, Steel traction wheel</td>
<td>150 lbs</td>
<td></td>
</tr>
<tr>
<td>Rotary Tiller (Jr.) (Milwaukee Equip. Co)</td>
<td>Rotary</td>
<td>-</td>
<td>12 in.</td>
<td>3 hp. engine</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Tillivator (Ariens Co.)</td>
<td>Rotary for 3-6 row, adjust.</td>
<td>27 in.</td>
<td>Oliver Cle-trac tractor</td>
<td>V-belt, Either in front. Uniform depth on uneven ground.</td>
<td>8&quot; ground clearance</td>
<td>1¼ - 3 mph, 4 - 8 row job.</td>
<td>129 lbs</td>
</tr>
</tbody>
</table>
### Table 7

#### Truck Crop Cultivators (British)

<table>
<thead>
<tr>
<th>Trade Name</th>
<th>Type</th>
<th>Size</th>
<th>Effective Width</th>
<th>Engine</th>
<th>Drive</th>
<th>Outstanding Features</th>
<th>Approx. Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Acre</strong></td>
<td>Rotary W</td>
<td>L - 74”</td>
<td>21”</td>
<td>“Villiers” 3hp</td>
<td>Chain and sprockets</td>
<td>Depth control</td>
<td>3 cut</td>
</tr>
<tr>
<td></td>
<td>W - 21”</td>
<td></td>
<td></td>
<td>4 stroke, single cylinder</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>H - -</td>
<td></td>
<td></td>
<td>“Renold”</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>“Auto Culto” (Allen &amp; Simmonds Ltd.)</strong></td>
<td>Rotary</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Cultivating blades offset from the machine</td>
<td></td>
</tr>
<tr>
<td><strong>Bantam Rotavator (Rotary Hoe Ltd.)</strong></td>
<td>Rotary</td>
<td>L - 62”</td>
<td>10”</td>
<td>“Villiers”</td>
<td>Power drive</td>
<td>2 speed gear box. 0”-8” adj. depth of cultivation</td>
<td>135 lbs</td>
</tr>
<tr>
<td></td>
<td>W - 15”</td>
<td></td>
<td></td>
<td>2 stroke</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>H - -</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>British Anzani Motor Hoe (British Anzani Ltd.)</strong></td>
<td>Shovel</td>
<td>13” wide</td>
<td>8”</td>
<td>Anzani J.A.P.</td>
<td>Direct clutch when throttle is open. 1-2½ mph</td>
<td>104 lbs</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>to 13”</td>
<td>1 hp. 4 cycle engine</td>
<td></td>
<td>Adj. width (up to 2’) &amp; handle height</td>
<td></td>
</tr>
</tbody>
</table>

Continued on next page
<table>
<thead>
<tr>
<th>Trade Name</th>
<th>Type</th>
<th>Size</th>
<th>Effective Width</th>
<th>Engine</th>
<th>Drive</th>
<th>Outstanding Features</th>
<th>Approx. Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>&quot;Clifford&quot;</td>
<td>Rotary</td>
<td>L - 48&quot;</td>
<td>11&quot;</td>
<td>J.A.P. 1.2 hp.</td>
<td>Power drive</td>
<td>depth of cultivation up to 2 cut</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>H - 30&quot;</td>
<td></td>
<td>4 stroke</td>
<td></td>
<td>3 in.</td>
<td></td>
</tr>
<tr>
<td>&quot;Colwood&quot; Rotary Hoe</td>
<td>Rotary</td>
<td>L - 60&quot;</td>
<td>9&quot;</td>
<td>Villiers MK.10</td>
<td>Roller</td>
<td>3 speed gear box</td>
<td></td>
</tr>
<tr>
<td>(Dashwood Eng. Co.)</td>
<td></td>
<td>H - 36&quot;</td>
<td></td>
<td>1.2 hp.</td>
<td>chain drive</td>
<td>1-2½ mph. Depth of control.</td>
<td>170 lbs</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Standard power take off pulley</td>
<td></td>
</tr>
<tr>
<td>&quot;Commando&quot; (Power Hoes Ltd.)</td>
<td>Rotary</td>
<td>9&quot;-14&quot;</td>
<td>Varies</td>
<td></td>
<td>Gear drive</td>
<td>Speed ½ - 1½ mph</td>
<td>165 lbs</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gravely Motor Cultivator</td>
<td>Shovel</td>
<td>L - 62&quot;</td>
<td>15&quot;</td>
<td>Gravely 5 hp.</td>
<td>Gear drive</td>
<td>Double acting</td>
<td>210 lbs</td>
</tr>
<tr>
<td>(Gravely Overseas Ltd.)</td>
<td></td>
<td>H - 25&quot;</td>
<td></td>
<td>4 cycle</td>
<td></td>
<td>Automotive cone type clutch</td>
<td></td>
</tr>
<tr>
<td>Howard Rotavator</td>
<td>Rotary</td>
<td>L - 72&quot;</td>
<td>20&quot;</td>
<td>&quot;Gem&quot; 6 hp.</td>
<td>Chain depth</td>
<td>5½ cut</td>
<td></td>
</tr>
<tr>
<td>(Rotary Hoes Ltd.)</td>
<td></td>
<td>H - 19&quot;</td>
<td></td>
<td>single cylinder, 4</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>stroke</td>
<td>Chain</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wrigley (Wessex Industries Ltd.)</td>
<td>Single</td>
<td>L - 66&quot;</td>
<td>10&quot;</td>
<td>J.A.P. 1 hp.</td>
<td>V-belts and</td>
<td>worn reduction combination</td>
<td>120 lbs</td>
</tr>
<tr>
<td></td>
<td>wheel</td>
<td>W - 19&quot;</td>
<td></td>
<td>4 stroke</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>type</td>
<td>H - 34&quot;</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
EXPERIMENTAL

Objectives

Mechanization of vegetable crop production can be greatly improved by an efficient cultivator for an effective weed control without causing serious damage to the tender vines of the plants.

The cultivation and weeding operations in truck crop farming are either carried out manually consuming more man-hours, or performed by machines which being inflexible, knock the plants down while operating in the fields where the vines of the plants have out-grown affecting the inter-row spacing.

Under such circumstances a machine which will do the weed control job without damaging the plants will be very useful and time-saving for commercial vegetable growers.

The main purpose of the work mentioned in this thesis was to design and construct a cultivator that would meet the following requirements:

1. A machine that must give an effective weed control.
2. A machine that must operate nearer to and under the plants and get in between them without disturbing or ruining the plants.
3. The machine must operate faster than hand weeding thus reducing man hours of labor.

4. The shearing resistance offered by the soil while moving the blades through the soil should be minimum.

5. The blades or shovels must not penetrate deeper than one inch in the ground so as not to harm the main roots of the plants.

6. The moving parts of the weed cutting mechanism must be protected from dirt and dust.

7. The machine must be free from undesired vibrations so as to ensure a smooth operation.

8. It should be a very low cost machine with low maintenance and operating cost.

9. The machine must be compact and easily handled.

10. It must be easy to operate.

The above factors were consistently emphasized all through the development and design of the vibrating cultivator. After completing the design, the machine was constructed to full scale. Some of the parts that were either difficult or uneconomical to construct were obtained from the College Central Stores.

Design and Construction

Research work on the development of a cultivating machine to work satisfactorily in solanaceous and similar truck crops was started in the early part of the summer of 1949. An ex-
Experimental machine with a vibrating sickle using the cement vibrator (1) was constructed and tried during the summer. It did not prove satisfactory due to high frequency vertical vibrations produced. After some changes made to eliminate vertical vibrations, the machine was tried again in the later fall 1949. The new attachment eliminated vertical vibrations but not entirely. The things noticed during the later test were:

1. Vibrating mechanism used in this and previous test was not very desirable from the standpoint of greater vertical vibrations.

2. Clogging of the soil against the rear bend of the soil shearing sickle created a problem, for it produced resistance against the forward motion of the machine.

In the experimental machine constructed later, efforts were made to overcome these difficulties by taking them into account during the design work.

**General outline of the design**

For the convenience of starting the design work on the actual machine the following specifications were used:

1. The width of cutting blade was taken as 24 inches.
2. The maximum depth of blade penetration was set for one inch.
3. The cutting blade was provided lateral vibrating action by rotating off-centre disks.
4. The rotating mechanism was covered by an outer plate shell having the slope for optimum resistance to the soil shear. Moreover, the soil going up the slope was to fall sideways due to lateral vibrations, before getting into the rotating disk assembly.

The entire structure can be divided into the following parts:

1. The front part consisting of the vibrating mechanism.
2. The frame work.
3. The power transmitting unit.

Figures 8 and 9 will further clarify the statement.

The Vibrating Mechanism

The vibrating mechanism mainly consisted of eccentric loads rotating freely on a fixed vertical shaft. A pulley that was hollowed on one side received power from the source. This pulley was made up of two parts as shown in Figure 10, the upper half being a cup-shaped steel half-pulley and the lower half made out of cast iron and hollowed on one side. Another eccentric weight was furnished by an off-centre cast iron disk rotating in the opposite direction from the pulley made eccentric for balancing the centrifugal force due to the off-centre pulley.

Three gears with equal number of teeth were used to transmit power from the pulley to the bottom disk. The gear
Fig. 8. Assembly drawing - left side view.
Fig. 9. Assembly drawing - top view.
Fig. 10. Vibrating mechanism.

Main

Pulley with Eccentric Weight

Protecting Shield

Off-centre Disk

Cutting Blade

Hollowed Port

Scale: 1" = 2"
press-fitted in the lower half-pulley delivered power to the
meshing gear rotating on a fixed horizontal shaft. This gear
in turn meshed with the gear mounted on the off-centre disk.
The top views of the lower half pulley and disk and also the
centre of gravity of each are shown in Figure 11.

The entire rotating unit was shielded by a shell to
which the stationary shaft was welded. The shell was made of
a 1/16" thick sheet metal and consisted of the following:

1. A base 24"x9".

2. A rear wall 24"x3" perpendicular to
the base.

3. A plate 24"x9½" welded to the front end
of the base plate and to the top of the
rear wall at a slope of 3:1.

The two sides were also covered by 1/16" steel plates
welded to the main shell. In addition to this a protecting
shield having the shape of a mold-board plow was brazed to
the top plate on either side of the rotating unit so as to
keep the dirt away.

In front of the shell eight International Harvester mower
blades each 3 inches wide were riveted at the lower side of
the shell base. These blades were bent at the front so as to
have the desired suction in the ground.

When the power was supplied, the entire vibrating assembly
as a unit was imparted lateral vibrations caused by the centri-
fugal forces of the rotating parts.

The vibrating principle of the unit is illustrated in
Fig. 11. Top views of rotating disk and pulley.
Figures 12, 13, 14 and 15, where four stages of the centrifugal forces of the pulley and disk are shown. The first condition was at the instant when the centre of gravity of both eccentric weights was at the same side of the axis of rotation, and the centrifugal forces were added to give a lateral movement. After a 90° rotation the centrifugal forces were counteracting each other which resulted in a state of equilibrium of forces. After another 90° turn, both eccentric loads were again on the same side of the axis of rotation giving lateral motion to the vibrating unit. Rotation of the loads another 90° resulted in the cancellation of centrifugal forces which as in the second case were equal and acting on the opposite sides of the axis of rotation thus imparting no vibrations.

This principle was worked out to avoid all the undesirable vibrations in several directions and to have a system with vibrations in the desired directions only.

The entire vibrating mechanism was supported by two spring steel bars fixed at one end to the main frame and hinged on the other end to the rear wall of the shell. Hinge joints were used to have the minimum dampening effect on the vibrations. The spring bars were used as supports with the intention of having additional vibrations imparted to the vibrating front and also to allow the unit to vibrate independent of the frame.
Fig. 12

The first two stages of centrifugal forces.

Fig. 13

Scale: 1' = 2"
The last two stages of centrifugal forces.
Supporting Framework

The frame which supported the vibrating unit and on which the engine was mounted is shown in Figure 16. The rectangular frame 23"x10" was made out of 1½"x1½"x3/16" angles. On either side of the frame the spring steel bars supporting the vibrating unit at the hinge joint were bolted. At the front part of the rectangle a steel plate with a vertical stationary shaft was welded. An idler cone pulley rotating on the stationary shaft was used to transmit power from the engine to the rotating unit in the front. A similar steel plate welded at the rear end of the rectangle supported the engine. Two vertical bars on the sides were used to support the frame on the horizontal stationary shaft of the moving wheels.

The frame when mounted on wheels stood 1½" above the ground level and the vibrating unit was resting at the ground surface.

Power Transmission

A small air-cooled 4-cycle gasoline engine of the following specifications was used for supplying power to the vibrating unit:

Make - Lauson Company, New Holstein, Wis.
Horsepower - 3/4
RPM - 4000
Bore - 1 3/4 inch
Stroke - 1 1/2 inch
No. of cylinders - 1
Fig. 16. Framework - top view.

- Guides for Engine Slide
- Engine Sliding Slot
- Cone Pulley
- Stationary Shaft
- Bearing 1/16" thick
- Welded Pipe 1/8" thick
- Spring Steel Bar 2 1/4" x 2 1/8" x 2 1/8"
- Spacers on 1/4" Bolts
- Steel Plate 8" x 2 1/4" x 2 1/8"

Scale: 1" = 4"
The idler cone pulley mounted on a vertical stationary shaft received power from the engine pulley on a horizontal shaft by means of a quarter turn belt.

The minimum centre distance between the two perpendicular shafts was obtained by the formula (19):

\[ C.D. = 6(D + B) \]

Where \( D \) = the diameter of the larger pulley
\( B \) = its width.

In this particular case, both the driving and driven pulleys were of the same size. Using the dimensions of one of them:

\[ C.D. = 6(2.0 + 0.75) \]

Center Distance = 16.5 inches

However, this centre distance is for V-belts in continuous service. A value of 15.5 inches was taken for the centre distance so as to suit the available conditions assuming that in intermittent operation, reducing the centre distance would not vary the conditions much.

The centre distance between the idler pulley and the rotating pulley in the vibrating unit was taken as 7 inches which according to the specifications (19), was greater than the diameter of the large pulley and smaller than the sum of the diameters of both pulleys. The distance was closer to the diameter of the large pulley because on high-speed drives short centre distances give smoother running.
Engine sliding mechanism

The engine sliding device was used to tighten or loosen the belt between the engine and the idler pulley. A slot 7/16" wide and 2" long was made in the steel plate welded to the rear of the rectangular frame structure. A 7/16" bolt passing through the engine base was fastened to the slotted plate.

A lever arrangement was used to slide the engine back and forth so as to make the quarter turn belt tight or loose thus transmitting or cutting out power to the vibrating unit any time desired. This mechanism in fact works in place of a clutch and gave a better operating control. The lever arrangement is shown in Figure 17. A steel bar 1/8"x1" was used to construct the simple device shown. The horizontal lever just under the handle lever was lifted and either pushed or pulled according to the need.

No mechanical power was supplied to the wheels carrying the entire assembly.
Release Lever
1/4 Rod
1/2 Rod
18" x 1" x 1/8"
Coil Spring
Lever
Engine
Steel Plate 9" x 3" x 1/2"
Engine Base 7" x 5" x 1/2"
Angle 1 1/2" x 1/2" x 1/16"
Slot 2" x 1/16"
Bolt 7/16"
Vibration Analysis of the Vibrating Unit

Natural frequency

The vibrating unit was supported by the spring steel bars of equal size on either side. This entire unit had vibrations in the lateral direction only. For the analysis of the system the two spring bars were assumed as a beam fixed at one end and the weight of the vibrating mechanism at the other end.

Having the system set-up as a cantilever beam the maximum force giving a deflection $x$ was obtained as

$$ F = \frac{3 \, EI \, x}{L^3} \quad (9) $$

where $E =$ the modulus of elasticity of the spring steel.

$I =$ the moment of inertia of the two bars about the axis perpendicular to the plane of the paper.

$x =$ the deflection of the bars

$L =$ the length of the one bar.

Due to the reversible nature of the centrifugal force which acts in either direction with the same magnitude, the average force will be equal to the maximum force, and the
potential energy will then be equal to:

\[
\text{Potential Energy} = \frac{3 \, EI \, x^2}{L^3}
\]

Knowing that the weight of the vibrating unit did not have a variable effect on the system, the kinetic energy stored in the weight was

\[
KE = \frac{1}{2} \left(\frac{W}{g} \right) \left(\frac{dx}{dt}\right)^2
\]

Summing up the kinetic and potential energy:

\[
\frac{1}{2} \frac{W}{g} (\frac{dx}{dt})^2 + \frac{3 \, EI \, x^2}{L^3} = C
\]

where \(C\) is a constant.

Differentiating of the above equation with respect to time and simplifying, the following equation can be obtained:

\[
\frac{W}{g} \frac{d^2x}{dt^2} + \frac{6 \, EI}{L^3} x = 0
\]

From which the equation for frequency of the system will be:

\[
f = \frac{1}{2 \pi} \sqrt{\frac{6 \, EI \, g}{WL^3}} \quad \text{cycles/sec.}
\]

Before calculating the natural frequency of the system, the moment of inertia of the bars was determined considering both bars as a built-up beam.

Referring to Figure 18 the moment of inertia was determined in the following manner:
Fig. 18. Free body diagram and moment axis of the vibrator.

Free body Diagram of Vibrating Unit and its Supports.

Axis About which Moment of Inertia was Determined

Centrifugal Forces:
- a) 116.7 lb
- b) 115.5 lb

Weight of the Unit: 23 lb
\[ I = I_0 + Ad^2 \]
\[ I_0 = \frac{bh^3}{12} \]

where \( b = 1\frac{1}{2}'' \)
\( h = 3/16'' \)
\( A = \) the cross-sectional area of the bar as shown in the figure.
\( d = \) the distance from x-axis to the centre of the rectangle.

\[ I = 2 \left[ \frac{(1.5(0.125)^3)}{12} + (1.5 \times 0.125)(5)^2 \right] \]
\[ I = 9.36 \text{ in.}^4 \]

\( E = \) modulus of elasticity = 30 x 10^6 psi
\( W = 23 \text{ lb.} \)
\( L = 22 \text{ in.} \)

Substituting the actual numerical values in the frequency equation, the frequency equals:

\[ f = \frac{1}{2\pi} \sqrt{\frac{6 \times 30 \times 10^6 \times 9.36 \times 386.4}{23 \times (22)^3}} \]

Natural Frequency = 26 cycles/sec.

The true vibrating frequency of the unit will be proportional to the power supplied and the engine speed.

**Maximum velocity**

The maximum velocity of a particle with simple harmonic motion is the product of the radius of the related circle
(the amplitude) and its angular velocity.

The amplitude is the maximum deflection of the system due to the force. The graph (16) shown in Figure 19 giving relationship between natural frequency and the deflection was used. The value read for a frequency of 1560 cycles per minute, was 0.0015 inches.

Therefore the amplitude was

\[ \text{Amplitude} = 0.00125 \text{ feet.} \]

The angular velocity is \(2 \pi f\)

\[= 9800 \text{ rad./min.} \]

The maximum velocity is \(9800 \times 0.00125\)

\[= 12.25 \text{ feet per min.} \]

**Transmissibility**

Using the graph (16) in Figure 20 the disturbing frequency of 2600 cycles per minute was obtained. Calculating the ratio of disturbing frequency to natural frequency and using the graph (17) of Figure 21 the transmissibility value read from the graph was 70 per cent, indicating that 0.7 of the vibrations were transmitted from the unit to the supporting spring steel bars.
Fig. 19. Relationship between static deflection and natural frequency.
Fig. 21. Relationship between transmissibility and ratio of disturbing frequency to natural frequency.
Critical Speed of the Eccentric Pulley and Disk

The uniform 5/8 inch stationary shaft 5 inches long between two supports carried the pulley and the eccentric disk. The weights of the pulley and disk being very low compared to the centrifugal forces were neglected and only the centrifugal forces were considered.

In determining the critical speed, the shaft was assumed to be a beam of uniform section. The deflections due to the centrifugal forces were obtained analytically by using the principle of superposition which states that the total deflection caused by several weights is equal to the deflection caused by any one weight plus the deflections at the same point caused by each of the other weights acting individually.

The deflections at the two points where the centrifugal forces were acting was obtained first by using the equation for deflection due to a concentrated load (9).

\[ y = \frac{Pbx}{6EI} (L^2 - b^2 - x^2) \text{ for } x \leq a \]

From this, the deflection under the centrifugal force of the pulley was

\[ y_1 = \frac{Pa^2b^2}{3EI} \]

where P (centrifugal force) was expressed in terms of speed.
\[ p = \frac{3.25 \times 0.26 \omega^2}{32.2} \]
\[ P = .00262 \omega^2 \]
\[ a = 7/8" \]
\[ b = 4 1/8" \]
\[ L = 5" \]
\[ E = \text{modulus of elasticity} = 30 \times 10^6 \text{ psi} \]
\[ I = \frac{\pi d^4}{64} = .00745 \text{ in}^4 \]

\[ y_1 = \frac{0.00262 \omega^2 (7/8)^2 (4 1/8)^2}{3(30)10^6 (.00745) 5} \]
\[ y_1 = 1 \times 10^{-8} \omega^2 \]

The deflection under the centrifugal force of the disk was:

\[ y_2 = \frac{0.00284 \omega^2 (15/16)^2 (4 1/16)^2}{3(30)10^6 (.00745) 5} \]
\[ y_2 = 1.225 \times 10^{-8} \omega^2 \]

The deflection at the pulley due to centrifugal force of the disk was:

\[ y_3 = \frac{0.00262 \omega^2 (7/8)(15/16) \left[ \frac{5^2}{2} - 7/8^2 - 15/16^2 \right]}{6(30)10^6 (.00745) 5} \]
\[ y_3 = 1.075 \times 10^{-8} \omega^2 \]

The deflection at the disk due to the centrifugal force of the pulley was:

\[ y_4 = \frac{0.00284 \omega^2 (15/16)(7/8) \left[ \frac{5^2}{2} - 15/16^2 - 7/8^2 \right]}{6(30)10^6 (.00745) 5} \]
\[ y_4 = 1.165 \times 10^{-8} \omega^2 \]
The total deflections at the pulley and disk were:

\[ y_{\text{pulley}} = 1 \times 10^{-8} \omega^2 + 1.075 \times 10^{-8} \omega^2 \]
\[ = 2.075 \times 10^{-8} \omega^2 \]

\[ y_{\text{disk}} = 1.225 \times 10^{-8} \omega^2 + 1.165 \times 10^{-8} \omega^2 \]
\[ = 2.39 \times 10^{-8} \omega^2 \]

The natural frequency (36):

\[ f = \frac{1}{2 \pi} \sqrt{\frac{g(W_p Y_p + W_D Y_D)}{W_p(Y_p)^2 + W_D(Y_D)^2}} \]

\[ f = \frac{1}{2 \pi} \left[ \frac{386.4 \times (0.00262 \omega^2)(2.075 \times 10^{-8} \omega^2)}{(0.00262 \omega^2)(2.075 \times 10^{-8} \omega^2)^2} \right]^{\frac{1}{2}} \]
\[ + (0.00284 \omega^2)(2.39 \times 10^{-8} \omega^2) \]
\[ + (0.00284 \omega^2)(2.39 \times 10^{-8} \omega^2)^2 \]
\[ = \frac{1}{2 \pi} \sqrt{\frac{4.74 \omega^4}{0.0274 \times 10^{-8} \omega^6}} \]

\[ f^2 = \frac{1}{4 \pi^2} \frac{173 \times 10^8}{\omega^2} \]

Since the natural frequency for lateral vibration is the same as the critical speeds for a rotor

\[ \omega^2 = (f \times 2 \pi)^2 \]

Substituting the above value for \( \omega^2 \)

\[ f^4 = 0.111 \times 10^8 \]

\[ f = 0.578 \times 10^2 \text{ cycles/sec.} \]
Critical Speed = 3470 rpm.

The speed thus determined was much above the maximum speed of the pulley which was 2000 rpm.

Stress Determination in the Vibrating Mechanism Shaft

The 5 1/4" pulley with off centre weight received power horizontally from the idler pulley on the frame as is shown in Figure 22. The figure also illustrates various forces acting on the fixed shaft. The power to the off-centre disk was delivered by the bevel gear mounted on the pulley through a gear train.

Assumptions made:

1. The eccentric weights of the pulley and the disk were both on the same side of the shaft axis where the centrifugal force will be greatest and the stresses will be maximum.

2. The horsepower transmitted was assumed to be 3/4 hp (neglecting slippage and friction losses on quarter turn and horizontal belts; for these effects were considered in determining pulley rpm with respect to engine shaft rpm.)

3. It was assumed that the 5 1/4" pulley was rotating at 2000 rpm (the maximum possible value of the rpm was used so as to evaluate the maximum stresses produced).

4. It was assumed that the shaft was made of SAE 1045 steel (yield point 58,000 psi, and ultimate stress 72,000 psi).
Fig. 22. Free-body diagram for stress determination in vibrating mechanism shaft.
5. The load was assumed to be applied gradually.

6. It was also assumed that the efficiency of the gear train was 97 per cent (8).

Since the bevel gear on the pulley delivered power upward, the reaction on the shaft will be acting down as is shown in the end view of Figure 22. The power delivered to the disk will then be

\[ 0.97 \times 0.75 = 0.727 \text{ hp.} \]

By using the driving force of the gear on the pulley, the bending produced by the normal component on the shaft was neglected. Neglecting normal component will have little effect on the design for 14\(^\circ\) involute teeth. The subscripts P, D, L and R refer to the pulley, disk, left support and right support respectively.

**Torque**

\[ \text{Torque} = \frac{63,000 \text{ hp}}{n} \]

where \( \text{hp} = \) horsepower

63000 is a constant

\( n = \) speed in rpm

Torque received and delivered by the pulley:

\[ T_p = \frac{63,000 \times 0.75}{2000} \]

\[ T_p = 23.6 \text{ in-lb} \]
Torque received by the off-centre disk:

\[ T_D = 23.6 \times 0.97 \]
\[ T_D = 22.85 \text{ in-lb} \]

**Centrifugal forces**

\[ C.F. = \frac{W \omega^2 r}{g} \]

where \( W \) = weight of rotating body in lb.

\( \omega \) = speed in radians per seconds

\( r \) = the distance between centre of rotation and centre of gravity, in feet.

\( g = 32.2 \) feet per sec\(^2\)

Centrifugal forces for the pulley and the disk were determined for maximum rotating speed only. At other speeds the forces will be proportional to the square of the speed all other factors being constant.

**Pulley:**

\[ wt = 3.25 \text{ lb.} \]

Assumed speed = 2000 rpm.

\[ = 209 \text{ rad/sec} \]

\[ r = 0.3125\" = 0.026 \text{ ft.} \]

\[ C.F. = \frac{3.25 \times 0.026(209)^2}{32.2} \]

\[ C.F. = 115.5 \text{ lb.} \]
Disk:

wt = 2.07 lb.
Assumed speed = 2000 rpm

= 209 rad/sec.

\[ r = 0.5'' = 0.0416 \text{ ft.} \]

\[ C.F. = \frac{2.07 \times 0.0416 (209)^2}{32.2} \]

C.F. = 116.7 lb.

The difference between the two calculated forces was negligible and it was assumed that for all practical purposes the forces were equal. At lower speeds this difference further decreased proportionally.

**Bending forces**

The bending force produced by a gear was equal to the driving force.

\[ F_P = \frac{T_P}{r_P} = \frac{23.6}{.8125} = 29.3 \text{ lbs.} \]

and

\[ F_D = \frac{T_D}{r_D} = \frac{22.85}{.8125} = 28.15 \text{ lbs.} \]

(Values for \( r_P \) and \( r_D \) are equal for the radius of the gear was used in both cases.)

Let the bending produced by the belt be:

\[ F_{belt} = 1.5 \left( F_1 - F_2 \right) \]

\[ = 1.5 \frac{T_P}{r_P} \]

\[ = 1.5 \frac{23.6}{2.125} = 16.65 \text{ lbs.} \]
The bending produced by the centrifugal forces was
\[ F_P = 115.5 \text{ lb.} \]
\[ F_D = 116.7 \text{ lb.} \]

The maximum bending moment and its location

Two coplanar force systems were used for the convenience of the analytic solution. Clockwise moment was assumed as positive. First the forces on the horizontal plane were considered as shown in Figure 23.

\[ F_{belt} = 16.65 \text{ lb} \]

was the only force acting.

The reactions at the left and right support were:
\[ M_L = 4.25(16.65) - 5(R_x) = 0 \]
\[ R_x = 14.15 \text{ lb.} \]
\[ M_R = -0.75(16.65) + 5(L_x) = 0 \]
\[ L_x = 2.5 \text{ lb.} \]

Since the shear diagram in Figure 23 crossed zero at P the maximum bending moment in the horizontal plane must be at that point. The moment at P was:
\[ M_{P_x} = 2.5(4.25) \]
\[ = 10.65 \text{ in-lb.} \]

The forces on the vertical plane are shown in Figure 24.
\[ P_y = P_p = 29.3 \text{ lb.} \]
\[ D_y = P_D = 28.15 \text{ lb.} \]
Fig. 23. Locating the point of maximum bending moment in horizontal plane.

Fig. 24. Locating the point of maximum bending moment in vertical plane.
In addition to this the following centrifugal forces were also acting in the same plane:

\[ C.F_P = 116.7 \text{ lb.} \]
\[ C.F.P = 115.5 \text{ lb.} \]

The reactions at the left and right support were:

\[ M_L = -0.875(28.15) + 0.9375(116.7) \]
\[ + 4.125(115.5) + 4.25(29.3) \]
\[ - 5(R_y) \]
\[ = 0 \]
\[ R_y = 137.5 \text{ lb.} \]

\[ M_R = -0.75(29.3) - 0.875(115.5) \]
\[ - 4.063(116.7) + 4.125(28.15) \]
\[ + 5(L_y) \]
\[ = 0 \]
\[ L_y = 97.8 \text{ lb.} \]

The shear diagram in Figure 24 crossed zero at P where there will be the maximum moment. The moment was:

\[ M_{Py} = 0.125(114) + 0.75(137.5) \]
\[ M_{Py} = 117.25 \text{ in-lb.} \]

Obviously the maximum moment was at P. Summing up the rectangular components of that moment:

\[ M_p = \left[ (M_{Px})^2 + (M_{Py})^2 \right]^{\frac{1}{2}} \]
\[ = \left[ (10.65)^2 + (117.25)^2 \right]^{\frac{1}{2}} \]
\[ = 117.8 \text{ in-lb.} \]
It had been determined by the foregoing that the maximum bending and maximum torque both occurred at P, which was therefore the critical section.

Using the formula (8):

\[
S_s = \frac{16}{\pi D^3} \left[ (K_t T)^2 + (K_m M)^2 \right]^{\frac{1}{2}}
\]

the size of the shaft required can be figured.

Shearing stress used should be the smaller of the two values:

\[
S_s = 0.3 \text{(yield point)} = 0.3(58000)
\]

\[
S_s = 17,400 \text{ psi}
\]

or

\[
S_s = 0.18 \text{(ultimate tensile strength)} = 0.18(72,000)
\]

\[
S_s = 12,960 \text{ psi}
\]

Therefore, the design stress will be:

\[
S_s = 12,960 \text{ psi}
\]

Values of \(K_t\) and \(K_m\) were obtained from Faires (8).

\[
K_t = 1.0
\]

\[
K_m = 1.0
\]

Substituting the numerical values in the above equation and solving for \(D\):

\[
D^3 = \frac{16}{12,960 \pi} \left[ (23.6)^2 + (117.8)^2 \right]^{\frac{1}{2}}
\]

\[
= 0.1485 \text{ in}^3
\]

\[
D = 0.53 \text{ in.}
\]

\[.
\]

The selected \(5/8\)" shaft was satisfactory for strength.
Power Required

The power supplied by the engine was used only for operating the vibrating unit. The mode of power transmission was the V-belts. The idler pulley through a quarter turn belt arrangement received power from the engine and delivered to the pulley on the vibrator.

The minimum power required to operate the vibrator was estimated theoretically as shown in the following discussion.

The two main resistances acting against the movement of the vibrator were:

1. The shearing resistance of the soil and weeds.
2. The friction due to slide of the vibrator base over soil surface (10).

The shearing resistance of the soil was produced due to the slicing action of the cutting blades. The blades penetrated through the soil and a layer of soil up to one inch was sliced by the lateral vibrations of the blades.

Assuming the following:

1. The vibrating frequency of 1000 cycles per minute giving a maximum deflection of 0.25 inches.
2. The effective cutting width of the blades to be 24 inches.

First the approximate shearing resistance of the soil was estimated.

The following values were assumed for an average truck
growing soil (31).

Cohesion = 1000 pounds per sq. ft.
Angle of internal friction = 15°

Using these values in the following:

Shearing resistance = Cohesion + P tan Ø
where P = the weight of the vibrator
Ø = the angle of internal friction.

Shearing resistance = 1006.15 pounds per sq.ft.

Not knowing the resistance produced due to the cutting of weeds an approximate value of 2000 pounds per sq.ft. was taken as the total shearing resistance.

Next the lateral velocity of the vibrator was calculated.

Maximum linear velocity = Maximum deflection x angular velocity in radians

Assumed deflection = 0.25 inches
Velocity = 131 feet per minute

The shearing area based on one stroke of cut was:

\[ 131 \times \frac{24}{12} \times \frac{1}{2000} = 0.131 \text{ ft}^2 \]

The force required to cut the slice of soil at above rate was:

\[ 0.131 \times 2000 = 262 \text{ pounds.} \]

The friction caused by the sliding of the vibrator base over the soil surface was also one of the chief factors affecting vibrator movement. The magnitude of friction being dependent on several variables (21), the exact value of coefficient of friction could not be determined. An approximate value of
15 per cent of the shearing resistance was taken as the part required to overcome friction (10)(21).

Therefore the total force required per stroke was:

\[ 262 + 39.3 = 301.3 \text{ lbs.} \]

The distance covered in one stroke was 0.25 inches. The torque was:

\[ 301.3 \times \frac{25}{12} = 6.27 \text{ ft. lbs.} \]

The power delivered to the vibrator was:

\[ 12,540 \text{ ft. lbs. per minute.} \]

Assuming an efficiency of power transmission as 75% taking into account friction and slippage at the two belts, the minimum power required to operate the vibrator only would be:

\[ \frac{12,540}{33,000} \times \frac{1}{0.75} = 0.507 \text{ hp.} \]
PERFORMANCE

A trial run of the machine was made in the laboratory to check its operation. The main difficulty was with the turning and directing of the entire assembly in any direction for the frame being only 1 1/2 inches above ground the unit was partially sliding on the ground.

The frame was later raised by 3 1/2 inches thus giving a clearance of 5 inches above ground surface.

A second test run was made on July 3rd 1950, on the Agricultural Engineering farm south of Ames. It was discovered during the field trial that the vibrator did not have as vigorous vibrations as the first trial unit used before this machine. This was the greatest difficulty encountered and hence the machine did not operate as expected showing a lack of powerful vibrations which would enable the forward motion of the machine. This trouble was partly due to the wet ground on account of a heavy rain.

Due to lack of time a machine with stronger vibration has not been constructed. However, this unique idea of using vibration in cultivation and weeding seemed to have possibilities. Further suggestions are covered in scope for study section.
Fig. 25. Front view of cultivator.

Fig. 26. Rear view of cultivator.
Fig. 27. Two views of the cultivator.
SUMMARY

The studies of various implements used for truck crop cultivation led to the design and construction of a machine to cultivate satisfactorily in the weed infested vegetable farms.

The development of vibrating mechanism for cutting weeds was discussed with the detail description of the device used for producing only the desired lateral vibrations.

The frequency of vibrations and the maximum lateral velocity of the cutting blade were determined. The critical speed of the rotating pulley and disk producing centrifugal forces was also calculated along with the maximum stresses developed in the vibrating mechanism shaft justifying the use of the shaft of a particular size.

The framework and the method of power transmission were also described indicating engine sliding device used in place of clutch to transmit and cut off power to the vibrating unit as desired.
SUGGESTIONS FOR FURTHER STUDY

1. Construction of a sturdier two-row, high-clearance, self-propelled machine with a single cutting blade of smaller width and centrifugal forces of a greater magnitude giving more vigorous vibrations.

2. A refinement in the design and experimental determination of the slope of the main shell of the vibrating unit, that might give minimum resistance to soil shear and also throw laterally all the rising soil on the slope.

3. Experimental trials of the machine in the vegetable farms of tomatoes, egg-plants, water-melons, etc.
LITERATURE CITED


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