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Design and development of an experimental corn plot harvester

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DESIGN AND DEVELOPMENT OF AN
EXPERIMENTAL CORN PLOT HARVESTER

by

Merlyn Duane Bass

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

Maize, or corn as it is commonly called, is a cereal plant indigenous to the Western Hemisphere. World corn production for 1966 was 252,750,498 acres with an average yield of 37.19 bushels per acre (24). Corn production for grain in the United States during 1968 was 55,886,000 acres with an average yield of 79.4 bushels per acre (25). Much of the yield advantage we enjoy in this country today can be attributed to superior production tools and techniques developed by corn researchers. This requires extensive use of experimental plots as indicators of superior yield potential.

The harvesting of experimental corn plots is a time-consuming and tedious task for Agronomists and Agricultural Engineers. As shown in Figure 1, the primary method of harvesting these plots has been by hand. The principle limitations of this method of harvesting are:

1. High time requirement per plot
2. High cost
3. High seasonal labor requirements
4. Measurable differences in harvesting by different individuals
5. Measurable differences in harvesting by an individual during different periods of the day and from day to day.
Figure 1. Many experimental corn plots are still harvested completely by hand
Due to the limitations cited, it has been increasingly desirable to mechanize the harvesting of such plots. If efficiently and accurately accomplished, mechanized harvesting could have the following advantages:

1. Low time requirement per plot
2. Lower costs
3. Low labor requirement
4. Less variation in % of gross yield harvested
5. Machine-harvested yield potentials, more realistic for application directly to the farm situation
6. More replication of an experiment or larger plots giving a higher degree of confidence in the results
7. More plots or experiments can be performed, thus increasing research capacity and productivity.
LITERATURE REVIEW

Corn Plot Harvesting Machinery

Some early attempts were made at mechanizing plot work. In 1955, Hansen and Robertson (12) built a one row picker on a tricycle-type carrier, Figure 2. The unit was able to operate satisfactorily in 36 inch rows. Figure 3 shows that the snapped ears were collected in a hopper behind the unit which in turn was elevated to the husking device on the truck. Here, the sample was weighed, husked, and sampled for moisture.

In 1962, Hunter and Kjelgaard (15) mounted a one row picker on a chassis which was capable of harvesting 38 inch rows, Figure 4. This unit incorporated weighing, moisture sampling and bagging operations on a trailer behind the picking unit. Approximately 500 pounds of bagged corn could be carried on the platform at a time.

Commercial one and two row pickers have also been used as plot harvesters with some success, Figure 5. In 1968 Leme and Ometto (18) of Brazil developed a new one row mounted picker which picks, shells, and bags corn in one operation, Figure 6. As seen in Figure 7, the unit uses a helicoidal conveyor to gather the crop over the two hexagonal snapping rolls and to convey the ears into the sheller. The shelling is accomplished by a steel spiral sheller rubbing against a circular steel concave. The shelled corn is then conveyed by
Figure 2. Self-propelled snapper for plot work by Hansen and Robertson

Figure 3. Snapped ears were delivered to a bin, elevated hydraulically, and dumped into a truck-mounted husker for weighing
Figure 4. Hunter's one row self-propelled plot picker with bagging and weighing as an integral operation

Figure 5. Two row picker used as a plot harvester. Shelling and weighing are a separate operation
Figure 6. Leme and Ometto corn harvester uses an auger for gathering and conveying

Figure 7. Rear of the machine with bagger, discharge chute for husks, and drives for the auger and sheller
a rotary throwing device to a bagging platform where it is cleaned by a fan blast and then sacked.

In the late 1950's, the combine began to be a popular method of harvesting corn. Johnson and Lamp (17) listed the factors which influenced the rapid trend toward field-shelling as increasing farm size, commonality of equipment with soybeans, government programs, storage facilities, labor availability, and drying technology. In 1968, Indiana, Illinois, Minnesota, and Iowa farmers combined 58.8, 57.0, 40.3, and 34.6 percent of their corn crops respectively. In 1969, 61.5, 62.0, 42.7 and 40.5 percent was combined respectively (16).

Following the trends toward shelled corn and narrow rows, which were a result of the use of the combine to field-shell the crop, researchers were faced with a void of plot harvesting machinery for narrow rows. Pioneer Seed Corn Company\(^1\) and others began to use modified commercial combines for corn research work, Figures 8 and 9. Such machines were capable of operating in narrow row research plots, although they are not well suited for plot work.

Figure 8. Commercial combine modified for corn research work

Figure 9. "Cab" houses operator and equipment for plot yield determinations
OBJECTIVES

In June, 1968, a cooperative agreement between the Crops and Agricultural Engineering Research Divisions of the United States Department of Agriculture and the Agriculture and Home Economics Experiment Station of Iowa State University was signed. The objective of the research agreement was to develop "a corn harvester that will harvest efficiently small corn plots that are used in conducting basic research in corn breeding and quantitative genetic studies in Iowa.

1. Specifically, to develop a harvester that will operate efficiently at slow speeds through the corn plots, but have a relatively fast rate of self cleaning in preparation to harvesting succeeding plots within the experimental fields.

2. Investigate and develop a head on the harvester that will be adaptable for harvesting plots of varying row widths.

3. Develop a harvester that includes the feature for depositing dropped ears that have been gleaned from the experimental plots. The feature is to be designed so that the ears can be included in the harvested plot, but not hazardous to the individual depositing the ears in the harvester.

4. Develop a harvester that will harvest ear corn, shell, and deposit grain in a bin for determining plot yields.
and for obtaining a sample for making grain moisture determinations.

5. In the development of the harvester, the machine will be sufficiently mobile to move from fields in close proximity but capable of being transportable to fields some distance apart."

Additional Design Parameters

Using the project objectives as a guide, Iowa State University corn researchers were contacted for more specific information about their problems and needs. These discussions resulted in the following information:

1. Generally, four row plots are planted with the center two rows harvested for data while the remaining two eliminate border effects.

2. Row spacings vary considerably, with some 20 inch rows, but the majority being 30 and 40 inch spacings.

3. Many of the plots are planted on farms throughout the state by the farm operator using his own equipment and cultural practices. Thus, the machine needs to be quickly adjustable for any normal row spacing.

4. Normal plot lengths are 9 feet to 30 feet.

5. Maximum yield is 200 bushels per acre.

6. Normal moisture content at harvest is 15 to 25%.

7. Approximately 14,000 plots per year are harvested.
8. A 40-inch strip separates the ends of adjacent plots, with a series of such strips and plots grouped together. Roadways for turning are permissible but the size and frequency should be minimized.

9. Harvested material is used for estimating relative yield potentials. Damage need not be a major consideration since the material will not be used for breeding purposes.

10. Operation and maintenance should be simple and safe, requiring no more than normal mechanical ability.
EXPERIMENTAL CORN PLOT HARVESTER

Conceptual Design

One of the limitations of hand-harvested experimental corn plots is that this method may give a false indication of machine-harvestable yield potential. Because of this, it was felt that the plot harvester should process the crop in a manner similar to normal corn harvesting machinery.

The first efforts toward developing a plot harvester were concentrated on commercial machines. It was hoped that a suitable production machine could be located or modified for plot work. Commercial one-row pickers, mounted or pull-type, were eliminated since these machines would be unable to harvest corn from an experimental plot without disturbing adjacent rows. Hansen and Robertson (12) and Hunter and Kjelgaard (15) had designed and built one-row plot harvesters that could harvest a single row without disturbing adjacent rows. These machines were only suitable for row spacings of 36 inches or more. It was felt that such a machine with a wheel tread for narrow rows would be too unstable for safe operation.

Two-row pickers have previously been used for plot harvesting. Mounted pickers are highly maneuverable and can be adapted for shelling. The primary difficulty with such units is that the tractor-picker combination can harvest only
wide row spacings. Some narrow-row pull-type pickers are available. These units cannot open fields or plots without disturbing adjacent corn rows nor are they very maneuverable. Pickers, in general, are unsuitable plot harvesters.

Two of the most common mechanisms for field-shelling corn are the cage sheller and the combine cylinder. A survey of production machines revealed that cage shellers are generally smaller in physical size than combine mechanisms. The difference is primarily in the large separating area required behind a cylinder as compared to the cage sheller which does its separating through the cage surrounding the rotor. Also, cage shellers are commonly designed for use on the rear of a tractor-mounted picker which places compactness among the original design parameters. Shelling is accomplished by the agitation of the corn inside the cage by the rotor. This agitation causes the ears to rub against each other as well as the rotor and cage, thus rubbing the grains from the cob. Most cage shellers require that the snapped ears of corn be husked before shelling, as large quantities of trash are detrimental to separation of the shelled corn from the cobs. For satisfactory operation, the cage must be partially filled with corn at all times. Thus, cage shellers would probably not be well suited for processing snapped corn nor give the required cleanout characteristics desired in a plot machine.
Combine cylinders when equipped with grate-type concaves generally give good cleanout of the material after it has reached the cylinder. However, many machines have cylinders which are fed by undershot conveyors, generally consisting of chain and steel paddles. These conveyors are not self-cleaning, as can be verified by examination of production combines. Material often collects at the rear of the conveyor, between it and the cylinder. Some machines also have ledges and crevices where shelled corn can collect above the cylinder, to the rear of the cylinder, and in the separating area. The majority of the combines surveyed were of large physical size. These units would be difficult to load for transport and would require special permits for truck transport on public highways. Most of these machines have corn heads which feature some adjustment for different row spacings. However, this adjustment is limited and often requires a considerable amount of time and physical manipulation. In general, two-row heads for narrow rows are commercially unavailable. A commercial combine could be modified for use as a plot harvester but would probably not be any better suited for the purpose than modified machines already in use.

The conclusion drawn from the survey was that no complete production machine would satisfy all or even most of the project objectives. However, due to the complexity of the problem and the time element, it was deemed desirable to use
production components of proven design and reliability if compatible with the design parameters.

A Ford "601" sheller unit (9) was selected as the primary processing unit for the plot harvester, Figure 10. This unit was originally designed for use with a mounted corn picker and was located on the rear of the tractor. The unit was intended to process corn from a one-row picker. Although most manufacturers use a cage sheller for such applications, this unit has a 16 inch diameter rasp-bar cylinder for shelling.

The sheller has all the major components found in large combines; a grate-type concave, a cleaning fan, a unit rack, and a unit sieve. The primary difference is the overall size, especially the length. Most of the saving in length comes from the short separating area to the rear of the cylinder and concave. In conventional service, most of the husks and trash would normally be eliminated by the husking bed prior to shelling, thus only light trash and chaff would be delivered to the rack and sieve. The heavy trash throughput of a normal combine necessitates the large separating area found in these machines. The actual rack area in this Ford "601" machine is 1035 square inches while the sieve area is 650 square inches. This contrasts with large commercial combines with sieve and rack areas of several thousand square inches each. Although the body width is only 18 inches, it was felt that the unit would have sufficient capacity in plots for two
Figure 10. Ford "601" sheller used in the plot harvester. Engine mount, support columns, and additional reinforcements carry loads imposed by the engine and grain tanks.
rows, since the plots are short, ground speed is very limited, and the harvesting is only intermittent. Loss in separating efficiency from the smaller unit was felt to be negligible since little time was available in each plot for an overload to develop on the rack or sieve. This would probably not be true for steady field harvesting where a constant flow of material and steady forward travel might easily cause excess separating losses due to overloading of the rack and sieve.

This particular shelling unit has several important advantages over others surveyed for this application. The cylinder is gravity fed, thus not leaving the material residue between the end of the conveyor and the cylinder as in other machines. This gives excellent cleanout of material delivered to the cylinder. The inside of the machine is relatively free of crevices and ledges which trap shelled corn, as were commonly present on some machines. The concave, rack, and sieve are relatively self-cleaning, but the collection system for clean grain is not. However, this portion is easy to remove for installation of a self-cleaning collection system. The number of machine adjustments is kept to a minimum; only one cylinder-to-concave clearance adjustment is required, the cylinder input speed is a constant 670 revolutions per minute, and the sieve is adjusted easily. This is an important factor to consider when those operating the machine may have no more than normal mechanical abilities. The unit is simple, compact,
and could be easily adapted for processing corn plots.

Having located a suitable shelling device for the plot harvester, conceptual work was undertaken on a chassis with a provision for harvesting variable row spacings. It was felt desirable to use some type of conventional gathering device of proven design for reasons previously stated. The result of this work was two alternative designs.

The first of these was a four-wheel chassis with a fixed wheel tread compatible with the narrowest row spacing to be harvested. The gathering units would be adjusted hydraulically on a telescoping framework.

The second design also would use gathering units on a telescoping framework. The right-hand drive wheel of the machine would be supported by a telescoping axle which could be varied hydraulically with the row spacing. This right wheel frame would also serve as the support for the telescoping framework of the right row unit; the right row unit and wheel would move simultaneously while the remainder of the machine would maintain a constant physical relationship.

Row units from several production corn harvesters were reviewed with the two alternative designs in mind. Several units appeared to have desirable design features and physical dimensions suitable for 20 inch row spacings.

Two John Deere "40" series row units were finally selected for the plot harvester. The simplified mounting
and drive arrangement were the primary reasons for selecting these units over other designs. The units clamp securely to a single rectangular tube, Figure 11. Power is transferred to each row unit by a slip-clutch which "floats" on the 1 1/8 inch hexagonal drive shaft. The slip-clutch assembly drives an enclosed gear case which powers both the snapping rolls and gathering chains. The snapping units have self-tightening, spring-loaded gathering chains. The low profile of the units and the cantilevered snapping rolls, Figure 12, should give excellent performance, especially in lodged corn. The bearings which support each snapping roll run in oil inside the gear case and there is no bearing near the lower end of the roll. The wrapping of weeds and grass around the bottom bearing as well as contamination and wear of this bearing due to running in the soil are minimized by the cantilevered roll design. Smoother flow of material through the rolls results since the "dead" spots where the bearing hangers of previous designs were located have been eliminated. These design features have reduced the overall physical size of the units which makes them very desirable for use with the plot harvester.

The ground drive for the plot harvester presented some formidable design problems. Either of the two possible chassis designs previously mentioned, a fixed wheel tread or a variable wheel tread, presented much the same problems. It was desirable to have a variable speed drive so that the actual
John Deere snapping units are driven by a single hexagonal shaft and clamp securely to the rectangular tubing frame.

Figure 12. Snapping units have a low profile. Features include spring-loaded gathering chains, adjustable stripper plates, and cantilevered snapping rolls.
plot harvesting could be accomplished at a suitable low speed but a higher speed could be used for transport between plots or fields. Thus, a gear-type transmission would require a minimum of two forward speeds and a reverse. However, a single operating speed would be very restrictive since it would have to be matched to the heaviest crop conditions expected to prevent overloading of the harvesting mechanism. With such a system, a larger engine might be required since the engine horsepower would have to be sufficient for extreme ground conditions of mud or snow. A transmission with two speeds or more in the normal operating range with a transport and reverse would be much more desirable.

Complete transmission-differential units are generally not available, but are custom designed for specific applications using stock components. Since the gear-type drive is not well suited for the intended purpose, the fabrication of a suitable transmission is almost beyond the capability of available shop facilities, and the expense of a single custom-built unit is prohibitive; this type drive was quickly eliminated from consideration for the plot harvester.

A second transmission considered for the plot harvester was a belt drive with variable-pitch drive sheaves. Such drives are commonly used on agricultural machines where more speed variation is desired than is easily attained with a gear drive. Further investigation revealed that a variable-speed
belt drive is generally combined with a conventional transmission-differential assembly to give good speed control in the field operating range and a higher transport speed, especially if substantial amounts of power must be transferred to the ground drive system. If only the variable-pitch drive arrangement is used, the range of speeds attainable is limited primarily by the sheave diameters. Such units are available as "shelf" items, although the selection is somewhat limited. Also, some provision for reverse and a differential must be provided for use with such a drive. For these reasons, this driving arrangement was also eliminated.

Hydrostatic drives are gaining favor in agricultural equipment. Most of these systems use a variable-displacement engine-driven pump to drive a fixed displacement motor connected to the input shaft of a transmission-differential assembly (22). This has been necessary because of the low efficiency of hydraulic motors when operated with a direct wheel connection and the extreme size of the motor required to generate sufficient axle torque at these low speeds. However, hydrostatic drives have obvious advantages in plot work. First, the number of engagement-disengagement cycles per harvest season with associated heat generation problems had severely shortened tractor clutch life in plot work.²

Therefore, since the hydrostatic pump eliminates the clutch, it was felt that maintenance alone could nearly justify the hydrostatic drive over more conventional drives. Hydrostatic drives provide nearly infinite speed control, an obvious advantage in the variable conditions found in plot work. Nearly instantaneous direction changes allow operation in muddy conditions where other drive types become stuck more easily. Design is somewhat simplified since the only connection between pump and motor are high pressure oil lines; thus the power source and pump may be remotely located from the motor if desired.

A Borg-Warner #8 Power Wheel (5) with a Borg-Warner gear-type motor (4) was tentatively selected as the propulsion unit for the plot harvester. The primary advantage of these units is that the transmission formerly required as a separate unit with hydrostatic drives has been built into each wheel unit in the form of a dual stage planetary gear reduction. The entire drive train is inside the wheel, in space formerly wasted around the wheel hub. Because of this dual stage planetary reduction, a small hydraulic motor can be used as a power source because of the 30.89 to 1.0 torque magnification through the planetary arrangement to the wheel. At the same time, because of the subsequent reduction in speed from the hydraulic motor to the wheel, the hydraulic motor can be operated at speeds where the motor efficiency is fairly high.
and still give the desired machine operating speeds. Both wheel motors can be powered from a single pump of suitable capacity. If the wheel motors are connected in parallel, the flow will be divided giving an effective "hydraulic differential" for turning. Also, since each wheel is physically independent, a variable wheel tread appears to be feasible since only hydraulic lines need be extended or retracted when the tread is varied.

Design and Construction

Having located some suitable components for part of the plot harvester, the physical limitations of the problem were applied along with objectives and design parameters.

Estimates of the weight of the total machine and the weight carried by the drive wheels were considered in selecting the drive wheel size. While it was desirable to limit the section width for clearance in narrow rows, a 12.4"-24" drive tire was selected because of its load-carrying capacity, the extra floatation obtained by using the wider section, and because it is a tire size often stocked by tire distributors.

Table 1 shows the minimum and maximum center to center wheel spacing for various row spacings. These values were determined using 12 inches as the approximate tire width and allowing the tire to come no closer than 5 inches to the adjacent rows of standing corn or to the two rows being harvested.
Table 1. Minimum and maximum center to center wheel spacing for 12 inch wide tire while maintaining 5 inch row clearance for various row spacings

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<tr>
<th>Row spacing inches</th>
<th>Minimum wheel tread inches</th>
<th>Maximum wheel tread inches</th>
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<tr>
<td>20</td>
<td>38 (3 inches from harvested rows)</td>
<td>38</td>
</tr>
<tr>
<td>22</td>
<td>44</td>
<td>44</td>
</tr>
<tr>
<td>24</td>
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</tr>
<tr>
<td>40</td>
<td>62</td>
<td>98</td>
</tr>
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</table>

If the design with simultaneous movement of the right row unit and right wheel is used, no possible combination of acceptable wheel tread exists for the 20 inch movement from 20 to 40 inch rows. One possible combination exists for the 18 inch movement from 22 to 40 inch rows. Several combinations are available for the remainder, although it appears highly desirable to use the maximum wheel tread possible to insure machine stability in transport or side-hill operation.
If a fixed wheel design is used, the wheel tread must be 38 inches for harvesting 20 inch rows. If 22 inch rows are the minimum to be harvested, a 44 inch wheel tread is required.

After a review of the estimated machine size with the projected weight distribution, row spacings less than 24 inches were eliminated, regardless of which design was used. It was felt that a two-row machine using the proposed components would be too unstable on the narrow wheel tread required for either 20 or 22 inch row spacings. Further layout work revealed that even the 26 inch row spacing could be used only with the sacrifice of some "mud clearance" between the tire and frame, even using the maximum possible wheel tread. The only possible method of achieving a narrower wheel tread would be the relocation of the hydrostatic pump from the axle-mounted position to a position adjacent to the engine on top of the sheller unit. Such relocation could be done only at a sacrifice in drive simplicity, loss of space, and loss of machine stability due to the concentration of heavy components at a relatively high position on the machine.

The design using simultaneous movement of the right row unit and right wheel by hydraulic cylinders was selected as the better design. Using this design, the plot harvester is capable of harvesting row spacings from 26 to 40 inches. The maximum wheel tread possible with 26 inch rows, 56 inches, was used as a base with the 14 inches of movement giving a
maximum wheel tread of 70 inches. When the row units are positioned for harvesting 30 inch rows, the machine is nearly symmetrical about its centerline.

A 4" x 4" x 0.250" wall section of structural square tubing was selected as the telescoping member of the drive axle, using the assumed loading of the member for selection. The interior dimensions were suitable for installation of the Prince PMC 3314 2 inch bore, 14 inch stroke double-acting cylinder. This cylinder operates at 1625 psi to supply 5100 pounds of force to extend the axle with the full machine weight on the drive wheels. The stationary section of the axle is a 5" x 5" x 0.250" section of structural square tubing. This size was chosen since it is the next larger size of tubing into which the 4" x 4" telescoping section would slide. The smaller tube was then built up by welding 8 0.250" thick "rails" to it, 2 on each side of the tube. This fabricated section was milled for a sliding fit inside the 5" x 5" tube, Figure 13.

At this time it would be well to note that most of the strength calculations on frame members and power transmission components were based on rough estimates of the loading situations. Since little time was available for redesign caused by mechanical failure, most components have a considerable margin of safety. It was felt that this extra material expense could be justified on this machine since a high degree
of reliability is desirable. Serious failures of the harvesting equipment could jeopardize research plots worth many times the value of the added expense of an "overdesigned" unit versus one of marginal mechanical strength.

The wheel support masts for both wheels are welded fabrications of 0.50" and 0.25" mild steel plate. The heavier material is used where the Borg-Warner wheel units bolt to the mast. The lighter material is welded into a rigid box section with the heavier material to serve three primary functions. First, the mast serves as the attaching point for the wheel units. The top of each wheel mast contains a 1 1/4" pivot pin for the row unit framework. The bottom of each mast supports the bottom pivot pin of the hydraulic cylinders which raise the row units. The left mast also serves as a support for the auxiliary hydraulic pump and the clutch which controls the row units.

The row unit support framework consists of a 4" x 3" x 0.250" wall rectangular tube which is mounted on the lift arms at a 25 degree angle with respect to the theoretical ground line. A 3" x 2" x 0.250" tube with "rails" much like those of the telescoping axle is plug-welded into the left hand half of the 4" x 3" tube. The right half of the fabricated section is milled for a sliding fit inside the right half of the 4" x 3" tube. The individual row units are clamped securely around the 4" x 3" tube, each on its respective half of the
A 1.25 inch bore, 14 inch stroke double-acting cylinder is pinned to attaching points on the two row unit support tubes, Figure 14. This cylinder is connected in parallel with the one located inside the telescoping axle. The 2000 pounds of force generated by this cylinder plus the 5100 pounds of force from the axle cylinder give a total force of 7100 pounds available to overcome the metal-to-metal friction on the two telescoping sections and the tire-to-soil friction force. By pressurizing these two cylinders, the plot harvester head can be adjusted from 26 to 40 inch rows while the wheel tread is adjusted simultaneously from 56 to 70 inches. Sufficient force is available to extend the wheel and row unit while the machine is stationary but it is more desirable to move the machine while making the adjustment, primarily to ease the side-load on the tire and rim.

The frame under the machine is fabricated from 5" x 3" x 0.250" rectangular tubing. The two side members of the frame are butt-welded to the 5" x 5" tube which serves as the stationary section of the telescoping axle. At the rear of the 85 inch long frame, the tubing is mitered and joined by a cross member of the same material. This whole framework must be welded "oil-tight" since it doubles as hydraulic reservoir and oil cooler.

A steering axle from a production combine was cut down to give a 60 inch rear tread. It was felt that the wheel
Figure 13. Telescoping axle with an internal hydraulic cylinder

Figure 14. Telescoping row unit support frame with hydraulic cylinder
spindles and steering linkages would be difficult to fabricate so the complete axle including the tie rods and power-steering cylinder was modified for the plot harvester. Figures 15 and 16 show the underslung frame and the oscillating rear axle assembly. The rear axle support doubles as the rear attachment for the sheller assembly. A hitch which may be used to tow a trailer or as an anchor point for chains when the harvester is transported by truck is incorporated into the bracing designed to absorb any thrust loads imposed on the rear axle. The underslung frame gives approximately 10 inches of ground clearance.

Before further calculations on total reservoir requirements for the hydrostatic system could be completed, the pump size and flow rate had to be established. The only quantities really well defined were the tire size with a 20.8 inch rolling radius and the desired speed range of 0 to 10 miles per hour. From Borg-Warner Model 8 Power Wheel (5) performance data, the input speed required to achieve various ground speeds is plotted with the tire rolling radius. From Figure 17, an input speed of approximately 2400 revolutions per minute is required for an output of 10 miles per hour from the wheel unit. Figure 18, from Borg-Warner M25-22 hydraulic gear-type motor (4), indicates an input flow of 28 gallons per minute to each motor or a total flow of 56 gallons per minute to the two wheel motors is required. Sundstrand (23) engineering
Figure 15. Rear view showing axle and underslung frame. A hitch for wagons or an anchor point for chains is incorporated into the axle support bracing.

Figure 16. Complete steering axle is from a commercial combine.
Figure 17. Ground speed as a function of tire rolling radius and input speed for Borg-Warner Model 8 Power Wheel (5)

Figure 18. Input flow required for various output speeds for Borg-Warner M25-22 hydraulic motor (4)
personnel recommended a 22 series pump for this installation. Use of a Sundstrand 3000 psi relief valve manifold with the pump gives performance as indicated in Figure 19. Even by driving the pump at its maximum speed of 3000 revolutions per minute, a flow of only 50 gallons per minute could be obtained from the pump.

Alternative solutions were to use the 22 series pump and settle for a lower speed or select a larger pump such as the 23 series pump, which has a capacity in excess of that required. After a review of the problem, the increase in size of the 23 series pump was determined to be more detrimental than a slight loss of machine transport speed by using the 22 series pump.

The output stall torque of the hydraulic motor at 3000 psi is 1100 inch-pounds. Assuming 100% mechanical efficiency through the wheel unit, this would be transmitted to the wheel as approximately 33,000 inch-pounds of axle torque. From Figure 20, if the machine were operating at 2 miles per hour (500 revolutions per minute) in extreme tractive conditions requiring nearly the full 3000 psi system pressure, the power output per wheel would be approximately 12 horsepower. This results in approximately 2250 pounds of rim pull with the 20.8" rolling radius of the wheel. Thus the plot harvester would have a total tractive effort of 4500 pounds, which was considered adequate.

Since performance curves for both pump and motor at the
Figure 19. Pump output as a function of pump input speed at an 18 degree swashplate angle for Sundstrand 22 series pump (23).

Figure 20. Motor horsepower output as a function of motor speed for Borg-Warner M25-22 hydraulic motor (4).
low flow rates required for operation at 2 miles per hour were not available, the overall efficiency of the pump was estimated at 80 per cent and the gear motor at 60 per cent, resulting in an overall system efficiency of 48 per cent. Thus, about 50 horsepower could be required under extreme ground conditions for propulsion alone. However, assuming such extreme conditions would be encountered only infrequently, only 25 horsepower was allotted for continuous vehicle propulsion.

The Ford "601" sheller with the additional power required for operation of extra material handling equipment was expected to consume about 18 horsepower.

A Cessna (6) B15511 gear pump operating at 1625 psi was selected to furnish hydraulic power for machine control. A Char-Lynn V-66 flow divider split the pump output into two separate portions. The flow divider furnishes 3.5 gallons per minute at 1000 psi as a priority flow for steering with the remainder for other functions. By driving the pump at 2500 revolutions per minute, a flow rate of 12 gallons per minute would be available with 3.5 gallons per minute already allotted to the power-steering circuit. The remaining 8.5 gallons per minute was deemed adequate for the projected demand of the other hydraulic components of the machine. From pump performance data, about 12 horsepower would be required from the engine for powering this pump.
The John Deere "40" series row unit has a static slip-clutch torque setting of about 240 foot-pounds. However, under normal operating conditions, the input horsepower for each unit is estimated to be less than 5 horsepower.

By summing the horsepower estimates for each of the plot harvester components, the total horsepower required was estimated at 65 horsepower. Since the ground drive should not require the full 25 horsepower allotted to it continuously, it was felt that an engine with a maximum of 65 horsepower could be safely used for continuous operation without being overloaded.

A Wisconsin V-465D (27) air-cooled industrial engine was selected as a power source. It was felt that this air-cooled engine had important advantages over others considered. Physical size and weight were probably two of the most significant advantages noted. The overall length is only 29 inches and the engine weighs 530 pounds. Since the engine is air-cooled, several pieces of heat exchanging equipment used with water-cooled engines are eliminated as well as the problems of servicing these items. The 177 cubic inch displacement engine can produce a maximum of 65.9 horsepower at 3000 revolutions per minute from its 4 cylinders. The crankshaft has tapered roller main bearings at each end and a roller center main bearing. This design permits direct-coupling of heavy side loads to the crankshaft extension without using outboard
bearings. The valves have rotators which extend valve life considerably, especially as compared to air-cooled engines without this feature.

Figure 21 shows the performance curves for the engine. After a review of power requirements, it was decided to run the engine at 2600 revolutions per minute rather than 3000. The engine will develop 63 horsepower and 1510 inch-pounds of torque at this speed. Only an additional 2.9 horsepower can be gained by the extra 400 revolutions per minute. It was felt that extra engine life would probably more than offset the slight drop in power at the lower speed. The engine has good torque characteristics as the speed drops under heavy load. Maximum engine torque is 1660 inch-pounds and occurs at 1800 revolutions per minute.

The engine has a 12-volt distributor ignition system, a 12-volt starter, mechanical fuel pump, anti-dieseling solenoid valve on the carburetor, and a high-temperature safety switch as standard equipment. A 35 ampere alternator instead of the normal 17 ampere generator was ordered, in case lighting or other electrical equipment should be desired at a later date. A dry-element air cleaner with a self-dumping dust valve was ordered instead of the oil-bath air cleaner. Finally, an oil pressure safety switch was ordered as special equipment. Operating much like the high-temperature safety switch, this switch grounds out the engine ignition in case the oil pressure should fall below a minimum level.
Figure 21. Engine performance curves for Wisconsin V-465 D air-cooled engine (27)
Having established the governed speed of the engine at 2600 revolutions per minute, possible drive arrangements for the hydrostatic pump were reviewed. The final decision was to drive the pump at engine speed. The curve of pump output versus input speed, Figure 19, shows that a maximum output of 47 gallons per minute is available. Figure 18 indicates that such a flow when divided equally between the two wheel motors gives a motor speed of 2000 revolutions per minute. Figure 17 indicates that the maximum machine transport speed from the Borg-Warner Power Wheel, Figure 22, would be 8.5 miles per hour; still quite acceptable for the plot harvester.

With the Sundstrand pump operating 2600 revolutions per minute, the charge pump flow rate at the maximum pump output is 0.75 cubic inch per revolution or, 1950 cubic inches per minute. This converts to 8.45 gallons per minute which is the amount of oil taken from the reservoir to make up internal leakage in the main circuit, to provide a flow of oil for cooling purposes, to maintain positive pressure on the inlet port to the main pump, and to provide sufficient oil for control purposes. As noted previously, the Cessna pump used for auxiliary hydraulic functions has a maximum flow of approximately 12 gallons per minute. Thus, the total withdrawal rate from the reservoir is about 20 gallons per minute.

The machine frame has a volume of 2100 cubic inches or capacity for about 9.1 gallons of oil. Sundstrand (23) and
Vickers (22) both recommend reservoirs up to 2 or 3 times the pumping rate per minute, if possible. Sundstrand recommends 0.5 times the pumping rate per minute as the absolute minimum reservoir capacity. Using this recommendation, a minimum reservoir capacity of 10 gallons is required, but any additional capacity that can be gained will be beneficial.

Sundstrand engineers recommend installation of a 10 micron filter between the reservoir and the intake port of the charge pump. It is also recommended that a strainer with a 100 mesh screen be used on the inlet to the intake line. In an effort to satisfy these two requirements and gain additional oil capacity at the same time, a rectangular tank was designed for installation between the two main frame members. This tank is located directly to the rear of the Sundstrand pump and under the fan screen of the sheller unit, Figure 23. The tank is fabricated from 0.188" mild steel plate and is welded directly to the machine frame. The tank has a volume of approximately 1850 cubic inches or 8 gallons of oil. Thus, the total machine reservoir capacity is about 17 gallons. This reservoir volume will provide for a minimum of 50 seconds fluid dwell for removing entrained air from the oil at the maximum system flow rate.

This tank serves several functions besides providing extra fluid capacity. The tank is divided into two distinct sections by a welded baffle. The right portion of the tank
Figure 22. Hydraulically driven Borg-Warner power wheel is a dual stage planetary reduction unit.

Figure 23. Pump, filters, and reservoir for the hydrostatic ground drive of the plot harvester.
receives "hot" oil from the hydrostatic system cooling circuit and from the return side of the auxiliary hydraulic system. This oil flows from the tank into the machine frame through 4 holes bored into the inside of the rectangular tubing, Figure 24. The oil then circulates through the complete machine frame dissipating heat from the oil via the large surface area of the machine frame. The "cool" oil flows through similar holes into the left portion of the tank. Assuming equal flow rates through each of the 1" diameter holes, the theoretical velocity through the holes is 2 feet per second. The maximum recommended velocity for suction piping is 4 feet per second.

The oil is drawn from the tank through a pair of Char-Lynn F-240 strainers, Figure 25, connected in parallel to handle the flow at an acceptable velocity. The strainers are located inside the tank and are serviced through the port on top the tank. Each of these strainers is equipped with an 80 mesh screen to remove large contaminants from the oil. The oil then passes through a Char-Lynn IF-30 dual element filter with 10 micron filter elements. The condition of the elements is monitored by a clear plastic indicator. Should the elements become clogged from neglected service, a bypass valve allows oil to flow unrestricted to the pumps, preventing cavitation at the pump inlets. The filtered oil is then divided into two streams, one going to the intake side
Figure 24. Holes which allow oil to flow from reservoir into the machine frame and the baffle which divides the reservoir

Figure 25. Two strainers on the intake side of the pumps shown before the cover was welded onto the reservoir
of the Sundstrand charge pump and the other to the Cessna pump.

The tank is equipped with a spring-loaded flip-top filler cap which has a micro-filtered air vent. A sight glass on the right end is provided for checking the oil level. This should help eliminate contaminants from the system since the oil level can be checked without "opening" the system to the atmosphere as with a dipstick. Magnetic drain plugs are used in both sections of the tank and at the rear of the machine frame for draining the reservoir. These plugs should help eliminate metallic particles from the complete hydraulic system.

As noted previously, flow from the Cessna pump is divided by a Char-Lynn V-66 flow divider valve. This valve provides a pressure compensated flow of 3.5 gallons per minute for the power steering circuit and is protected by an integral relief valve set at 1000 pounds per square inch. This flow is then directed to the steering valve unit. The power steering control pump, located in the top of the steering column, is powered by the machine operator as he turns the steering wheel. The flow from the control pump opens the appropriate valves in the steering valve unit. The regulated flow of oil from the flow divider is then directed to the proper line of the power steering cylinder, turning the machine in the direction dictated by the movement of the steering wheel.
The remainder of the oil from the flow divider is delivered to a Cessna model 30000 stack type directional control valve with five working sections (7). This valve is located under the front portion of the plot harvester's control console, Figure 26. The open center, parallel circuit valve is composed of two four way-three position sections, three three-way three position sections, an inlet-outlet port plate with integral system relief valve, and an end plate. Each of the working sections has a spring centered spool and a lock-out. The lock-out feature is a safety device to prevent accidental dropping of loads when the machine is stopped and the system is at zero pressure, even if the valve control lever is actuated. The complete stack of valve sections is protected by a single pilot operated relief valve set at 1625 pounds per square inch which is installed in the port plate.

One four way-three position valve section is used to control the hydraulic cylinders which vary the wheel tread and the row spacing of the head. Another controls the double-acting cylinder which rotates the unloading auger from the transport position to the unloading position, Figure 27.

A three-way three position valve section is used to control the two single-acting hydraulic cylinders which raise and lower the row units. Two additional sections control the single-acting hydraulic cylinder which raises the unloading auger and the Char-Lynn M201B hydraulic motor which powers the unloading auger system, Figure 27.
Figure 26. The Cessna stack type valves which control various machine functions are located as shown here under the front part of the control console.

Figure 27. The valves control the cylinders which rotate and raise the unloading auger and the hydraulic motor which powers it.
A complete schematic of the hydraulic circuit including the hydrostatic ground drive and auxiliary machine control system is shown in Figure 28.

Generally, agricultural equipment design is characterized by operation of a machine for only a small portion of the year. Thus, belts, clutches, bearings, and other drive components are commonly designed for a limited life because of economic considerations and sometimes tend to be marginal in design. In contrast, industrial equipment generally is designed for more intensive use with a larger margin of safety for overloads. Also, weight and physical size are not as critical in industrial applications where the equipment is stationary. The more intensive use of industrial equipment can justify higher quality components having greater initial cost.

Because of the high value of the research plots to be harvested, the effect of time lost upon data taken, and the large number of engagement-disengagement cycles expected in plot harvesting, all power transmission components of the plot harvester were designed using industrial rather than agricultural machine design procedures. This was done in order to secure a higher degree of machine reliability.

A 1750XH300 Browning "gearbelt" was selected as the main drive belt for the plot harvester, Figure 29. This 7/8" pitch, 3" wide belt is capable of withstanding severe shock loading while delivering non-slip power to the Sundstrand pump,
Figure 28. Schematic of the hydraulic circuit of the plot harvester
Figure 30, and the input shaft to the cylinder clutch. Browning 30XHR300 sheaves are used on the engine crankshaft and the 1 3/8" shaft which powers the Sundstrand pump. A Browning 40XHR300 sheave is used on the input shaft to the cylinder clutch, reducing its speed to 1950 revolutions per minute. This 1 1/2" shaft is supported by a pair of Browning FB200 bearings located on each main frame member. The left portion of the shaft runs whenever the engine is running; the right portion can be connected to the "live" portion by engaging the appropriate lever on the operator's platform. This lever controls the Dodge 108MI clutch mechanism located between the two shaft sections, Figure 31.

Power transmitting pressure on the friction disc of the clutch is developed by a circle of steel balls which are forced into a wedge shaped groove by a sliding cam. The principle of the inclined plane is used to multiply the force exerted on the shifter collar into a much greater force on the friction disc. The clutch is simple and compact with all the operating parts completely enclosed. The driving ring is keyed to the left shaft section while the clutch mechanism is keyed to the right section. A pilot bearing is used to maintain positive alignment of the shaft ends. This 8" clutch is designed to transmit 33.3 horsepower at 1950 revolutions per minute. The clutch chosen has only a single clutch plate and is purposely oversized for the application to improve heat dissipation and clutch life under frequent engagement.
Figure 29. Rugged, industrial belt delivers non-slip power from the engine to the other components

Figure 30. The main drive belt delivers power directly from the engine to the Sundstrand pump
When the cylinder drive clutch is engaged, power is transmitted from the clutch shaft to the cylinder of the sheller by a 97.5" "L" section "poly-v" drive belt. A 5.6" pitch diameter sheave drives the 16" pitch diameter sheave located on the cylinder input shaft, giving a cylinder speed of 670 revolutions per minute. The belt idler is spring-loaded so the belt can slip if the cylinder should stop turning due to an obstruction or overload. Power is transmitted to the separating mechanism and conveyors by an extended pitch number 50 roller chain driven from the cylinder shaft, Figure 32.

A hardened 14 tooth number 60 roller chain sprocket is located on the extreme left end of the input shaft to the cylinder clutch. The roller chain driven by this sprocket furnishes power to the row unit drive clutch and the Cessna hydraulic pump, Figure 33. The 11 tooth hardened sprocket on the pump shaft has a maximum speed of 2480 revolutions per minute. The clutch driving ring is bolted to a 45 tooth sprocket which turns at 606 revolutions per minute on a sealed bearing when the row unit clutch is disengaged.

The 8" Dodge 108MI clutch is located on the pivot point of the row unit lift arms, Figure 34. When the lever controlling this clutch is engaged by the operator, the clutch mechanism, which is keyed to the 1 1/4" shaft, is locked to the clutch driving ring by the friction disc. An 18 tooth number 60 roller chain sprocket keyed to the shaft then drives
Figure 31. Power is supplied to the cylinder and separating mechanism through this over-center clutch controlled by the machine operator.

Figure 32. Cylinder is driven by a belt. The separating mechanism and conveyors are driven by a chain from the cylinder shaft.
Figure 33. The Cessna hydraulic pump and the clutch controlling the row units are driven from the left end of the shaft which powers the cylinder clutch.

Figure 34. The clutch for the row units is controlled independently of the cylinder clutch.
a 19 tooth sprocket bolted to a hub on the hexagonal drive shaft of the row units, Figure 35. This hexagonal shaft runs about 575 revolutions per minute, and drives the cross auger at 130 revolutions per minute through a chain drive from the left end. This same shaft also drives the two slip-clutch assemblies which power the row units. The 1 1/8" hexagonal drive shaft is of sufficient length to drive both row units in the 40" position. The excess shaft slides through the right row unit slip-clutch assembly and is hidden behind the standing corn shield in narrower row positions, Figure 36.

Commercial two row corn heads generally have three sheet metal gatherers; a left and right gatherer and a center gatherer which covers the entire space between the rows being harvested. Sheet metal gatherers of unconventional design were necessitated by the variable row spacing feature of the plot harvester. Because only very simple sheet metal bends could be made with the equipment available, the design shown in Figure 37 was used. The gathering units are fabricated from 16 gauge cold-rolled sheet steel with 1" x 1" x 1/8" angle iron reinforcements. The outside of each gatherer is open to allow easy servicing or replacement of the gathering chains. A short, floating gathering point is used to help shorten the overall length of the machine while maintaining a degree of contour following ability. As seen in Figure 38, four gatherers are used in place of the normal three. Because of the 14 inch variation in the distance between the two row
Figure 35. The row units are driven by a number 60 roller chain.

Figure 36. The hexagonal drive shaft slides through the slip-clutch of the row unit and extends out behind the standing corn shield (removed here) in narrow rows.
Figure 37. Sheet metal gatherers designed for the snapping units of the plot harvester.

Figure 38. Four, rather than the normal three, "snouts" are required with the variable row spacing feature of the harvester.
units, no solution using only three gatherers appeared feasible. Although the gap between the units is not desirable from the standpoint of ear loss through the opening, this appeared to be the most practical solution of the alternative designs considered.

Two overlapping sheet metal sections, one fastened to each half of the row unit framework, form the rest of the corn head. The sheet metal section on the right side of the head slides over the top of the left section when the row spacing is changed. A narrow section of the back sheet must be removed in row spacings less than 30 inches because it begins to restrict the throat over the feeder conveyor. This requires the removal of six 1/4" cap screws and the sheet metal filler strip. No other changes are required when row spacings are varied.

A close fitting auger transfers the corn from the rear of each row unit to the throat of the conveyor, Figure 39. The right portion of the auger telescopes along the square connecting shaft when row spacings are varied. The right and left hand flighting are 180 degrees out of phase to give more uniform feeding to the cylinder and to eliminate plugging by trash in narrow rows.

An overshot belt conveyor was designed for use as the feeder conveyor from the corn head to the cylinder of the sheller. A 2 ply cotton-nylon belt with 3" high cleats was selected for this application. The 16" wide belt is
approximately 136" long with cleats bolted to the belt approximately every 10", Figure 40. The rubber cleats each have an aluminum strip molded into the base and are attached to the belt with flat-head screws threaded into the aluminum strip. The belt turns on 6" diameter steel rollers with a small idler roller located under the center of the loaded belt to help support the weight of the corn. The ears of corn fall by gravity onto the belt. The boot of the conveyor is self-cleaning since it is designed with essentially "zero" clearance in relation to the top of each paddle. As each paddle passes around the bottom roller, it wipes the boot of the material which has collected there. The conveyor discharges at the top end where the ears fall to the cylinder by gravity, Figure 41. It was felt that this type conveyor would be more nearly self-cleaning than any other design considered.

The processing of the crop by the 16" diameter cylinder and concave is similar to that of larger commercial combines. The rack and sieve are both very conventional in operation; only after the shelled corn drops through the sieve is there any departure from convention.

The original grain collection system of the sheller was eliminated because the combination of an auger and a side-sloping housing was not self-cleaning. These items were replaced by two conveyor belts which deliver shelled corn to the short cross-auger. The two belts run on small steel rollers, Figure 41.
Figure 39. Telescoping cross-auger has a close fit to clean the corn head of shelled corn

Figure 40. Overshot conveyor belt with 3" cleats delivers crop from the corn head to the cylinder
Figure 41. Schematic of the functioning components of the plot harvester
A short cross auger transfers the corn to the left side of the machine, delivering it axially into the vertical elevator. The elevator is fabricated with a tight housing equipped with a divider between the loaded and unloaded sides of the elevator. Steel roller chain and rubber paddles which have "zero" clearance inside the elevator housing give excellent clean-out characteristics. Essentially, all the corn that is delivered to the cross-auger of the corn head is discharged from the grain elevator, with the exception of threshing and separating losses.

The shelled corn from the vertical elevator is delivered to a small bin located on top of the grain tank, Figure 42. This weigh bin has capacity for about 105 pounds of corn. A light tubular framework is supported on one pair of bearings. This frame supports another set of bearings upon which two strain-gauged load cells are mounted. The weigh bin is then mounted securely on top of the load cells. The use of the two pairs of bearings 90 degrees apart insure that the weigh bin can remain level regardless of the terrain that the machine encounters. This is critical since the load cells will give accurate results only when the weigh bin is level.

The Budd strain indicator shown in position in the machine control console, Figure 43, will convert the electrical output from the load cells directly into pounds contained in the weigh bin, after the system has been calibrated initially and the appropriate gauge factor set manually on the indicator.
Figure 42. Weigh bin for determining plot yields

Figure 43. Budd strain indicator in the machine control console
Siemens (21) originally used a load cell of the general shape shown in Figure 44 for measuring forces from model tillage tools in soil bins. A design derived from this original one has been used successfully in the weighing system of a combine used to harvest research plots at Iowa State University for several years.  

Four 1/2" strain gauges are mounted on each load cell and are connected with two gauges of each load cell in series in each arm of the Wheatstone bridge of the strain indicator, Figure 45.

Power for operation of the electrical circuit is supplied by a battery in the strain indicator which can be recharged by plugging the unit into a 110 volt outlet.

The system is calibrated by placing known loads in the weigh bin. By setting the digital readout of the strain indicator to the known value of the weight in the bin, the required gauge factor can be found by adjusting the gauge factor knob until the needle is balanced on zero. The system should be checked for linearity through the maximum bin capacity, although no problems of non-linearity were encountered in the plot harvester weighing system.

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Figure 44. Load cell configuration with four gauges used in the plot harvester weighing system

Figure 45. Strain gauge circuit using the eight gauges of the two load cells with a Budd P-350 digital strain indicator for weighing plot yields
For normal operation, the indicator with the known gauge factor set is balanced to zero using the balance knob which is then locked. A known load is placed in the weigh bin and the digital readout knob advanced until the needle again returns to zero. The value shown by the digital readout should agree with the known value if the correct gauge factor was used originally and the empty system was correctly balanced to zero. If the known value and the indicator value agree, the load can be removed and the zero balance again checked. Harvesting of a plot then proceeds. After the plot yield is in the weigh bin, the indicator is activated and the digital readout knob advanced until null balance is obtained. The indicated weight is the plot yield within one tenth of a pound. A moisture sample is manually removed from the weigh bin, the bin trip is activated, dumping the corn from the weigh bin into the larger storage tank, and harvesting of the next plot can proceed.

Two cautions are worth noting at this time. First, the weigh bin should be leveled each time a plot yield is weighed. Two line levels are mounted on the bin as a convenient check of this. If the bin is not level, only the vertical reaction of the weight in the tank will be recorded by the indicator, thus giving an indicated plot yield less than the true yield. Second, strain gauges are sensitive to temperature changes of even a few degrees. Thus, the system should be checked for zero balance and correct digital readout under a known load at
various intervals throughout the day. Slight adjustments in
the indicator settings may be required to compensate for the
temperature effect, even though the arrangement of the gauges
in the bridge provides for temperature compensation. Also,
the indicator should be recharged after each day of use to keep
the battery at peak electrical condition.

A grain tank with an approximate capacity of 20 bushels
is located on the rear of the machine, directly behind the
engine. The small weigh bin dumps directly into this tank.
It is a "saddle-tank" design, with corn being stored along
each side of the separating mechanism as well as over the top
of it. This was done for more efficient space utilization and
greater machine stability due to a lower center of gravity
than could be obtained by storing the material only in a top-
mounted tank. The tank extends across the full width of the
machine.

The grain tank is emptied by a system of 6" diameter
augers. Corn flows down the inclined tank floor into an auger,
Figure 46, which passes under the separating area from the left
"saddle" through the right one to a vertical auger. The corn
is transferred from the vertical auger into an auger which is
normally stored along the right side of the harvester. This
auger can be rotated hydraulically to unload into wagons or
trucks, Figure 27. The auger also raises hydraulically to
unload into trucks up to 9 1/2 feet high. The complete auger
system is driven by a Char-Lynn M201B hydraulic motor located
on the pivot axis of the rotating section of the unloading auger, Figure 47. The single motor supplies power through chain drives and bevel gears to all sections of the unloading auger system.

Most machine controls are located in the control console at the operator's station, Figure 43. This groups the controls conveniently at the operator's right hand, leaving his left hand free for steering.

The hydrostatic speed control lever is located adjacent to the operator's seat on the front portion of the console. The lever must be in neutral before the starting circuit of the machine can be activated; a safety switch prevents "jack-rabbit" starts. Oil flow rate from the Sundstrand pump is controlled by this lever; pushing the lever forward results in forward travel. By moving the lever farther forward, the ground speed of the machine is increased. Moving the lever to the rear of neutral will cause the machine to move to the rear.

The next lever to the right is for raising or lowering the row units of the harvester. Forward motion of the lever lowers the row units.

The three levers in a cluster to the rear are controls for the unloading augers. Movement of the first of these results in raising the unloading auger, the second rotates the auger to the unloading position, and the third activates the hydraulic motor for unloading.
Figure 46. Lift-off shields allow easy servicing and machine adjustment. Note also the included tank floor which empties corn into the unloading auger.

Figure 47. Extensive shielding of the hydraulic motor drive for the unloading system.
The remaining lever, on the far right, varies the row spacing and wheel tread of the plot harvester.

The center portion of the console contains an ammeter, an oil pressure gauge, a fuse, the choke control, a horn switch, a key-type ignition switch, and the engine speed control lever. A tray for pens and pencils is located near the top of this section of the console. The forward sections of the console are painted flat black to decrease light reflection.

The rear section of the console contains one of two alternate positions for the Budd strain indicator. Immediately below this is a shelf for storage of a clipboard with field maps, data sheets, and other papers.

The control levers which operate the clutches for the row units and the separating mechanism are just to the left of the steering column. The right lever controls the cylinder and separating mechanism while the left one controls the row units. The cylinder clutch lever must be engaged before the row unit clutch lever can be engaged, to prevent plugging of the machine.

A second seat and platform are provided to the rear of the grain tank for use by a person obtaining moisture samples. While it would be desirable to complete this operation on the machine, the associated problems make it easier to collect a sample for later moisture determination at the laboratory. A moisture sample is manually removed from each plot yield for
moisture determination by the oven method or a suitable moisture tester. A console for the strain indicator is provided on this rear platform as an alternative to the position in the machine control console. A clipboard rack and storage for moisture samples is also provided. A horn switch is included so that the man on the rear platform may signal the machine operator when the necessary operations are complete and the next plot may be harvested.

Safety features have been incorporated to reduce the risk of physical injury. Both platforms are covered with a non-skid rubber matting and are equipped with handrails. Mounting ladders having non-skid steps are used at both platforms. Extensive use of shielding prevents operating personnel, or spectators, from becoming entangled in moving parts, Figures 46 and 47. Slow moving vehicle emblems and bright orange paint with yellow trim increase visibility of the machine when it is transported on public highways.

The completed plot harvester is shown in Figures 48, 49, 50, and 51. The total machine weight is approximately 7000 pounds with about 5600 pounds on the two drive wheels. The overall length is 17 feet with a maximum height of 113 inches. The width to the outside edges of the standing corn shields is approximately 89 inches with the row units spaced for 40 inch rows.

The completed harvester can be loaded on a tilt-bed implement truck for transport and still remain below the 13 1/2 feet maximum height dictated by Iowa law. No special transport
Figure 48. Right side of completed plot harvester

Figure 49. Left side of completed plot harvester
Figure 50. Plot harvester adjusted for operation in 26 inch rows

Figure 51. Plot harvester adjusted for operation in 40 inch rows
permits are required because the unit is also within legal limits of length and width.

Field Testing

Even when extensive use is made of commercially available components, the design and construction of a complex experimental machine like this corn plot harvester is a time consuming process. The design work was initiated in December, 1968, but little construction work was started before May, 1969. Nevertheless, the plot harvester was subjected to limited field testing in late November and December, 1969.

Testing was started as soon as the functional portions of the machine were complete. Although many items desirable for operating even an experimental machine, such as handrails, platform, clutch control levers, and safety shields were temporarily omitted, it was felt to be far more desirable to gain some field experience with a partially completed machine than to have no test time on a completed one.

Several days were required for adjustment and minor alterations. The first serious functional difficulty resulted in jamming of the feeder conveyor, Figure 52. The random orientation of the ears of corn in the feeder conveyor occasionally resulted in an ear wedging between a paddle and the top of the conveyor. The original design had insufficient clearance between the paddles and cover so a 2 1/4" extension was added to each of the conveyor sides. The extra clearance
eliminated this source of plugging.

A significant amount of recirculation of corn and trash was noted in the same conveyor. The access door to the top of this conveyor was removed and the conveyor observed during operation. Much of the problem seemed to be caused by ears rebounding from the top of the shroud which covers the cylinder above the axis of rotation. The trajectory of these rebounded ears placed them in a position to be returned down the conveyor by the same paddle which brought them up. A rubber flap was installed in front of the shroud to absorb energy. The problem was reduced considerably but some recirculation still occurs. This is partially due to the restricted throat of the sheller. Part is probably due to the small roller diameter and the relatively high speed of the conveyor which allows insufficient time for the ear to fall clear of the paddle before it "snaps" around the roller, carrying the ear around a second time. The material can make the trip down the return side of the conveyor and around the boot easily so further efforts to better the situation were dropped. A larger roller and a slower conveyor speed would alleviate much of the problem, but space is at a premium at both ends of the conveyor and a slower conveyor would create more intermittent feeding with more ears per paddle. It is felt by the author that further work would bring greater returns in other areas of the machine than on the recirculation problem. A problem with slippage between the conveyor belt and the steel drive roller at the top of the
conveyor was encountered, but occurred only when the conveyor became jammed. The top roller became polished as a result of this slippage so the belt was removed and intermittent weld beads were applied to the roller. A better method would be to bond a traction surface to the roller but this was not done. This would have required a completely new shaft and roller of smaller diameter because of the close tolerances between the belt and conveyor housing. It is recommended that this be done on any future machines in spite of the fact that improvements in the rest of the conveyor eliminated the problem as a serious defect.

Very intermittent feeding from the auger in the head to the conveyor was noted in 40 inch row spacings. Ears of corn would pile in front of the throat between the ends of the auger until the whole pile would be pushed into the conveyor at once. Two "kicker" paddles were added to the auger flighting to give a steady flow of material into the conveyor. These two paddles were rectangular and caused stalks to wrap on the square shaft between the sections of the auger. The rectangular paddles were then replaced with triangular shaped paddles, Figure 53. These solved the feeding problem without causing wrapping of stalks on the shaft.

Some problem of "hair-pinning" of lodged stalks around the sheet metal gatherers was noted while operating in narrow rows. However, the corn had been severely damaged by hail and wind which caused very unusual harvesting conditions. Some of
Figure 52. Occasional wedging of ears against the conveyor top resulted in a plugged conveyor. The situation was later corrected.

Figure 53. Uneven feeding in wide rows was corrected by adding two "kicker" paddles to the auger.
the stalks would enter the head with the top going up one gatherer and the root of the plant up the gatherer of the adjacent row unit. No corrective action was taken for this unusual condition because commercial combines operating in the same conditions were plagued with identical problems.

A problem existed with the two conveyors under the separating area. Unless very accurately adjusted, the belts would run to the side of the rollers. The resultant rubbing frayed and stretched the edges of the belts. Wider canvas seals were installed along the edges to prevent shelled corn from dribbling over the edges of the belts. However, the adjustment of the rollers is still very critical to prevent damage to new belts since the damaged belts have little tension along the edges and might still allow corn to dribble out of the machine onto the ground.

A considerable amount of time was consumed making the noted modifications, but finally the machine appeared to be functioning well, Figures 54 and 55. The only exception came in the form of excessive losses of corn from the separating mechanism. Since the Ford sheller was not designed for operation with all the husks and trash being delivered to the rack, higher losses would possibly be expected.

Table 2 shows field data collected indicating losses from the row unit snapping rolls and the separating mechanism. The input speed to the row units was approximately 600 revolutions
Figure 54. Operator's view of corn head operating in 30 inch rows

Figure 55. The low profile of the row units gave good performance, even in lodged corn
per minute and the sheller cylinder was run at 670 revolutions per minute.

Table 2. Measured shelled corn losses from the plot harvester

<table>
<thead>
<tr>
<th>Run</th>
<th>Plot size, acre</th>
<th>Source</th>
<th>Loss^bushels/acre</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.01000</td>
<td>separating</td>
<td>1.84</td>
</tr>
<tr>
<td></td>
<td>0.00063</td>
<td>snapping roll</td>
<td>1.13</td>
</tr>
<tr>
<td>2</td>
<td>0.01000</td>
<td>separating</td>
<td>0.89</td>
</tr>
<tr>
<td></td>
<td>0.00063</td>
<td>snapping roll</td>
<td>2.29</td>
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<tr>
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</tr>
<tr>
<td></td>
<td>0.00063</td>
<td>snapping roll</td>
<td>1.71</td>
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<tr>
<td>4</td>
<td>0.01000</td>
<td>separating</td>
<td>1.44</td>
</tr>
<tr>
<td></td>
<td>0.00063</td>
<td>snapping roll</td>
<td>1.43</td>
</tr>
</tbody>
</table>

^Corrected to 15.5 percent moisture content. Wind and hail damaged corn with light test weight.

Visual examination of the material discharged from the separating mechanism yielded some clues to the source of the separating losses. Many of the discharged husks contained several corn kernels shelled from the butt of the cob but not separated by the rack. In addition, a heavy initial loss of shelled corn travelling at high velocity over the rack was observed before a sufficient volume of trash was established on the rack. This loss appeared to be from the first few
ears shelled by the cylinder and discharged from the rear of the concave extension. The material was thought to have sufficient energy to completely clear the short rack of the machine and be lost before a wall of trash and husks was established to dampen the discharge from the cylinder.

A rubber curtain 8 inches long was added to the sheller about 6 inches to the rear of the concave extension. This was done to retard the discharge from the cylinder and allow more time for separation on the rack. Also, the curtain should create more agitation of discharged material for better separation. Table 3 confirms that a considerable improvement in performance due to this curtain was obtained. The input speed to the row units was lowered to about 575 revolutions per minute, giving an improvement in snapping roll losses.

The row unit input shaft was further slowed to 540 revolutions per minute by another sprocket change, but trash movement appeared to be sluggish. The shaft speed was returned to 575 revolutions per minute, which appeared to be the lowest satisfactory speed under the particular field conditions encountered.

The improvement in operation due to the various changes and modifications eventually led to satisfactory field performance, Figures 56 and 57. No mechanical failure occurred during the entire period of field testing. The hydraulic system and the hydrostatic ground drive performed flawlessly. Sufficient power and tractive effort made operation quite
satisfactory in mud and snow with no handling problems encountered in either. The plot harvester was able to start and pull heavy loads of shelled corn, even in soft field conditions. A total of about 500 bushels of corn was harvested before the weather ended further field testing.

Table 3. Measured shelled corn losses from the plot harvester

<table>
<thead>
<tr>
<th>Run</th>
<th>Plot size, acre</th>
<th>Source</th>
<th>Loss&lt;sup&gt;a&lt;/sup&gt; bushels/acre</th>
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</thead>
<tbody>
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<td>0.00063</td>
<td>snapping roll</td>
<td>0.57</td>
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<td>snapping roll</td>
<td>0.50</td>
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<td>snapping roll</td>
<td>1.14</td>
</tr>
<tr>
<td>3</td>
<td>0.01000</td>
<td>snapping roll</td>
<td>0.54</td>
</tr>
<tr>
<td></td>
<td>0.00063</td>
<td>snapping roll</td>
<td>1.43</td>
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<tr>
<td>4</td>
<td>0.01000</td>
<td>snapping roll</td>
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</tr>
<tr>
<td></td>
<td>0.00063</td>
<td>snapping roll</td>
<td>0.86</td>
</tr>
</tbody>
</table>

<sup>a</sup>Corrected to 15.5 percent moisture content. Wind and hail damaged corn with light test weight.

The turning circle diameter of the plot harvester was measured to be 27 feet. Observations and comparisons of the plot harvester to commercial combines, Figures 58 and 59, confirmed that the small machine was definitely superior in handling characteristics, especially in small plots and narrow roadways.
Figure 56. Plot harvester was performing well by the end of the test season

Figure 57. A portion of the corn harvested during field testing
The plot harvester is small compared to large commercial combines. Short wheelbase and overall length give superior handling as compared to commercial machines.
SUMMARY

A two row self-propelled experimental corn plot harvester was designed, constructed, and tested. The project was the result of a contract between Iowa State University and the United States Department of Agriculture to develop "a corn harvester that will harvest efficiently small corn plots that are used in conducting basic research in corn breeding and quantitative genetic studies in Iowa."

The machine is capable of harvesting row widths from 26 to 40 inches by virtue of the hydraulically adjusted corn head. The ears of corn are shelled and delivered to a weighing system for yield determinations. The small physical size, hydrostatic drive, variable width corn head, integral weighing system, and self-cleaning conveyors are features well suited to plot work. Although many requirements of corn researchers have been fulfilled by the design, extensive use of the machine will certainly indicate desirable refinements which could be included in future designs.

More detailed information and layout drawings of the corn plot harvester can be obtained from Dr. Stephen J. Marley, Agricultural Engineering Department, Iowa State University, Ames, Iowa.
CONCLUSIONS

The completed plot harvester fulfilled all the essential design objectives of the project. Specifically the harvester:

1. Is capable of harvesting corn planted in row widths from 26 to 40 inches.

2. Picks and shells the ears of corn and deposits the shelled corn in a weighing bin. The weight of the harvested grain can be observed and recorded at the operator's station or the moisture sampling station adjacent to the weigh bin.

3. Provides for infinitely variable speed from 0 to 8.5 miles per hour, by means of hydrostatic drive.

4. Has a top speed adequate for rapid transport between plots or experimental fields in close proximity.

5. Physical size is small enough to permit transport on a trailer without exceeding legal height and width limitations in Iowa.

6. Is highly maneuverable within plots and from plot to plot.

7. Incorporates safety features to minimize the risk of physical harm to operating personnel or bystanders.

8. Has undergone limited field testing, leading to some design changes to improve field performance. However, final evaluation of reliability and adequacy of the design will have to be made by the corn breeders who use the machine.
ACKNOWLEDGMENTS

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LIST OF REFERENCES


