AUTOMATIC FEED-RATE CONTROL OF COMBINES

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by

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ABSTRACT

This thesis describes the development of an automatic control system which controls the forward speed of a self propelled combine in relation to the load on the cylinder. The purpose of the system is to keep the feed rate of the combine constant in order to increase the machine's efficiency. The system can be adjusted according to the crops and fielded conditions encountered.

Field tests were carried out in wheat, oats, and barley at several moisture and yield levels. Tests indicated that the system was able to control the feed rate at a preset level, within the range of the system dead zone. On the basis of these tests and previous grain loss vs feed rate tests, it is concluded that the control system is able to keep the grain losses from the combine at a preset level, within a range determined by the system characteristics.

The system is low in cost, simple to operate, and does not interfere with the control or operation of any other combine mechanisms.
Cereal grains represent the largest part of the world's food supply. In many countries, cereals are almost the only type of food generally consumed. Approximately four pounds of grain are needed to produce one pound of meat, and for this reason meat is considered to be a luxury in many heavily populated countries. Although much of the world's meat supply is produced from forage crops, in almost all cases grain is also used to some extent. The economical production of grains therefore, is of critical importance in sustaining life in all parts of the world, and will become even more important with the increase of population, and the urbanization of arable farmland. The amount of new land that can be opened up for grain production is already severely limited in many countries, and in all cases, the price of claiming this land is becoming very high.

Maximum production from existing grain land is therefore a necessity in order to maintain an economical and adequate food supply. This necessitates the use of good cultural practices, fertilization, irrigation and elimination of waste. This thesis will consider one aspect of waste elimination. Grain losses due to poor harvesting methods cost Western Canadian farmers millions of dollars annually. Inefficient use of time and mechanical limitations are the chief source of these losses.

Grain losses and harvesting time are very closely interrelated, and yet it is possible to have large variations in the relationship between these two factors. Low grain losses do not necessarily mean that an excessive amount of time must be spent in the field. With the early self propelled combines, there were only a few ground speeds that could be used,
due to the limitations of the transmissions, which allowed two or perhaps three working speeds. Obviously, crop conditions and density were not always suited to these particular speeds and underloading and overloading were common problems. This often resulted in poor field efficiency and high grain losses. Even now, with pull type combines this may be a problem, although it has been alleviated somewhat by a large increase in the number of working speeds, and by power shift transmissions which have eliminated time loss due to shifting of gears.

In the last twenty years self-propelled combines and other self-propelled machines have been marketed with variable speed drives, which enable the operator to select any speed within a range that he desires, without stopping the forward travel of the combine. This has increased harvesting efficiency very greatly, both in terms of reduced grain losses and time spent. However, the choice of a speed is still left up to the operator and the selection of the optimum speed for proper loading, depends on whether the operator has the skill to detect the load, and whether he is willing to make a speed change to correct the load, and whether he is willing to make a speed change to correct the load when necessary. The importance of maintaining the load (throughput)* at the machine's optimum working level, has been shown by numerous capacity tests. Although the relationship between grain losses and throughput is well known to combine manufacturers and experimental farms, and to a lesser extent to farmers, the maintenance of a constant feed rate* at or near the optimum level is largely disregarded, due mainly to the extreme difficulty involved in detecting this optimum rate.

*Refer to Appendix A
A device which would indicate, or better still, control the loading of the harvesting machine would seem to be of great value in increasing harvesting efficiency. If the loading were controlled automatically at the correct level, separation losses would be greatly reduced and in addition, operator fatigue would be lessened.

The object of this thesis is to describe the development and testing of a control system which will automatically control the feed rate of the combine in an effort to obtain a maximum field efficiency and a minimum grain loss from the machine.

**Grain Losses**

Many refinements and improvements have been made on present day combines, but the grain combine still remains the most complex and expensive piece of machinery commonly used by farmers. Despite the many improvements found on modern combines, accurate adjustment of the various mechanisms is of critical importance if the combine is to operate satisfactorily. A considerable amount of skillful judgment is required to make these adjustments. Much effort has been made to reduce the complexity of the adjustments, and the methods to be used in making them are included in most combine operator's manuals.

The combine operator's objective is to harvest as much grain as possible in a given time with the least amount of grain loss. Much research has been carried out to determine the cause of grain losses, especially on their relationship to the amount of material passing through the combine in a given time. Several of these tests are discussed here, as well as the results of the tests.
The method of loss testing described is the one used by the Agricultural Machinery Administration (A.M.A.), Regina, Saskatchewan. However, a very similar method is used by the National Institute of Agricultural Engineering in England, the Massey-Ferguson Co. of Toronto and others.

Large canvas bags are used to catch the material as it comes out of the combine. The material from the shoe and the walkers is caught in separate bags. To start the test, the combine is moved into the crop at a uniform speed until all the mechanisms are loaded. When an equilibrium is reached between input and output, the bags are placed in a position to catch the material as it leaves the combine. The grain entering the grain tank is caught at the same time. When any one of the bags is full, the test is stopped. The time and length of run are measured and recorded. Each bag of material is weighed separately and the weights recorded. The loose grain is then separated from the chaff and straw from the shoe and walker bags and weighed. All the straw and chaff is then put through a threshing device, and the grain which is threshed out is weighed to obtain cylinder loss. Using the weight of the grain caught at the grain tank and the weights of grain obtained from these separations it is possible to calculate the per cent grain loss from the shoe, walker and cylinder.

The straw and chaff throughput is calculated, and the yield of the crop may also be determined. By running successive tests at various throughputs, a relationship between grain losses and throughput may be established for the combine.
Tests carried out by A.M.A. in this manner have consistently indicated sharp increases in grain loss with increases in throughputs (Fig. 2). The loss-throughput curves generally plot as straight lines on log-log paper (Nyborg 1963). The equation for the walker, shoe, cylinder, or total loss vs feed rate is of the form:

\[ Y = KX^n \]  

where \( Y \) = percent loss
\( X \) = feed rate (throughput), and
\( K, n \) = constants whose values depend on the particular machine and crop.

Some typical values of \( K \) and \( n \) are \( 8 \times 10^{-6} \) and 2.3 respectively. These values represent walker loss obtained with a Massey Ferguson combine in a crop of wheat at 35 bus/acre, and with a grain/straw ratio* of 1.04. The rapid increase in grain loss with throughput is indicated by the high value of the exponent (\( n = 2.3 \)). Maximum loss in this particular test was 4%, but with a slightly lower grain/straw ratio the maximum loss was 11%.

By the end of 1963, six different combines had been tested by A.M.A. and each one showed the same type of relationship between grain loss and throughput. The National Institute of Agricultural Engineering has also carried out several combine tests. Their results (Fig. 3) indicate high grain losses when the optimum feed rate is exceeded. In the discussion of the results, Phillipson (1964) states, "---since relatively small changes in forward speed could result in a considerable change of the straw shaker losses, it was considered that the provision of some means of checking the

*Refer to Appendix A
speed was highly desirable. He also states that "in all cases there was a relatively small range of straw throughputs, in any field, over which the performance of any machine was at or near optimum". The need for precise control is pointed out in the statement, "It should be remembered that an increase of 1/2 mph or less may double the rate of grain loss when a machine is working at or near its optimum rate of work".

Mark (1963) reported that in tests by Massey-Ferguson in five different countries, losses were found to increase very rapidly with increasing feed rates (Fig. 4). A typical test showed a loss of 1.5% at 200 lb./min. and 7% at 300 lb./min. Losses approached 30% in some cases before a limiting throughput was reached. Limiting throughput was determined by the inability of the machine to cut or convey the material uniformly.

At Davis, California, Goss (1958) tested a 12 ft combine in barley and his results indicated a loss of 2% at a feed rate of 100 lb./min., while at 200 lb./min. the loss was over 20%.

Similar results have been found in Germany, Russia, Sweden and Denmark (Nyborg 1963). This indicates that the problem of grain losses is universal, and also of great economic importance due to the magnitude of the losses caused by overloaded machines. In Saskatchewan alone, it was estimated by Harrison (1964) that farmers lost $20 million on the 1963 crop of wheat and barley due to excessive grain losses. This represents a loss of several hundred dollars per farmer in one year.

An automatic load control system will not necessarily prevent excessive grain losses since the operator can still set the system to maintain a very high feed rate. If it is set properly, however, the operator
is assured of a uniform feed rate near the optimum level. A more accurate load judgment is possible with automatic control since the operator can make his decision on the basis of an extended operating period, rather than on an instantaneous period, as with manual control. With the losses being equalized over the entire field, it is more likely that the operator will make a check of the amount of grain being lost, since he will know that this represents the whole field rather than just a particular small area. It is unlikely that the operator would set the control system so the load is at such a high level that the grain losses would be very high all over the field. If the same harvesting time is used in automatic control, as in manual, the losses will be reduced due to the maintenance of a more even rate of feeding with automatic control. If equal grain losses are to be accepted in automatic and manual operation, less time will be required to harvest the field when using automatic control.
2. LITERATURE REVIEW

Automatic control of grain combine harvesters is not completely new but at the present time there are no combines with automatic control of feed rate on the North American market. Several attempts have been made at automatic control of feed rate, but they have not been satisfactory for commercial use. Some of the devices will be described here, together with comments on their suitability and limitations. Although a literature search was made before the research was initiated, the translation of Nastenko's paper, (1959) and the patent by Budzich (1964) were not available until after the completion of field tests of the control system described in this thesis.

An attempt at automatic control of grain harvesters was made by Pasturczak (1953). In his device, the forward speed of the combine was regulated by the manifold pressure of the engine. This method did not take into account the varying ground resistance and so cannot be considered to control the feed rate accurately. In hilly country especially, this method would be extremely inaccurate and could be quite dangerous.

One of the most successful attempts at automatic control was made by a Russian scientist, V. G. Dymnich (1956). He used a pull-type S-6 combine harvester with an auxiliary engine for his analysis and field tests. Initially a laboratory test was carried out to determine the relationship of feed rate to torque on the cylinder and grain losses in the straw. A canvas belt was used to feed the crop into the cylinder, and torque was measured and recorded using strain gauges, amplifiers and an oscillograph. Samples were taken to determine grain losses in the straw. From these tests, the conclusion was reached that the torque on the cylinder shaft was a sufficiently
accurate means of reflecting the feed rate and also the grain losses in
the straw. Although the torques measured in the laboratory did not agree
exactly with those measured in the field, for the same feed rate, the
general relationship between torque and feed rate remained the same, and
thus merely pointed out the need to adjust the torque level to the specific
harvesting conditions.

For the actual field tests, a control system consisting of a
mechanical strain gauge, a reversible dc motor, electromagnetic relays,
and a power source was used. The mechanical gauge measured the cylinder
torque and positioned a sliding contact in relation to the load. It was
alternately suggested that a spring-type torque meter could be used to
measure the torque. When the torque reached a certain level, the contact
closed one of the relays and this switched on the reversible motor. The
motor controlled the position of the tractor governor lever, and thus
controlled the forward speed of the entire unit. A dead zone was provided
between the underload and overload relays to prevent hunting. This system
thus provided floating control with a dead zone.

The conclusions reached by Dymnik as a result of the field tests
may be summarized as follows:

1. Automatic forward speed control considerably decreased the grain
losses and increased the average throughput.

2. Automatic control made the driver's work much easier.

3. A torquemeter on the cylinder shaft was a very suitable means of
measuring the feed rate.

4. The automatic mechanism was reliable in maintaining an optimum
feed rate.
Although the control system described appeared to be very satisfactory, certain limitations are immediately evident. First, this system is suitable only for a pull-type combine with a separate power source on the combine. Power take-off and self-propelled combines could not use this system. Also, no mention was made of adjustment of response rate. The problem of "hunting" was countered only by an adjustment of the dead zone, which does not provide optimum operation under all conditions. Floating control was used rather than proportional control. Advantages of proportional control are discussed in the section on automatic controls. Several of the other requirements listed in Sec. 3 are not discussed by Dymnich and so it is not known whether or not these requirements were satisfied by the control system. However, the first limitation mentioned is sufficient to make it unsuitable for most harvesting machines.

Nastenko (1959) built and tested an automatic control system for self-propelled combines. An idler pulley loaded by two springs was mounted on the driving side of the V-belt which drives the threshing drum. The shaft of the idler pulley controls the position of the valve which controls the oil flow to the hydraulic cylinder of the V-belt traction drive. The forward speed of the combine is thus controlled by the load on the drum. In field tests grain losses were found to be reduced by 26% on the average, and the rate of grain harvesting was increased by 18%. Limitations appear to be the lack of a response rate adjustment and dead zone, and the lack of a manual override.
Budzich (1964) under U.S. patent No. 3,138,908 described a method of controlling the forward speed of the combine in relation to the load of the entire threshing mechanism. The load sensing mechanism consists of a torsion spring mounted in the drive sheave body. The angular displacement of the spring moves an indicator which actuates the spool valve of a single acting hydraulic system. A second load sensing device consists of an idler on the drive belt which is mechanically connected to a piston in a hydraulic cylinder, which in turn is connected by a pressure line to another cylinder in which a compression spring is placed between the piston and the end of the cylinder. The position of the second piston is a measure of the oil pressure exerted on it, which is a function of the force exerted on the first piston by the idler. The position of the second piston regulates the position of the spool valve. A third type of torque sensor consists of a gear pump mounted in the body of the drive sheave, which is arranged in such a manner as to provide an outlet oil pressure which is proportional to the torque on the sheave and which is then transmitted to the second piston previously mentioned. Although this system meets many of the requirements listed in Sec. 3, it does not satisfy them all.

Limitations of the system devised by Budzich are as follows:

1. Although the total power input to the combine threshing and separating mechanism may give an accurate indication of the throughput, a mechanical torque sensing device must be built to take more load then if cylinder power alone were considered. This will tend to increase the cost of the unit.
2. No means of damping out the instantaneous load fluctuations is included, and this could become a serious problem, especially since the walker and shaker loads are also included. Excessive hunting and wear of mechanical parts would probably result.

3. No provision is made for adjusting the response rate, and this could also result in hunting, or a sluggish response, depending on the actual fixed rate provided.

4. No dead zone provision is made in the system, and this would very likely result in excessive hunting, producing unstable operation and accelerated wear of the system and the combine.

5. The system is devised only for variable speed units that have a single acting hydraulic system. Most of the newer self-propelled combines now have double acting systems so this control system could not be used in the given form.

6. The cost of the system would be quite high. The torque sensing devices include many machined parts and would require close tolerances. This would result in high manufacturing costs, and thus reduce the net value of the automatic control.

7. The system is quite complicated and could easily deter an operator from becoming involved with it.

The combination of limitations listed under 2, 3 and 4 above, must result in erratic performance, and it appears very doubtful that accurate feed rate control could be accomplished with this system.
3. CONTROL SYSTEM REQUIREMENTS

The load sensor and control system must meet several functional requirements to provide satisfactory field operation.

1. The load sensor must be able to sense the load accurately and produce an indication or signal that is a function of the load. A linear relationship would be the most desirable.

2. The load sensor should produce a signal which is a function of the average load rather than the instantaneous load peaks. This calls for the incorporation of a damper in the load sensing system. The damper should preferably be adjustable to eliminate various vibration frequencies.

3. The rate of response of the system must be adjustable, to compensate for changes in the crop variations. Different speed ranges would also call for changes in the response rate. Where crop variations are abrupt and frequent, a slow response rate would be necessary to prevent "hunting". In a high speed range, the absolute speed change would be larger than in a low speed range, for a given sheave displacement, and so a slower response rate would be needed in a higher gear.

4. The load level must be adjustable to permit operation at the optimum rate in various crops and under various conditions.

5. The sensitivity and dead zone of the system should be adjustable. This would allow the operator to choose the range and frequency of operation of the system.

6. The time delay between change in crop volume and actuation of the system should be as small as possible.

7. The system should provide proportional control - that is, the magnitude of the correction should be proportional to the magnitude of the
deviation from the desired load.

8. The load sensor and control system should be structurally sound, and require little or no maintenance.

9. All adjustments should be easily made, and have a sufficient range to cover all crop conditions.

10. A manual override should be included, which would allow the operator to retain control of the forward speed whenever this is necessary. The change from automatic to manual control and vice versa should be easily and quickly accomplished.

Other factors must also be considered in the design of the system. These are:

1. If the control system is to achieve maximum possible usefulness, the initial cost must be kept to a minimum to encourage its use on as many harvesting machines as possible. A quick return of investment would then result, and the farmer would realize a net gain in a short time.

2. The load sensor and control system should be simple to construct, and easy to understand. Simplicity in construction would not only reduce the initial cost, but also encourage farmers to accept the device. Learning to adjust the device and repair it (if necessary) should be relatively easy.

3. There should be very little difficulty in attaching the device to the combine. If possible, it should be built in such a way, that it would not be necessary to drill any holes in the frame, or weld any part of it to the frame.

4. The device should not interfere with any of the ordinary combine adjustments or with the normal operation of the combine.
5. The device should be adaptable to practically all the current models of self-propelled combines with only a few structural support changes necessary. This would help to decrease the cost of the unit, and also help to make it more generally acceptable.

Although some of the requirements listed are somewhat idealistic, they should all be quite closely adhered to, since a serious deviation from one or two of the requirements could make the device either functionally or economically unsuitable. Specific construction requirements are included in Sec. 5.

Since it has been proven that losses increase rapidly with throughput (Sec. 1) and that cylinder horsepower is proportional to the feed rate, it is only necessary to build a control system that will control the cylinder horsepower requirements, in order to be able to control the losses. The overall requirement therefore, is to build a system that will accurately control the cylinder horsepower.
Mechanization has played a large part in increasing farm production, while at the same time, it has decreased the labor requirements. There is often a tendency to confuse mechanization with automation, and although automation necessarily includes mechanization, the reverse is not true. There are many machines that perform a certain operation when a lever is moved or a button is pushed, and these operations are referred to as being "automatically" performed. This is probably due to the fact that a large job is completed with a very small manual effort. However, the machine will do only the exact mechanical operations that it was designed to do, without regard to the success of its purpose, and will continue to operate in the same manner, until it breaks down, is adjusted, or stopped by the operator or some other means. Often a time device is built into the machine to stop and start it, and this almost invariably causes it to be referred to as being automatic.

In many cases, pure mechanization, with or without timing devices, is very satisfactory for the operation intended. However, there are cases where something more than a steady mechanical operation is required. Continuous adjustment by the operator may correct the limitations of a purely mechanical system, but this is time consuming, laborious, and very often extremely inaccurate due to difficulty in judging the results of the operation. If a system is to perform satisfactorily under varying conditions without human guidance it must have the following properties as pointed out by McDonald (1961).

1. The machine must sense the result of its action and use its findings to take corrective measures.
2. It must operate without regard for external disturbances.
3. An automatic machine must have the capacity to correct errors or minimize the difference between desired result and obtained result, all with great accuracy and high speed."

These properties are included in an automatic system by means of a process called feedback.

In a feedback system, the actual result of mechanical actions is compared with the desired result, and if these are not the same, a correction of the controlling variable is made, and the actual result is brought closer to the desired result. In a linear system, the magnitude of the correction is proportional to the magnitude of the error signal (error = desired output - actual output).

In farmstead livestock systems, where automation is probably the most common, several advantages are noted by Skromme (1963). These may be summarized as follows:

1. Reduction of labor requirements. This is important from a sociological standpoint as well as economically.
2. Greater precision resulting in improved efficiency and a better quality product.
3. Increased income due to expansion of the operation.
4. Decreased size of mechanical units resulting in lower initial and operating costs. This is due to having more time available for any given operation, when human labor or supervision is not required.
The automatic controls for watering, feeding, and providing adequate ventilation for 100 steers can be provided for $750 (Skromme, 1963), which is about 3% of the investment in livestock alone. Relatively then, automatic controls in this instance are very inexpensive and can produce significant increases in efficiency.

Many of the advantages listed above can also be claimed for automation in other areas. An additional advantage might be that of being able to use unskilled help for some operations, and still retaining optimum efficiency. This would be the case where several independent machines were being operated by different people, with part of the machine operation being automated, leaving only the elementary processes to be controlled by the unskilled operator. A skilled operator could preset the desired output and then leave the unskilled operator in charge of the machine. In this way, one skilled operator could control many machines and attain optimum efficiency with ordinary unskilled farm laborers.

**Types of Automatic Controls**

Automatic control systems are generally made up of several elements, and these may be either linear or nonlinear.* Usually both linear and nonlinear elements are included in a mechanical system. Some common nonlinearities are backlash, saturation, dead zone, hysteresis, and relays. Analysis of linear systems is usually easier than that of nonlinear systems. For this reason, an attempt is often made to design linear elements; or when this is not possible or desirable, nonlinear elements are sometimes approximated by linear elements for purposes of analysis.

*Refer to Appendix A
Three common types of control systems are: (Cloud, 1963):

1. two-position control,
2. floating control, and
3. proportional control.

The simplest of these is the two-position control and the most complex is the proportional control. However, under some circumstances it is just as simple to provide proportional control, depending on the particular elements readily available for the system. Variations of these three types are possible and in fact, few systems adhere strictly to these basic types.

**Two-Position Control**

As previously mentioned, this is the simplest of the three types of control. In this system, the output variable signals the control device to move to either one of two extreme positions instantaneously. This type is also called a relay control. When the output variable is adjusted to the predetermined value by the control system, the control device moves to the opposite position and remains there until the sensor signals a change. If the two positions are fully open and fully closed, the system is called an on-off system. The two-position on-off system produces less cycling than a two-position system that makes corrections in both directions. However, it can only be used when there is an external factor involved, which influences the controlled variable in such a way that the sensor signals the control device to move to the on position. An example of this type is a thermostat controlled heating system in a house. It can only provide the desired temperature when the outdoor temperature is lower than the indoor temperature.
In many cases, such as the one mentioned, this type of two position control is considered acceptable. A large factor in these systems is the simplicity and economy associated with this type. The two-position control with negative and positive corrections has limited use, due to cycling and wear of mechanical parts. The possibility of excessive cycling is inherent in two-position control, due to the extreme position supplying too much in one direction or the other.

Another variation of the two-position control is the three-position control, or more commonly - two-position control with a dead zone. In this system, corrections may be made in either direction, but between the two extreme positions there is a dead zone. Thus, the system may be moving in either of two directions or else it may be stopped. In a system of this type there is reduced cycling, but if mechanical parts are involved, rapid wear is still likely due to the large accelerations. Precise control is not obtained with such a system, since all changes are made in step fashion. A system of this type was used by Kaminski (1962) in his header height control.

**Floating Control**

A floating control system is intermediate in refinement and complexity between the two position and proportional controls. In this system the control device is moved at a constant speed in either direction when an error is sensed. A dead zone is usually incorporated to reduce cycling. The speed of movement of the control device must be carefully adjusted to suit the response time of the system to prevent excessive cycling. These systems are not suitable where there are large time lags
in the control system response. Rapid changes in load cannot be corrected for with this type of a system, due to the time required for the control device to open fully.

Floating control with a dead zone was used by Daum (1964) for an automatic silo unloader control. A steady flow of silage was required, and this was accomplished by means of lowering and raising the gathering auger in relation to the impeller load. When the load on the impeller reached either of the dead zone limits, a contact was closed and the motor driving a winch was started. The winch raised or lowered the gathering auger at a constant rate until the dead zone was reached and the motor stopped.

A similar system was used by Dymnich (1956) in his combine speed control. The governor lever was moved at a constant rate when the cylinder torque was outside of the dead zone limits.

**Proportional Control**

In a proportional control system, the control device is moved to a definite position corresponding to the deviation from the desired value. In other words, the magnitude of the correction is proportional to the magnitude of the error. This type of system is suitable for slowly changing loads as well as rapidly changing ones. Large time delays in the response of the system cannot be tolerated however. A dead zone is often incorporated into the system to eliminate control changes due to small load changes. A system of this type provides the smoothest operation and results in the most precise control of the output variable. For these reasons, and also due to the fact that hydraulic spool valves provide this
type of control, the proportional control with a dead zone was chosen for the control system discussed in this thesis.

**Sensing Device**

The control parameter for the automatic control system is the amount of grain lost due to cylinder, walker and shoe losses. However, a device which would sense the amount of loss directly would be difficult to construct, and the large time delay involved would seriously reduce the effectiveness of the system. A feed rate detector appears to be the next best choice. It would be desirable to have the sensing device as near as possible to the feed intake of the combine. A volume measuring device could be incorporated into the feeder chain of the combine. However, not all combines have such a conveyor. Also, the relationship between volume of crop entering the combine and grain losses is not presently well known, and to prove any definite relationships between the two would necessitate considerable testing. There are, however, well defined and proven relationships between feed rate and grain losses (Sec. 1). Whether or not volume is directly related to feed rate would depend on several factors, including length of time the crop was in the windrow (for a windrowed crop), the manner in which the crop was fed into the machine, and the amount of compression to which the crop was subjected. A separate extensive study would be necessary to determine the suitability of a sensing device based on crop volume.

However, several tests (Appendix B) have been carried out to determine the relationship between cylinder load and throughput, and it has been shown that the cylinder torque is an accurate indication of the
throughput (Fig. 5). Furthermore, the relationship is very close to linear, and may be assumed so without serious error. Besides this, it is possible to obtain a large actuating signal from a cylinder torque sensing device due to the large load imposed on the cylinder. This is an advantage when using a mechanical control system. A cylinder torque sensing device was therefore chosen for the following reasons:

1. A direct relationship between grain loss and cylinder torque could be proven, by combining the results of loss-throughput tests and throughput-torque tests.

2. A large actuating signal could be obtained from the cylinder drive.

3. The cylinder is reasonably close to the intake of the combine, therefore minimizing the time delay.

4. The cylinder drives on most combines are relatively accessible, making it possible to mount load sensing devices in them without too much difficulty.

5. All the present combines use a cylinder to thresh out the grain, and so the device could probably be used for most of the combines without any major changes.

6. The cylinder is the most susceptible to plugging of all the various mechanisms. This gives some indication of its relative sensitivity, and since a mechanism that is very sensitive to load is desired, it appears that cylinder load should provide the most precise indication of the throughput. A torque sensing device similar in principle to the one built by Scott (1964) was constructed.
Operation of the Control System

The torque on the cylinder shaft is detected by means of a spring loaded idler, which is mounted on the tight side of the chain drive, above the line of centres of the driving and driven sprockets (Fig. 6 and Figs. 23 to 26). This device (subsequently referred to as the torque sensor) actually measures the torque applied to the cylinder shaft, but since the speed of the threshing mechanism remains essentially constant, it can be taken as a direct indication of the power consumed. The position of the pivot arm indicates the amount of power that is being absorbed by the cylinder. The pivot arm is supported by a pivot anchored to the frame of the combine, and it provides the means to transmit the force from the idler to the spring. The pivot arm is limited in its movement by two mechanical stops which are mounted on the support frame. There is also a valve control bracket on the pivot arm, which connects to the lever controlling the position of the spool inside the hydraulic valve. A damper is connected in parallel with the spring from the pivot arm to the support frame. In the hydraulic system, the spool valve at the pivot arm controls the oil flow from the pump to the hydraulic cylinder, which controls the position of the variable speed sheaves. It is possible to cut the automatically controlled valve (subsequently called automatic valve) out of the system by shutting off the pressure ports and directing the oil flow from the pump to the inlet port of the manually controlled valve (Fig. 7). When the automatic valve is being used, the selector valve is turned so that the oil flow is directed to the automatic valve. It is not necessary to shut off the pressure ports of the manual valve since they will be shut off inside the valve due to the fact that the valve spool remains
in the neutral position. However, when the manual valve is being used, the automatic valve may or may not be in the neutral position, so to prevent the oil from returning to the sump through the automatic valve instead of going to the hydraulic cylinder, shut off valves must be provided at both pressure ports of the automatic valve.

The operation of the system is as follows: The spring is initially stretched by the spring length adjustment, with no load on the cylinder. This causes the pivot arm to rotate until it reaches the no-load stop. If the combine traction drive is started and the system is placed in automatic control, without loading the cylinder, the variable speed drive will be adjusted to the full speed position. This is due to the valve being held fully open in the acceleration direction when the pivot arm is held against the no-load stop by the spring. If the cylinder is loaded, the tension of the cylinder drive chain will increase and tend to pull the idler down against the force of the spring. As the load increases, the idler is pulled down further and the hydraulic valve is brought towards the neutral position. When the tension in the chain is sufficient to move the idler down far enough to bring the hydraulic valve to the neutral position, equilibrium is reached, and there will be no more acceleration or tendency to accelerate. If the load increases further, so that the dead zone limit of the valve is exceeded, the valve opens in the other direction and causes the oil to flow to the variable speed cylinder and makes the combine slow down. Therefore, whenever the load is large enough to cause the idler to be displaced downwards from the equilibrium zone, the valve will open in the direction that will cause deceleration of
the combine through the variable speed drive. The deceleration will reduce the load, bringing the system back to equilibrium. Whenever the load is light enough to allow the idler to be positioned above the equilibrium zone by the spring force, the valve will open in the direction that will cause an increase in forward speed and thus an increase in the load on the cylinder. The equilibrium load is determined by the initial tension of the spring, when other factors such as spring preload and spring rate remain equal. The sensitivity of the system depends on the spring rate, the linkage between the spring and the idler, and on the linkage between the idler and the spool. The dead zone is most easily adjusted in the linkage between the pivot arm and the valve spool (Figs. 25 and 26). The damper is adjusted to eliminate excessive vibration, but to still allow fairly rapid pivot arm position changes. The response rate of the system is adjusted by means of the metering valve, which in connection with the pressure release valve regulates the oil flow rate to the hydraulic cylinder. It is possible to shut off the oil flow completely with the valve or to allow any flow rate up to the maximum of the pump capacity. It is not necessary to have a metering valve in the manual circuit. Response rate adjustment is very important in order to obtain the best possible operation of the system. The optimum rate depends on the gear range, the crop variations, the time lag of the system and the size of the dead zone. It is not difficult, however, to find the best response rate setting for any given conditions. The system is initially set for a fast response rate and then gradually reduced by means of the metering valve, until hunting is eliminated.
The manual control is required whenever the combine engine is running and the cylinder is not loaded, such as in road travel. Other situations may also arise where automatic control will not be suitable and a reversion to manual control is necessary.

In order to keep the cylinder chain from becoming too loose as the idler moves down, a bottom idler is mounted on the frame and connected to the top idler so that they move up and down together. A turnbuckle is included as part of the link, and it is used to adjust the initial chain tension.
5. DESIGN AND CONSTRUCTION

The automatic control system was constructed in the Agricultural Engineering laboratory, University of Saskatchewan and installed on a Massey-Ferguson No. 82 combine. Laboratory tests were conducted using this combine; the device was then removed and placed on another M-F 82 combine for field tests.

Since easy removal and attachment was considered to be desirable, the device was designed and built in such a way that it was not necessary to drill any holes in the frame of the combine, or weld any part of it to the combine. Only existing holes and supports were used; the complete device could be assembled and attached using ordinary tools.

It was also necessary to build the device in such a way that it would not interfere with any combine adjustments. The main ones concerned were cylinder speed, concave setting, and tightening of chains.

An effort was made to keep the cost to a minimum, and the design was carried out with this in mind at all times.

**Structural Design**

In order to build the mechanical supports for the load sensing device economically and still with sufficient strength, it was necessary to obtain a reasonable estimate of the horsepower absorbed by the cylinder. However, the maximum power used by the cylinder during operation was not considered to be satisfactory for the design of supports, due to the fact that plugging would cause much higher forces on the cylinder drive. When the cylinder plugs, it is common for the operator to stop the forward travel of the combine and use the full engine power in an effort to start the cylinder turning again. Furthermore, rapid clutch engagement causes
additional load on the cylinder drive chain and mechanical supports due to engine momentum.

The structural design was based therefore, on a maximum available engine power of 60 hp at the cylinder, and a shock factor of 1.5 (light shock) giving a design power of 90 hp. It was not necessary to design the spring for such a high value, since mechanical stops were provided for the pivot arm, which limited the amount of power that could be transmitted to the spring.

Pivot Arm Design

Several factors had to be considered in the design of the idler location and pivot arm. First the idler had to be located far enough above the line between the driving and driven sprockets, to obtain a fairly large downward force when the cylinder was loaded. Furthermore, when the idler is located at a greater distance above the line of the two sprockets, there is a smaller change in the sine of the angle, for a given amount of vertical movement of the idler. A minimum change in the sine of the angle is desirable, since the downward force on the idler should be dependent on the chain tension rather than on the position of the idler. For these reasons, it would be desirable to have the idler located as high as possible above the center line of the driving and driven sprockets. However, physical limitations such as length of chain and support of the idler also had to be considered. Then too, the higher the idler, the fewer teeth are in contact with the chain on the driving and driven sprockets. Reducing the number of teeth in contact with the chain may cause excessive chain and sprocket wear. A compromise was
necessary therefore, and an angle near 40° was chosen. This caused a reduction in arc of contact between chain and sprockets of about 29%, which was considered tolerable. The vertical movement of the idler was to be kept to a minimum to reduce the effect of changes in the sine of the angle, with changes in idler position. The centre distance between driving and driven sprockets was 15 inches, and the vertical distance to the centre of the idler was 6.5 inches at mid-position of the pivot arm. An idler movement of 0.3 inches vertically on either side of mid-position was found to be satisfactory for valve operation through a linkage. This resulted in the sine of the angle changing from 0.638 to 0.671, or about 5%.

In order to reduce the size of spring required, and to obtain a larger displacement for operating the spool valve, the pivot arm was built so that the movement at the point of spring attachment was four times as large as the movement of the idler. A bell-crank arm was used to gain clearance from the main combine body and to allow more room for attaching the valve, spring and damper.

**Spring Design**

The spring force controls the amount of torque that the cylinder absorbs, and so an estimate of cylinder power required was necessary for the design of the spring. Burrough (1954) found that a 7 ft. pull-type combine would require up to 5½ hp at the cylinder, for a throughput of 80 lb./min. of wheat straw. Bigsby (1959) using a 12 ft. self-propelled combine in wheat, measured power requirements up to 10 hp, for a threshing rate of 200 bu./hr. (or approximately 200 lb./min. throughput).
On the basis of these tests, an upper limit of 10 hp seemed reasonable, since the M-F 82 is a nominal 12 ft. combine. A range of horsepowers between 5 and 10 hp was chosen for the spring design.

![Figure 1](image)

The spring force is calculated as follows:

\[ F = \frac{2T \sin \theta}{4} = \frac{T \sin 40.9^\circ}{2} = 0.327T \quad \ldots \ldots \ldots \quad (2) \]

\[ L = \text{torque} = \frac{Td}{2} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (3) \]

\[ \text{HP} = \frac{2mLN}{33000} = \frac{mTdN}{33000} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (4) \]

\[ T = 3.06F \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (2a) \]

\[ \text{HP} = \frac{3.06mFdN}{33000} = \frac{FdN}{3430} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (4a) \]

\[ F = \frac{3430\text{HP}}{dN} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (5) \]
\[ F = \frac{3430 \text{ HP}}{920 \times 0.429} = (8.70) \text{(HP)} \quad \ldots \ldots \ldots \ldots \ldots \ldots (5a) \]

where \( F \) = spring force, lb., \( T \) = chain tension, lb.

\( L = \) torque on driving sprocket, ft. lb.

\( d = \) diameter of driving sprocket = 0.429 ft.

\( N = \) rpm of driving sprocket = 920 rpm

\( \text{HP} = \) cylinder hp

Thus for an operating range of 5 - 10 hp the spring force must have a range of 43.5 - 87.0 lb. at mid-position of the pivot arm.

The maximum spring displacement at any one initial tension setting was 2.4 inches between the mechanical stops, so the spring had to stretch 1.2 inches from the no load position before the mid-position of the pivot arm was reached. A spring with an initial preload was considered desirable to reduce the total length of the spring required, and thus make the system more compact. The spring selected had an initial preload of 22.4 lb. and a spring rate of 18 lb./in. Therefore, the minimum operating load for linear operation was obtained at a spring force of 22.4 + (1.2 \times 18.0) = 44.0 lb. This corresponds to 44.0 or 5.06 hp (Eqn. 5a). The highest point of operation (10 hp) would be obtained by stretching the spring 3.6 inches for mid pivot arm position, or 3.6 - 1.2 = 2.4 inches with the pivot arm against the no load stop. The free coil length of the spring was 14 inches. Increments of loading were determined by measuring the length of the spring body. Length adjustments were made by means of eyebolts on the pivot arm and on the stationary support. A larger spring was also obtained which could be used in case higher cylinder loads were desired.
Damper

The viscous damper was constructed by using a hydraulic cylinder and connecting the two ports together with a pipe. A valve was included in the pipe, to allow regulation of the oil flow from one side of the piston to another. By adjusting the valve, the amount of damping could be changed. The damper was filled with oil, and connected in parallel with the spring, between the pivot arm and the stationary support.

Hydraulic System

The hydraulic pump and valve originally used for variable speed control, were not used in the automatic control system, because it was not possible to get at the inlet and outlet ports of the pump and valve. A power steering pump, which could easily be mounted on the combine was substituted, and another control valve mounted near the pump. This control valve was connected to the manual control lever which previously operated the original valve. Copper tubing of 3/8" o.d. was used for pressure lines and connected by means of flared fittings, to make assembly and disassembly convenient. A pressure release valve set at 200 psi was mounted in the line to prevent damage to the system. The control valve to be operated by the pivot arm was mounted adjacent to the pivot arm to reduce the linkage required. Both control valves were 20 gpm Gresen 4-way spool valves commonly used on farm equipment. The pump had a capacity of 2 gpm, compared to 0.5 gpm for the original pump. A metering valve was used in the automatic system to reduce the flow and to allow for rate of response adjustment. Two shut-off valves and a 3-way selector valve were
used to enable the variable speed cylinder to be operated either manually, or automatically by the pivot arm. The three valves were inter-connected so that the change from manual to automatic and vice versa could be accomplished by moving one lever. The flow diagram of the hydraulic system is shown in Fig. 7.

**Bottom Idler**

A second idler was required on the slack side of the cylinder chain to maintain chain tension. To be the most effective it was desirable that this idler move up and down together with the top idler. To accomplish this, the bottom idler was mounted on a pivot arm anchored to the combine frame. A direct connection was made between the shafts of the top idler sprocket and the bottom idler sprocket with a turnbuckle as a part of this link. (Fig. 6). It was then possible to adjust the initial chain tension by regulating the distance between the sprockets with the turnbuckle. The bottom idler was mounted at approximately the same distance below the driving and driven sprockets as the top idler was above them. This minimized the variation in chain tension due to changes in idler position.
6. LABORATORY TESTS

The laboratory tests on the system were carried out in the Agricultural Engineering tractor laboratory. The cylinder was loaded by means of a tractor pto dynamometer. Transducers were used to indicate the position of the pivot arm and of the variable speed hydraulic cylinder. A magnetic speed pickup and torquemeter were also used. All measurements were recorded by an oscillograph. The variables recorded were cylinder torque, cylinder RPM, variable speed position, and hydraulic valve position (Figs. 8, 9 and 22).

Laboratory tests were mainly of an exploratory nature. Field conditions were difficult to replicate in the laboratory, and so an accurate knowledge of the system response was not obtained from these tests. Cylinder loading may be of many different forms in the field, and variations are not always of the type that can be simulated in the laboratory. Also, the momentum of the combine cannot be considered in laboratory tests, and this is of significant importance in the actual speed changes.

In spite of these limitations, however, some information was obtained from the tests. An investigation of the system response was carried out and results recorded on an oscillograph. A strain gauge position transducer was connected to the pivot arm to indicate its position and the position of the automatic control valve spool. A strain gauge cantilever was also connected to the indicating rod of the variable speed hydraulic cylinder. This indicated the position of the variable speed sheave, and thus the speed ratio of the combine traction drive. As the pivot arm was moved back and forth, the positions of the arm and cylinder were simultaneously recorded. The response rate was adjusted by means of the metering valve.
Observations were as follows:

1. The metering valve provided a satisfactory means of adjustment of the response rate. A wide range of adjustment was possible since oil flow rate to the hydraulic cylinder was reduced rapidly as the metering valve was closed. As the metering valve is throttled, an increased proportion of the oil supply is by-passed through the pressure release valve.

2. The acceleration rate of the combine was approximately equal to the deceleration rate. This was different from the original system on the combine, in which the deceleration was four times the magnitude of the acceleration. The ratio of exposed piston areas is 4 to 1 in the hydraulic cylinder, and therefore if the pressure release valve does not open, the piston will move four times as fast in one direction as in the other, when using a constant displacement pump. However, in the automatic system the release valve was set at 200 psi, (as compared to 250 - 350 psi on the original system) and the pressure drop in the line was much higher due to more line length (65 feet as compared to 10 feet) and several more valves, including the metering valve. Also, the oil flow was increased from 0.5 gpm to 2.0 gpm. This caused the pressure release valve to open on the deceleration stroke (small piston area exposed) and allowed some of the oil to return to the sump rather than enter the cylinder. The times required for the opposing strokes were equalized since the pressure release valve was not forced open on the acceleration stroke since a large force could be exerted for
the same pressure. Some variation in relative rates of speed change were noted and these can be attributed to variations in resistance to piston movement, and also perhaps some variation in the operation of the pressure release valve. If a higher release valve pressure had been used, the response rates could have been brought back to the original 4:1 ratio but it was considered desirable to have them approximately equal. In manual control a higher rate of deceleration is desired, to allow a quick reduction of speed in emergencies. However, in automatic control there should be less tendency for these emergencies (cylinder plugging) to occur, due to corrective changes being made continuously. This would decrease the chances of overloading the cylinder to the point where it would plug. If necessary the system is operated in manual control, and a quick deceleration is still possible due to the absence of a metering valve in the manual hydraulic circuit.

3. "Creep" of the piston was noticed at a certain valve position. This occurred when the valve spool was near the acceleration end of the dead zone, and caused the piston to move slowly in the deceleration direction. When the spool was moved very slightly, the "creep" would stop, or else the piston would move in the acceleration direction, depending on the direction of spool movement. This was attributed to the fact that the return port of the valve opened before the pressure port opened, and thus allowed the oil from the cylinder to return to the sump. This "creep" was
not considered to be important, since it occurred only at one position and it was not expected that this position would be maintained for long. This assumption was borne out later in field tests, where this "creep" was never noticed.

4. The dead zone of the system appeared to be too large. The large size of the dead zone was due to loose fitting links and the spool movement required for actuation. The dead zone was reduced for field tests by changing the linkage.

Unfortunately, the "no load" setting of the dynamometer actually corresponded to about 15 hp and so the spring chosen for field tests could not be used in the laboratory tests. A heavier spring was therefore substituted. Full engine power was applied to the cylinder drive without any failures in the mechanical load sensor. Furthermore, there were no excessive vibrations evident and so the system was considered to be structurally adequate for field tests.
7. FIELD TESTS

The automatic control system was mounted on the Standard combine, a M-F 82 owned by the Agricultural Machinery Administration, (A.M.A.) Regina. Field tests were carried out in cooperation with A.M.A., using their combine test equipment. The combine with the control system was used in rye, wheat, oats and barley crops in the Arcola, Sask. area.

The tests conducted were of two types: (1) throughput and loss measurements, and (2) comparisons between manual and automatic operation. Observations were made on the general operation of the system in an effort to determine the most desirable response of the system, and the adjustments required to obtain this response.

Throughput and loss measurements were similar to those used in the functional testing of combines by A.M.A. to evaluate capacity and loss characteristics. The test procedure briefly is as follows: The combine is loaded to the desired level to establish equilibrium with respect to crop intake, and straw and grain output. When this uniform condition has been established, two large canvas bags are placed under the ends of the straw walkers and the shoe. The straw and grain from the walkers is thus caught separately from the chaff and grain from the shoe. At the same time, the grain entering the tank is caught in another bag. When any one of the bags is filled, a signal is given by one of the technicians and the bags are all removed. The time of the run is noted and the distance is measured. The contents of the bags are weighed and all measurements recorded, along with comments on general crop and field conditions and test irregularities, if any. The contents of the shoe bag are put
through a "batch separator" to remove any loose grain from the shoe efflux. The batch separator is a converted combine that allows material to be fed through the cylinder and then over the walkers, or else over the walkers only. Bags for catching the grain and straw are attached to the appropriate outlets. The loose grain is collected and weighed to determine the amount of shoe loss. A similar operation is performed on the contents of the walker bag to determine walker loss. In both cases the material is only put over the walkers of the batch separator. After the loose grain has been removed, all the straw and chaff is then put through the cylinder of the batch separator, to determine the cylinder loss of the test combine. These runs were repeated at different spring settings to determine the spring setting-throughput and spring setting-loss relationships.

The main object of the field tests was to determine whether or not the automatic control system was able to control the throughput, and therefore the losses of the combine within the limits of the dead zone of the system. Six full scale field tests were conducted in different fields and under varied conditions. Four tests were carried out in wheat, and one each in oats and barley. The device was also used in several other fields to assist in qualitative evaluation of the control system.

Three different spring settings of 14, 14.5 and 15 inches were used in the tests. These corresponded to 5.06, 6.09 and 7.13 cylinder hp respectively. Higher values were not used since throughputs near the optimum level as determined by loss were desired for the tests. The graphs of throughput versus spring setting are shown in Figs. 10 to 15. Two graphs
of loss versus spring setting are shown in Figs. 17 and 18. The other tests did not show significant loss increases over the range of spring settings used, due to the relatively low throughputs for the particular crop condition. Total grain losses are plotted on the graphs. A breakdown into shoe, cylinder and walker losses, showed the walker losses to be the most severe at high throughputs.

An overall comparison of automatic and manual control was difficult since the performance in manual control will depend on the particular operator chosen. However, an effort was made to operate the combine manually in a typical manner when comparisons were made.

Using the torque level of the cylinder drive as an indication of the throughput, a comparison was made between automatic and manual control. Since the position of the pivot arm gives an indication of the torque, this was used to provide the comparison. A cantilever with strain gauges mounted on it was connected to the pivot arm, and the deflection of the beam then indicated the position of the pivot arm. The deflection of the cantilever beam was recorded on a single channel recorder which was mounted on top of the combine. Two six volt batteries provided the potential for the strain gauge bridge, and the recorder was driven from the combine battery, through an inverter (Fig. 19).

A spring setting of 14.5 inches was used for the comparisons, both for automatic and manual control. For the manual control runs, the automatic control valve lever was disconnected from the pivot arm, and the stops were removed to allow for a wider range of torque values.
Recordings were made in two fields of wheat and one of barley. In all cases, the test runs in manual and automatic control were taken in adjacent windrows and in immediate succession.

Evaluation of various components of the control system was accomplished by operating the control device over an extended period of time in various conditions. No mechanical failures or wear were detected during the test. The metering valve was found to be very useful for adjusting the response rate for different conditions.

It was estimated that the dead zone of the valve should be reduced to about 1/2 of the present value, or to about 1 hp. This estimate was based on personal judgment, but critical observations in many different fields seemed to produce similar estimates. With the combine operating at 6 or 7 hp, a dead zone of 2.4 hp accounts for a fairly large percentage of the total range, especially when the "no load" power requirement (2 hp) (Eqn. 6) is considered. A brief look at the graphs (Figs. 10 to 15) also serves to indicate that a larger variation in throughput is possible due to the dead zone, than would generally be considered desirable. With a smaller dead zone, it would be necessary to reduce the response rate in order to prevent "hunting".

When an abrupt change in the volume of the windrow occurred, it took 3.5 seconds from the time the pickup encountered the change in the windrow, until the control system changed the forward speed of the combine. This corresponds to about 15 ft. of travel at 3 mph.
8. DISCUSSION OF RESULTS

General Observations

Operation of the control system under various field conditions resulted in several observations being made concerning the requirements for the successful operation of the system.

The rate of response had to be adjusted for optimum operation under varying crop conditions. This was easily accomplished by means of the metering valve, and after a little experience, a very satisfactory response rate could be achieved. However, if the windrow was "bunchy" it was not possible to achieve a satisfactory response. This only occurred in a small part of one field of rye, where it was then necessary to revert to manual control. On long open corners the combine reached maximum forward speed in the gear being used, and under some conditions this was not desirable. It was then necessary to switch back to manual before reaching the corner, and switch back to automatic after the cylinder was loaded again. On very light windrows, pickup performance is the limiting factor, with regard to maximum speed, and so a switch back to manual or to a lower spring setting is necessary. The optimum response rate depends on the gear being used, the size of the windrow, and the size of the dead zone. In a higher gear, the absolute speed change is greater for a given amount of correction, and so a lower system response rate is desired. If the windrow is very large and heavy, and the pickup tends to pull it in by bunches, a lower response rate is necessary to prevent hunting. As the size of the dead zone is decreased, the response rate must also be decreased in order to prevent hunting.
When the end of a speed range is reached, and further speed correction is desired, it is necessary to shift gears. When shifting to a lower gear, the same procedure is followed as in manual control. As the load on the cylinder is reduced due to stopping the forward travel, the variable speed ratio will increase, and when travel begins again after shifting gears, the variable speed will be at or near the top of the range, which will be near the speed desired. When shifting to a higher gear, the hydraulic system must be switched back to manual control and the variable speed brought to the bottom of the range. The system is switched back to automatic when the cylinder is loaded. The shifting procedure is the same as in manual control, except for the switching from automatic to manual and back to automatic. However, both switches may be performed while the combine is moving and so no time is wasted.

The automatic control system made speed corrections much more frequently than any operator would. It was noticed in observing twenty different operators, that changes in forward speed were made erratically and often not at all. Many factors were observed to have an influence on the speed changes made by these operators. Some of these were the skill of the operator, the time of day, weather conditions, dust, and the operator's state of mind. Very often the last four factors had more influence than the first one, and many "skilled" operators were guilty of allowing extremely wide ranges of crop throughputs. Most combines have sufficient reserve engine power to allow large amounts of material through the cylinder and thus overload the separating components quite seriously. In addition, the engine and machine noise (100 db) make small load changes
very difficult to detect aurally. Visual judgment is often impaired by lodged crops, wind tossed or settled windrows, dust and darkness. Most grain harvesting in Western Canada is carried out under dusty conditions, and a large percentage is done at night as well. These two factors are probably the most serious and often make throughput control a guessing game. Automatic control is not affected by these external factors and thus is much more satisfactory.

After prolonged operation with automatic control, manual control seemed bothersome. As mental and physical fatigue increase, the operator is prone to allow greater variations in throughputs in both directions from the optimum.

**Throughput Tests**

The results of the throughput tests are shown in Figs. 10 to 15. The feed rates are plotted as a function of spring setting. Three spring settings were used for the tests. These are designated by the initial lengths of the spring, which were 14, 14.5 and 15 inches. The corresponding cylinder horsepower for these settings are 5.06, 6.09 and 7.13 hp when the front beater shaft (driving sprocket) speed is 920 rpm (Eqn. 5). The dead zone of the system covers a range of 2.4 hp, which means that the system can operate at equilibrium at 1.2 hp on either side of the above values. The size of the dead zone was determined by measuring the amount of pivot arm movement required at the point of spring attachment, to move the spool valve through the neutral position. This distance, multiplied by the spring rate results in the force dead zone, and by
applying Eqn. 5a, the horsepower dead zone may be calculated. The dead zone shown on the throughput scale is calculated from the horsepower dead zone and the slope of the least squares line. As an example, in Series 1, (Fig. 10) a throughput increase of 47 lb./min. occurs in 2.07 hp. The throughput dead zone is therefore

\[
\frac{47 \times 2.4}{2.07} = 54.5 \text{ lb./min.}
\]

One half of the dead zone is shown on either side of the least squares line.

In all cases, all the points fall within the dead zone, and do not even approach the limits of the dead zone. This was expected, since there was a slight averaging effect in each of the runs. This occurred due to speed corrections being made in both directions during the course of the run.

Equal spring settings did not produce equal throughputs in all the series. Large differences in throughputs for a given setting can be noted by comparing series 4, 5 and 6 (Figs. 13, 14 and 15). These differences can be attributed to variations in straw condition and concave settings. In series 4 and 6, with very dry straw, more material passed through the cylinder for a given power input. As a result, the average throughputs are displaced upwards on the graphs. In series 5, the straw was slightly tough, thus lowering the throughput for any given torque setting. Straw and grain conditions are noted on each graph.

By combining the results of all the tests and extrapolating to zero throughput, the overall relationship between power and throughput
was obtained. The equation for this relationship based on a graphical least squares analysis is:

\[ HP = 2.03 + 0.0455T \quad \ldots \quad (6) \]

where \( HP \) is the cylinder horsepower and \( T \) the throughput in lb./min.

This equation indicates an average power requirement increase of 1 hp for every increase of 22 lb./min. in throughput. The size of the throughput dead zone is therefore 53 lb./min.

In Fig. 16 all the test data are plotted on a single graph. The dead zone is again indicated on either side of the average line. Two of the test points fall outside the dead zone, this is due to the wide variation in straw moisture content and concave settings. The purpose of the graph is to illustrate the variation in throughput under various crop conditions. If very tough straw was encountered, the feed rates would drop to lower levels than those indicated, thus increasing the overall variation. The reasons for decreased throughputs with increased straw moisture are less straw breakage and a closer concave setting which is required to thresh out the tough grain.

The change in throughput for a given change in horsepower is less in tough crop conditions. This results in a smaller throughput dead zone if the horsepower dead zone is unchanged.

**Loss Tests**

Two illustrations of the grain loss vs spring setting relationship are shown in Figs. 17 and 18. These graphs indicate sharply increasing losses with increasing load level. This relationship was expected, since
previous tests (Sec. 1) had consistently given similar results (Figs. 2 to 4). Loss-throughput runs were taken in manual control at higher feed rates than those indicated, and correspondingly higher losses were obtained.

The loss curves from the other four series are not shown, since the throughputs were too low to show significant increases in grain loss. Higher feed rates were tested under manual control, and again the losses reached high values as maximum capacity was approached. In series 5 (barley) losses reached 17% and in series 6 (wheat) losses reached 11% at maximum throughput.

With sufficient proof of loss increases with increases in throughput available, (Sec. 1), the main purpose of the field tests was to determine whether the control system could accurately control the throughput of the combine under various conditions.

**Comparison of Automatic and Manual Control**

The results of tests comparing the operation of the combine under automatic and manual control are shown in Fig. 20. During manual operation the automatic control valve lever was disconnected from the pivot arm (page 41). The dead zone indicated on the chart sections (Fig. 20) was obtained from a static calibration. The width of this zone cannot be considered accurate for the dynamic case because the slackness of the valve control linkage allowed the valve to move further than during the static calibration, due to impulse from the pivot arm. The continuous motion of the pivot arm (as shown by motion pictures) caused the control valve spool
to be in constant oscillation. Hence the actual dead zone during operation would be considerably narrower than that indicated on the chart.

In the section of automatic control chart (Fig. 20), it appears that four corrections were made, two at low torque and two at high torque, if it is assumed that the "dynamic" dead zone was appreciably less than the static dead zone. Unfortunately, since no measurement was made of the dynamic dead zone, a complete analysis of operation from the charts is impossible. It is suggested that in figure tests, additional instrumentation be incorporated to measure and record automatic spool valve position (and therefore dynamic dead zone), combine ground speed, and the time interval during which the manual control valve is actuated.

The author was the combine operator for the comparison tests. Initially it was considered preferable to have various farmers operating the combine in manual control, but it was difficult to find a farmer who was familiar with the particular combine. Being unfamiliar with a combine almost invariably causes a farmer to be more conscious of the feed rate than he would normally be. For these reasons the author decided to operate the combine for the manual runs. A visual survey of many operators was made to determine the operator response to changes in crop density. The amount of change necessary to make the operator correct the speed, and other factors affecting speed changes were observed. An effort was then made to operate the combine in a similar manner for the comparison.

While operating in various fields together with other combines, a visual check was made to determine whether forward speed changes were
significantly different when automatic or manual control was used. It was immediately evident that under manual operation, the operator attempts to maintain a relatively constant speed around the field. When a light spot is encountered, the operator may or may not increase speed, depending to some extent on how long the lighter windrow exists. However, the increase in speed is very rarely as large in manual as in automatic control. When a light spot was encountered, the combine in automatic control would gain a considerable distance on the other combines, but in the heavy areas, the other combines would gain distance on the automatically controlled combine. It has been shown by Nyborg (1963) that with an increased grain/straw ratio, the losses are reduced, for a given straw throughput. Ordinarily the grain/straw ratio is higher in the lighter crop areas of the field. A higher feed rate is therefore desired. The higher feed rate obtained with automatic control results in greater efficiency in terms of time use.

When a heavy spot in the windrow occurs, the reduction in forward speed is not as marked under manual as under automatic control, resulting in high grain losses. The decrease in grain/straw commonly accompanying these heavier areas further increases the percentage of the grain loss.

The total time required to harvest a field may or may not be reduced by automatic control, depending on the relative size of the areas of light and heavy crop and on the load setting of the control system. However, by maintaining a uniform throughput, the losses will definitely be reduced in automatic control as compared with manual control.
Moisture Content

Operation under a variety of crop moisture conditions makes it necessary to adjust concave clearances for optimum threshing efficiency. A closer concave setting is used for tough conditions and the clearance is increased as the crop dries. Therefore, for a given torque setting of the control system, the throughput would increase with decreasing moisture content, due to increased clearance and increased straw breakage. An investigation by Johnson (1959) on the effects of moisture content on grain loss from combines, shows a sharply decreasing loss with decreasing moisture content for a cylinder concave clearance of 1/2 inch and a relatively small change in loss for a 1/4 inch clearance. Depending then on the clearance used, it may or may not be desirable to allow increased throughputs with reduction of moisture content. However, if it is desirable to keep the throughput constant throughout the day, which would most often be the case, the torque level should be adjusted at the same time as the concave clearance is changed. This is a very simple adjustment and it could even be arranged to have it adjustable from the operator's platform. To decrease the throughput it is only necessary to reduce the initial tension on the spring by means of an adjusting screw.

Loss-Horsepower Relationships

A mathematical relationship between cylinder horsepower and grain loss may be derived from the results of the preceding tests, and from other tests conducted by the A.M.A. on the MF-82 combine. By combining empirical equations for walker loss regressions at various grain/straw
ratios, as a function of feed rate, and equations for walker loss as a function of grain/straw ratio, an equation for loss as a function of feed rate and grain/straw ratio was derived by Nyborg (1963). This equation is the result of many tests made throughout a range of feed rates from 0 to 300 lb./min. and grain/straw ratios from 0.8 to 1.7. The following walker loss equation for the MF-82 combine in wheat is given:

$$\text{Walker loss (\%)} = 10^{-5.29} \cdot (\text{feed rate})^{2.39} \cdot (\text{grain/straw})^{13.26 - 7.48 \log(\text{feed rate})} \ldots (7)$$

Combining Eqn. (7) with Eqn. (6) (page 47) the following equation for walker loss as a function of horsepower is obtained:

$$\text{Walker loss (\%)} = 10^{-5.29} \cdot [22(\text{HP}) - 45]^{2.39} \cdot (\text{grain/straw})^{13.26 - 7.48 \log[22(\text{HP}) - 45]} \ldots (8)$$

For a grain/straw ratio of 1, the equation reduces to:

$$\text{Walker loss (\%)} = \left[136(\text{HP}) - .275\right]^{2.39} \ldots \ldots \ldots (8a)$$

Under normal cylinder load, the walker loss is approximately 75% of the total loss with the concaves and shoe properly adjusted.

The maximum throughput for the MF-82 is approximately 300 lb./min. By using this value in Eqn. (6), (page 47) the maximum cylinder hp is found to be about 16. When this value is used in Eqn. (8a), a maximum walker loss of 4.7% for a grain/straw ratio of 1 is obtained. If the grain/straw ratio is 0.8, the maximum walker loss is approximately 15%, by Eqn. (8).
A plot of Eqn. (8a) is shown in Fig. 21. The dead zone of 2.4 hp is indicated on the graph. The loss dead zone may be obtained by drawing a horizontal line between the dashed lines at the desired hp, and drawing vertical lines from the intersections of the horizontal line and the dashed lines to the line of the equation. The loss dead zone is the difference in ordinate values between the two points thus obtained on the line of the equation. The losses and the size of the loss dead zone increase sharply with increase in horsepower. The size of the loss dead zone is quite large at high horsepowers, and with a lower grain/straw ratio it would be still larger. A reduction in the size of the horsepower dead zone would therefore be desirable, as suggested previously.
9. COST ANALYSIS

The actual cost of materials for the control system was approximately $200. A large part of the cost was attributed to the new hydraulic system, which had to be employed due to the inaccessibility of the original system. If the control system were built by the original combine manufacturer, it would be possible to reduce the cost of the hydraulic system by about $75. Several extra springs were also purchased, which increased the cost by $20. The mechanical supports near the cylinder drive could be simplified if the control system were built by the original manufacturer of the combine.

An estimate of the cost of components as supplied by the original combine manufacturer is:

<table>
<thead>
<tr>
<th>Component</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>$15</td>
</tr>
<tr>
<td>Idler</td>
<td>10</td>
</tr>
<tr>
<td>Hydraulic valves</td>
<td>35</td>
</tr>
<tr>
<td>Hydraulic lines</td>
<td>30</td>
</tr>
<tr>
<td>Damper</td>
<td>5</td>
</tr>
<tr>
<td>Spring</td>
<td>5</td>
</tr>
</tbody>
</table>

**Total** $100

Assembling of the control system is not expensive since there are no complex parts involved.

Assuming that the unit would retail for $200 - $300, and through its use a grain saving of 1% (Nastenko 1959) were realized, the investment could be repaid in a week's operation. An increase in total area harvested should also result from the use of the control system (Nastenko 1959).
10. CONCLUSIONS

1. The relationship between cylinder horsepower (HP) and feed rate (T) for the Massey Ferguson 82 combine was found to be:
   \[ HP = 2.03 + 0.0455T \]
   for a throughput from approximately 50 to 140 lb./min.

2. The system was able to control the feed rate, and therefore the grain losses from the combine, within the limits of the system characteristics. The response rate, dead zone, and time delay of the system were compatible with all normal types of windrows, and thus allowed the feed rate to be controlled smoothly at the preset level. All adjustments necessary for satisfactory operation were made easily and quickly.

3. The control system was able to maintain a more uniform feed rate than a typical operator does. There was less deviation from the average feed rate, both in terms of magnitude and time.

4. It is believed that the control system will reduce operator fatigue, once the operator becomes familiar with the system. The mental effort required to constantly judge the feed rate and determine the magnitude of correction necessary is eliminated. The physical effort involved in changing the speed is also eliminated.

5. The combine operations and adjustments were not impeded by the control system.

6. The system is low in cost relative to its potential value.
11. SUGGESTIONS FOR FURTHER STUDY

1. Further work should be carried out in the application of a control system similar to the one described, to other agricultural or industrial machines, where a constant load is desired, and some means of adjusting the feed rate during operation is provided.

2. A control system with the electrical circuit described in Appendix C substituted for the hydraulic circuit, should be tested under field conditions.

3. The hydraulic or electrical circuits should be connected to other sensing devices, and used to control variables other than speed, such as height of cut.

4. It is possible that in the future, tractors will be provided with variable speed drives (hydrostatic or other means) and this would make it possible to use a similar control system for pull-type implements. The sensor would be on the implement and the tractor speed would be varied by hydraulic or electrical means. A load sensing device of the type described could be used for a P.T.O. combine feed rate control.
12. LITERATURE CITED


FIG. 2 LOSS CURVES FOR AMS-10 IN BARLEY

( NYBOG 1963 )
FIELD 21
WINTER WHEAT, VAR. HYBRID 46

FIG. 3 LOSS CURVES FOR FOUR COMBINES IN WHEAT (PHILLIPSON 1962)
FIG. 4 LOSS FOR M-F COMBINE (MARK 1963)
Legend:

- **x** - SOLID STEMMED WHEAT  \( \text{H.P.} = 1.59 + 0.0439 \text{ RATE} \)
- **o** - HOLLOW  \( \text{H.P.} = 1.35 + 0.0368 \text{ RATE} \)

**FIG. 5** POWER REQUIREMENTS FOR MM-S COMBINE (BIGSBY 1959)
FIG. 6  TORQUE SENSING DEVICE
FIG. 7  HYDRAULIC CONTROL SYSTEM
FIG. 8 LABORATORY TEST EQUIPMENT
Figure 9  Chart Section from Laboratory Test
Series I

SELKIRK WHEAT
STRAW - AVERAGE M.C.
GRAIN - 13.3% M.C.

FIG. 10  THROUGHPUT VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL.
FIG. 11 THROUGHPUT VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL.
**Series 3**

- OATS
- STRAW - AVERAGE M.C.
- GRAIN - 14.5% M.C.

**FIG. 12** THROUGHPUT VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL.
Series 4
SELKIRK WHEAT
STRAW - DRY
GRAIN - 13.5 % M.C.

FIG. 13 THROUGHPUT VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL.
FIG. 14  THROUGHPUT VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL.

Series 5
BARLEY
STRAW - SLIGHTLY TOUGH
GRAIN - 13.5 % M.C.

THROUGHPUT - lb./min.

SPRING SETTING - INCHES

0  14  14.5  15
Series 6

LEE WHEAT
STRAW - DRY
GRAIN - 10.8 % M.C.

FIG. 15 THROUGHPUT VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL.
FIG. 16 CYLINDER HORSEPOWER VS. THROUGHPUT FOR AMS-10 COMBINE IN WHEAT, OATS AND BARLEY.
SELKIRK WHEAT
STRAW — AVERAGE M.C.
GRAIN — 15.0 % M.C.

FIG. 17 GRAIN LOSS VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL.
FIG. 18 GRAIN LOSS VS. SPRING SETTING FOR AMS-10 COMBINE UNDER AUTOMATIC CONTROL

OATS
STRAW - AVERAGE M.C.
GRAIN - 14.5 % M.C.

SPRING SETTING - INCHES
GRAIN LOSS - %
FIG. 19  FIELD RECORDER CIRCUIT

STRAIN GAUGES ON CANTILEVER

RHEOSTAT

6V D.C.

POTENTIOMETER

SWITCH

VARIAN (GI4) RECORDER

INVERTER

6V D.C. BATTERY
Figure 20 Chart Sections from Field Tests
LOSS = \left[ 0.136 \ (\text{HP}) - 0.275 \right]^{2.39}

FIG. 21 WALKER LOSS VS. CYLINDER HORSEPOWER

M-F 82 COMBINE
IN WHEAT
GRAIN / STRAW = 1
Fig. 22 Photograph of Laboratory Equipment

Fig. 23 Pivot Arm Mounting
Fig. 24 Right Side View of Torque Sensor
Fig. 25 Front View of Control System

Fig. 26 Rear View of Control System
Appendix A

Vocabulary

Combine: a grain harvesting machine which cuts or picks up the crop and threshes it, all in one operation.
Cylinder: the threshing drum of the combine which knocks the grain from the heads.
Dead zone: a zone or position of a control system in which there is an input but no output.
Error: the difference between the desired output and the actual output.
Field efficiency: ratio of effective field capacity to theoretical field capacity.
Feed rate: see throughput.
Grain loss: the grain expelled from the rear of the combine, expressed as a percentage of the total grain entering the combine.
Grain/straw ratio: the ratio of the weight of grain to the weight of straw and chaff in the crop.
Hunting: cycling of the output above and below the desired output level.
Linear system: a system in which the output is directly and continuously related to the input.
Response rate: the rate at which the output is changed in relation to a given error signal.
Shoe: the mechanism in a combine which separates the grain from the chaff.
Throughput: the weight of straw and chaff passing through the combine in a given time, commonly measured in lb./min. Also referred to as the feed rate.
Walkers: the mechanism in a combine which separates the grain and chaff from the straw.
Introduction

This appendix will outline the characteristics of the various components of the control system. A discussion of the response of each element precedes and provides the basis for the analysis of the element. The block diagram (Fig. B-1) illustrates the relationship of the elements to each other, and shows the inputs and outputs of these elements. Initially, the input to the system comes from the block in the upper left hand corner, and from then on the system follows the closed loop.

The analysis presents the derivations of the transfer functions for the linear elements and the describing functions for the non-linear elements. The purpose of the derivations is to provide the necessary information for a complete closed loop analysis of the system. The block representing the windrow is left in doubt and if a closed loop analysis was to be carried out, assumptions would have to be made regarding this block. The complete analysis is not presented due to the complexity of the system. However, if a closed loop analysis were carried out it could be done in two ways - analytically or by computer simulation. In a mathematical analysis, all the non-linear elements would have to be grouped into one non-linear element, since there are no linear elements between them to filter out harmonics. For an analogue computer simulation, the describing functions could be used directly as derived in this dissertation.
FIG. B-1 BLOCK DIAGRAM OF CONTROL SYSTEM
Cylinder and Chain

The amount of power required to turn the combine cylinder depends on the wind pumping effect of the cylinder, friction, and the amount of material passing between the cylinder and the concave. An investigation made by Downing (1942) on two pull-type combines indicates a linear relationship between total material throughput and power required to operate the combine drives, after the initial no load resistance was overcome. On both combines there was an increase of 1 hp required for each additional 42 pounds of material (grain and straw) passing through the cylinder.

Burrough (1954) conducted tests in wheat and soybeans to determine the power requirements of various combine drives and how throughput affects these power requirements. He found that for soybeans, the power required at the cylinder increased linearly with throughput. However, in wheat the power required increased more quickly than the throughput. The tests were carried out using a 5 foot rasp bar cylinder and a 7 foot cutting width.

A test at Swift Current by Bigsby (1959) on cylinder power requirements indicates a linear power increase with threshing rate (bus./hr.) in two varieties of wheat (Fig. 5). A self propelled combine picking up a 12 foot swath was used for the tests. A linear increase from about 4 hp at 60 bus./hr. to approximately 9 hp at 200 bus./hr. was noted.

On the basis of these tests, the first block in the control system can be considered to be linear, above the no load point. The non-linearity found by Burrough for wheat is not significant when a small
range of throughputs is considered, since the relationship can be approximated by a straight line with very little error. The range of throughputs to be considered at any one setting will be small due to saturation in other parts of the system (i.e. hydraulic system), and so the force exerted on the cylinder drive chain can be taken as a direct indication of the throughput. The expression for the cylinder and chain can then be given as:

\[
\frac{F(t)}{T(t)} = \frac{a}{T(t)} + b
\]

where

- \(F(t)\) = force exerted by the chain
- \(T(t)\) = throughput (lb./min. of straw)
- \(a\) = a constant depending on the no load force exerted on the chain
- \(b\) = a constant depending on the rate of increase of force with throughput

**Spring, Mass and Damper**

The force exerted by the chain is transmitted to the spring-mass-damping system, and causes a displacement of the system. The amount of displacement \(x(t)\) depends on the force \(F(t)\), the mass \(M\), the coefficient of viscous friction \(f_v\), and the spring rate \(K\).

The relation is then:

\[
M \frac{d^2x(t)}{dt^2} + f_v \frac{dx(t)}{dt} + Kx(t) = F(t)
\]
Using the Laplace transform, where \( \frac{d}{dt} = S \) and \( \frac{d^2}{dt^2} = S^2 \)

\[
(MS^2 + f_vS + K) x(S) = F(S)
\]

The transfer function \( x(S) \) then becomes \( \frac{1}{MS^2 + f_vS + K} \)

This can be reduced to the form

\[
x(S) = \frac{K_1}{F(S)} \frac{1}{(1 + ST_1)(1 + ST_2)}
\]

where \( T_1 \) and \( T_2 \) are the time constants of the system.

\[
K_1 = \text{a constant} = \frac{1}{K}
\]

\[
T_1 = \frac{f_v + \sqrt{(f_v)^2 - 4M}}{K}
\]

\[
T_2 = \frac{f_v - \sqrt{(f_v)^2 - 4M}}{K}
\]

Hydraulic Control Valve

The displacement of the spring-mass-damping system is transmitted to the hydraulic control valve, and thereby regulates the position of the spool valve. Due to some slack in the linkage and the spool movement required to open the ports, there is a dead zone in the system. If the valve capacity (gals./min.) exceeds that of the pump, there will be saturation near the ends of the spool stroke. Mechanical stops are provided at the ends of the stroke to prevent damage to the valve linkage. When the pivot arm reaches a stop, the system becomes saturated.

In the following diagrams \( r(t) \) designates the input quantity as a function of time, and \( c(t) \) is the output variable.
Diagram showing relationship of oil flow and valve position, and response of system to a sinusoidal input of $A \sin \omega t$, where $A > X$.

For a sinusoidal input,

$r(t) = A \sin \omega t \quad A > X$

c(t) = 0 for $0 < \omega t < a$ \quad (Savant 1958)

$= A \sin \omega t - a$ for $a_1 < \omega t < a_2$

$= h$ for $a_2 < \omega t < \pi / 2$

Note: Response diagrams show straight line approximations of sine curves
The describing function \( b_n/A \) may be evaluated by using the Fourier series expansion where
\[
b_n = \frac{4}{\pi} \int_0^{\pi/2} c(t) \sin n\omega t \, dt \quad \text{(Savant 1958)}
\]
with \( n = 1, 3, 5, \ldots \).

Only the sine terms appear since the output is an odd function, and only the odd harmonics appear due to the sinusoid being symmetrical about \( \pi/2 \) between 0 and \( \pi \).

**Hydraulic Cylinder and Release Valve**

The oil flow from the control valve is directed to the hydraulic cylinder, which moves one side of a variable pitch sheave to change the ratio of driving to driven sheave sizes and thus change the forward speed of the machine. A pressure release valve allows the oil to return to the sump if the pressure exceeds a preset value.

For purposes of analysis, the fluid compressibility and line compliance are assumed to be negligible. The force required to move the sheave is assumed to be proportional to the rate of movement of the sheave.

In the acceleration direction, the full piston area is exposed to the oil and thus a larger force can be attained in that direction, than in the deceleration direction, where the piston rod decreases the exposed area. The ratio of these two areas is often in the range of 4:1.
Figure B-3

o = opening of pressure release valve
n = ratio of piston areas exposed

Diagram showing relationship between force on piston and oil flow, and response of system to a sinusoidal input of $A \sin \omega t$, where $A > nb$. For a sinusoidal input

$$r(t) = A \sin \omega t$$

$$c(t) = A \sin \omega t$$

$= nb$ for $0 < \omega t < a_1$

$= nb$ for $a_1 < \omega t < a_2$

$= A \sin \omega t$ for $a_2 < \omega t < a_3$

$= b$ for $a_3 < \omega t < a_4$

$= A \sin \omega t$ for $a_4 < \omega t < 2\pi$
The describing function $b_n/A$ may be evaluated with

$$b_n = \frac{1}{\pi} \int_0^{2\pi} c(t) \sin n\omega t \, d\omega t$$

and $n = 1, 3, 5, \ldots$.

**Variable Speed Drive**

The force from the cylinder is exerted on the variable pitch sheave and this causes the sheave to change its pitch, thus accelerating or decelerating the machine. There is less force required on the sheave to decelerate the machine than to accelerate it.

![Diagram showing force-acceleration relationship and time response to a sinusoidal input.](image)
In this case, maximum acceleration equals maximum deceleration. This will only occur when \( \beta \) (deceleration slope) is equal to \( n \) (ratio of piston areas exposed) of the hydraulic cylinder.

For a sinusoidal input

\[
\begin{align*}
    r(t) &= A \sin \omega t \\
    c(t) &= \beta c \\
    &= A \sin \omega t \\
    &= \beta A \sin \omega t \\
    &= \beta c \\
    &= \beta A \sin \omega t
\end{align*}
\]

Describing function = \( \frac{b_n}{A} \)

where \( b_n = \frac{1}{\pi} \int_0^{2\pi} c(t) \sin n \omega t \, dt \)

\( n = 1, 3, 5, \ldots \)

**Windrow**

The speed of the machine has been adjusted by the control system according to the throughput. Under uniform windrow conditions the speed remains constant, however, if a change in the windrow occurs, a correction is made. Changes in the windrow do not follow any particular pattern and therefore this block of the control system must remain in doubt. It has been found that some types of variation will permit the system to operate satisfactorily, while others will not. In particular, any changes that occur gradually permit satisfactory operation
while large, abrupt and frequent changes do not. Fortunately by far the most variations in windrows occur gradually and thus the system generally controls the throughput satisfactorily. In the few cases where variations are large, abrupt and frequent, the system can be bypassed and thus no efficiency is forfeited. The reason for unsatisfactory automatic operation in these cases is the time delay from the time the crop enters the machine, until the system senses the load.
APPENDIX C

ALTERNATE ELECTRICAL SYSTEM

An electrical control system may be used as an alternative to the hydraulic system. The variable speed drive ratio is then adjusted by electro-mechanical means rather than by the hydraulic cylinder. The circuit diagram of an electrical control system is shown in Fig. C-1.

The power source for the system can be a D.C. battery, or more likely, a D.C. generator driven by the combine engine. A rheostat control is located between the power source and the first selector switch, and is used to adjust the voltage supply to the potentiometer. By adjusting the voltage, it is possible to regulate the response rate of the system. The hydraulic counterpart of the rheostat is the metering valve. A voltage regulator (not shown) at the power source corresponds to the pressure release valve in the hydraulic system.

The selector switches are used to select either one of the potentiometers for use in the circuit. The switches are operated simultaneously to prevent short circuiting. Their function is identical to that of the shut off valves and selector valve in the hydraulic system.

The manually operated potentiometer is used when conditions are not suitable for automatic control (i.e. road travel). The wiper of the manual potentiometer is returned to neutral by means of a spring, so that when the control handle is released, the speed correction is terminated (similar to a hydraulic valve with spring return).

The automatic potentiometer is mounted near the pivot arm (Fig. 6) and the position of the wiper is controlled by the pivot arm through the
FIG. C-1 ELECTRICAL CONTROL SYSTEM
valve control bracket. A dead zone is provided in the potentiometer to help prevent hunting. The connections to the armature circuit of the motor are made from the wiper and the dead zone of the potentiometer. This allows the voltage to be reversed, and thus causes the motor to change direction of rotation.

The reversible D.C. motor is connected to a gear reducer, which reduces the speed of rotation and increases the torque. The gear reducer drive is connected to a winch, which controls the position of the variable speed sheave by means of a cable.

Several safety devices are required to prevent circuit overload. A thermal overload can be used to limit the current flow to the electric motor. Manual reset from the operator's platform, or automatic reset can be used. A normally open relay can be used to signal to the operator when the end of the variable speed sheave stroke is reached. The relay is closed mechanically and a warning light or buzzer indicates to the operator that a different gear should be selected.

The D.C. generator must supply twice the voltage requirement of the motor due to the method of connecting the motor into the circuit. The maximum power output of the hydraulic cylinder in the original system is 0.09 hp. Therefore, an electric motor of 1/10 - 1/4 hp should provide sufficient power to move the variable speed sheave. The constant field voltage for the motor is supplied either from the generator or the combine battery.

Armature voltage control for motor speed regulation was chosen due to the torque and speed range characteristics of this type of control.
is assured of a uniform feed rate near the optimum level. A more
accurate load judgment is possible with automatic control since the
operator can make his decision on the basis of an extended operating
period, rather than on an instantaneous period, as with manual control.
With the losses being equalized over the entire field, it is more likely
that the operator will make a check of the amount of grain being lost,
since he will know that this represents the whole field rather than
just a particular small area. It is unlikely that the operator would
set the control system so the load is at such a high level that the grain
losses would be very high all over the field. If the same harvesting time
is used in automatic control, as in manual, the losses will be reduced due
to the maintenance of a more even rate of feeding with automatic control.
If equal grain losses are to be accepted in automatic and manual operation,
less time will be required to harvest the field when using automatic control.
(Davis 1964). The motor torque remains constant regardless of speed. The horsepower delivered therefore varies directly with speed. This is desirable for this application due to the type of load encountered. The power required is approximately proportional to the rate at which the load is to be moved (App. B). The range of speeds available with armature control is infinite. Smooth control of motor speed can be obtained down to zero, followed immediately by acceleration in the opposite direction. Using field circuit control, a speed range ratio of 4:1 is considered maximum and the torque is inversely proportional to the speed, so this type of speed control would not be suitable.

An estimate of the cost of components required follows:

<table>
<thead>
<tr>
<th>Component</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>$15</td>
</tr>
<tr>
<td>Idler</td>
<td>10</td>
</tr>
<tr>
<td>Damper</td>
<td>5</td>
</tr>
<tr>
<td>Spring</td>
<td>5</td>
</tr>
<tr>
<td>Gearmotor (1/4 hp)</td>
<td>35</td>
</tr>
<tr>
<td>Generator</td>
<td>25</td>
</tr>
<tr>
<td>Potentiometers</td>
<td>30</td>
</tr>
<tr>
<td>Miscellaneous</td>
<td>35</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>$160</td>
</tr>
</tbody>
</table>

Although the total cost is higher than for the hydraulic system, some of the hydraulic components presently on the combine would be eliminated, if the electrical system were used. These would include the pump, reservoir, valve and hydraulic cylinder. If the cost of these items is subtracted from the total cost of $160, the net cost would be very similar to that of the hydraulic system (i.e. $100).