ABSTRACT

A HIGH-VELOCITY, HIGH-MOMENTUM IMPACT TESTING DEVICE FOR AGRICULTURAL MATERIALS

By

Thomas Harold Burkhardt

Several methods have been used for impact testing of agricultural materials including impact by rotating objects, using the specimen as a projectile and falling weights. When using the first two methods, it is nearly impossible to predict the orientation of the product upon impact. Falling weights can have a large mass, but they have to fall long distances to achieve a high velocity.

A testing device which has a large momentum and a high impact velocity and permits a predictable orientation of the specimen prior to impact is needed. For example, to determine the forces and energy involved in the grain threshing process, the impact velocity and momentum have to be comparable to those in a combine cylinder.

A device with these characteristics has been developed at Michigan State University. A massive

flywheel is located on the primary shaft which is powered by an electric motor through a variable speed belt drive. An impacting arm is attached to a secondary shaft. The primary and secondary shafts are connected with an electrically operated clutch. The arm is rotated approximately 330 degrees before impact. After impact, the drive clutch is shut off and the brake turned on by means of a microswitch and some control relays.

The impact testing machine has been used in over 1000 tests with no serious problems. The impact velocity ranges from 1500 feet per minute to 7500 feet per minute. Instrumentation for measuring the impact velocity, impact force and acceleration of the impacting arm is included. Efforts were made throughout the study to minimize the operating hazard.

Approved B A Stout
Major Professor

Approved Carl W. Hall
Department Chairman

A HIGH-VELOCITY, HIGH-MOMENTUM IMPACT TESTING DEVICE FOR AGRICULTURAL MATERIALS

By

Thomas Harold Burkhardt

AN AE 811 TECHNICAL PROBLEM REPORT

Submitted to
Michigan State University
in partial fulfillment of the requirements
for the degree of

MASTER OF SCIENCE

Department of Agricultural Engineering

1969

ACKNOWLEDGMENTS

The author wishes to express a deep appreciation to Dr. B. A. Stout (Agricultural Engineering) for his enthusiastic support and interest during this study. His patience, guidance and encouragement were very important to the success of this venture.

Thanks are due to Mr. Zenon Karasek for his assistance in constructing the impact apparatus.

The author wishes to express his gratitude to the National Science Foundation for the financial assistance which it gave to help make this graduate program possible.

The contribution of the Warner Electric Brake and Clutch Company in supplying the electric clutch and brake was deeply appreciated.

Many thanks are due to Mr. Larry Rose for his assistance in conducting the experiments and tabulating the data.

The author greatly appreciated the assistance of Dr. G. E. Mase (Metallurgy, Mechanics and Materials Science) who served as the minor professor for this graduate program.

Dr. G. L. Cloud (Metallurgy, Mechanics and Materials Science) and Dr. W. N. Sharpe (Metallurgy, Mechanics and

Materials Science) very generously gave of their time and advice to help solve some of the instrumentation problems encountered during this project.

Thanks are due to those colleagues who freely gave advice and encouraged this study.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF APPENDICES

1. INTRODUCTION

When a crop is harvested with a grain combine, it is subjected to high velocity impact by the combine cylinder. The impact velocity may range from as low as 1000 feet per minute for dry beans to as high as 6000 feet per minute for wheat (Bainer _et al._, 1963). Impact loading is characterized by very large contact forces which are applied during a very short interval of time. Because of these loading conditions, mechanical damage in the form of cracking or breaking may occur. The damage resulting from the threshing process frequently lowers the market value of a crop.

In the state of Michigan alone, products valued at over $142 million were harvested in 1967 by this method (Michigan Department of Agriculture, 1968). During the same year the farmers of the United States harvested over five billion bushels of grain and beans with combines. Since these crops are an important portion of the economy of American agriculture, unnecessary loss of a few cents per bushel due to mechanical damage could amount to a large economic loss.

Several researchers have investigated the threshing process. They have attempted to explain some of the phenomena associated with threshing. However, the threshing process is still not well understood.

Thus, there are two major reasons for impact testing of grain:

1. To develop an explanation of the threshing process.
2. To study the effect of impact loading on grain quality.

2. A STUDY OF PAST INVESTIGATIONS

The phenomena associated with impact loading are not easily understood and the associated experimentation requires some sophisticated instrumentation. According to Mohsenin (1968) no general impact theory has been developed.

Charpy and Cornu-Thenard (1917) conducted a detailed investigation of the work required to rupture small metal bars. A vertical drop-weight machine, a pendulum drop-weight machine and a Guillery rotary-tup machine were used to obtain impact loading. They reported similar results from all three types of loading even though the testing machines had "different motions". They did not detect any influence of impact velocity in their results.

Mann (1936) used a variable speed tension machine to study the rupture of metal subjected to impact tensile loads. The specimen was attached to a pendulum on one end and a tup on the opposite end. A wheel consisting of two disks was driven by an electric motor. When the wheel reached the proper rotational velocity, an external tripping device released two horns which impacted the tup on the end of the specimen. During impact, energy was transmitted to the pendulum through the specimen. The movement of the pendulum was used as a measure of the energy required for rupture. The machine was capable of velocities up to 60,000 feet per minute. He reported that high-velocity tests are necessary to determine the true dynamic properties of materials.

Manjoine and Nadai (1940) studied the forces required to deform metals at high strain rates and elevated temperatures. A modified version of Mann's impact tester was used in this investigation. According to this report the impact testing machine used by Mann (1936) was very similar to the Guillery rotary-tup machine used by Cornu-Thenard (1917). However, the former was capable of higher impact velocities.

The study of impact loading has not been limited to metals. Shortly after the turn of the century, the Forest Service of the United States Department of Agriculture began using a vertical drop-weight machine to test the strength of wood. About twenty years later Wilson (1922) used both a vertical drop-weight machine and a pendulum drop-weight machine to measure the rupture energy of several species of wood.

Burns and Werring (1938) were interested in the ability of molded phenol plastic telephones to withstand accidental dropping. They used a low energy pendulum drop-weight machine to load molded specimens of plastic materials. They reported that temperature and moisture content of the plastic materials should be carefully controlled in precision testing.

The investigations discussed above are representative examples of early attempts of engineers and scientists to understand some of the factors associated with impact. In the previous chapter the importance of impact loading in certain areas of agriculture was established. It is thus understandable that in recent years some similar techniques have been used for impact testing of agricultural materials.

Kolganov (1956) studied grain threshing by using conventional combine cylinders. A major portion of his tests used two-stage threshing with the wheat passing a low-velocity cylinder and the unthreshed portion passing a high-velocity cylinder. He reported that increasing cylinder velocity or decreasing concave clearance caused increased mechanical damage with the maximum damage resulting from maximum velocity and minimum clearance. Threshing with a single cylinder resulted in levels of unthreshed grain and mechanical damage higher than those achieved with two-stage threshing.

King and Riddolls (1960) used a single cylinder threshing rig to test wheat seed and pea seed. The cylinder velocities and concave clearances were varied. They reported that the percentage of damage increased as concave clearance was reduced. The percentage of germination was reduced as the cylinder velocity was increased.

Perry and Hall (1960) dropped pea beans from a height of 22.5 feet through a vertical tube onto other beans. They found that the amount of visible damage was appreciably affected by the moisture content, but not the temperature of the dropped beans. Both temperature and moisture content affected the percentage of germination and the quality of the seedlings.

Narayan (1969) impacted pea beans with a vertical weight-drop machine to investigate the machanical checking of seed coats. He measured the energy required to check the seed coats of fifty percent of the beans in a given sample as a function of moisture content. An optimum moisture level for withstanding impact loading without mechanical checking was found.

Two methods of impact loading were used by Bilanski (1965) to study the breaking strength of seed grains such as soybeans, corn, winter wheat, barley and oats. He used a pendulum weight-drop machine for medium-velocity loading. The high-velocity loading was achieved by dropping the seeds into the path of a paddle rotating in the horizontal plane. He reported that the breaking strength of the grain was influenced by its size, moisture content and orientation before impact.

A spring-loaded arm was used by Perry and Hall (1965) as a method for impacting pea beans. High-speed photography was employed extensively in this investigation. The high-speed motion pictures were used to measure the velocity of the striking bar and the velocities of the bean and the rebound restriction blocks after impact. They were also used to measure the maximum deformation of the bean and the total time of impact.

Kirk and McLeod (1967) used a blowpipe to impact cottonseeds against a flat steel plate. They reported that damage increased as velocity increased, but that it was independent of moisture content.

Leonhardt et al. (1961) made use of a spring-loaded gun to fire sorghum seeds against a cantilever beam. They measured the amount of energy absorbed by a seed during impact. It was reported that damage increased with an increase in impact velocity and with a decrease in moisture content.

Mitchell and Rounthwaite (1964) tested two varieties of wheat with a machine similar to the one used by Mann (1936). A circular plate rotating in the horizontal plane moved the seed into the path of the impacting horn. They indicated that at low levels of moisture the visible damage increased with an increase in velocity, while at high levels of moisture the percentage of germination was decreased with an increase in velocity. They found one variety to be more prone to shatter than the other.

A rotating arm with a flat metal plate on the end was used by Turner et al. (1967) for impact loading of peanuts. A horizontal conveyor carried the peanuts into the path of the arm. They found mechanical damage to be influenced by impact velocity, moisture content and orientation of the specimen.

For studying the phenomena associated with threshing, it is desirable to use a testing method which closely simulates the impact action of the combine cylinder. It has been shown that mechanical damage depends on impact velocity. Since impact forces are related to the changes in momentum during contact, the inertia of the impacter is also important. Thus if the action of the combine cylinder is to be simulated, the testing machine should at least be able to match the velocity and inertia of the cylinder.

The vertical weight-drop tester and pendulum weight-drop tester can be made very massive. However, neglecting air resistance, a drop of over 150 feet would be required to reach a velocity of 6000 feet per minute, which is a typical combine cylinder peripheral velocity. Thus it seems that these machines are not a practical way of matching the cylinder.

The blow-pipe and the spring-loaded gun which fire the specimen against a massive object can meet the velocity and inertia requirements. However, Bilanski (1965) reported that orientation of the specimen is an important variable to be considered. Orientation cannot be controlled when using these devices. A machine similar to these could be used to launch a projectile which would impact an oriented specimen, but a very large projectile would be needed to match the inertia of the combine cylinder. For example, a projectile with 100 times the kinetic energy of a corn ear would have to weigh at least 50 pounds. Certainly a projectile of this size would be very difficult to control.

Moving a small oriented specimen into the path of a continuously rotating arm has been used as a means for impact testing. There would be less than two-tenths of a second available to move a specimen into the path of an arm two feet long with an impact velocity of 4000 feet per minute. It would be difficult to move a specimen as massive as a wheat head or corn ear during this short time interval without losing orientation.

Thus, several methods of impact loading are presently available, but it is apparent that an improved method is necessary in order to more closely simulate the action of the combine cylinder.

3. OBJECTIVE

The objective of this research was to develop a general purpose impact testing machine with the following characteristics:

1. A wide range of impact velocities.
2. Suitable instrumentation for measuring impact velocity and force.
3. Sufficient inertia to prevent significant loss of velocity during impact.
4. Controlled orientation of the specimen.
5. Suitability for a wide range of products.
6. Minimum operating hazard.

4. MECHANICAL COMPONENTS

The impact testing machine which was developed utilizes a rotary motion similar to that of a pendulum. However, mechanical energy was used in lieu of gravity for accelerating the impacter. This rotary impact testing machine consists of two major units. The primary unit has high rotational inertia and the secondary unit contains an impacting arm with low rotational inertia. When the primary unit has achieved the selected rotational velocity, the two units are coupled and the impacting arm is forced to rotate. After approximately 330 degrees of rotation, the arm strikes a specimen. The arm is then decelerated and stopped approximately one revolution after impact.

The primary and secondary shafts were coupled by a Warner Electric clutch, Model SF-1000, (Figures 1 and 2). As a means of stopping the rotating arm after impact, a Warner Electric clutch, Model SF-825, was mounted on the secondary shaft and used as a brake. Electric clutches were chosen so the system could be controlled electronically.

The impact testing device is mounted on a reinforced concrete base (Figure 1). The source of mechanical power

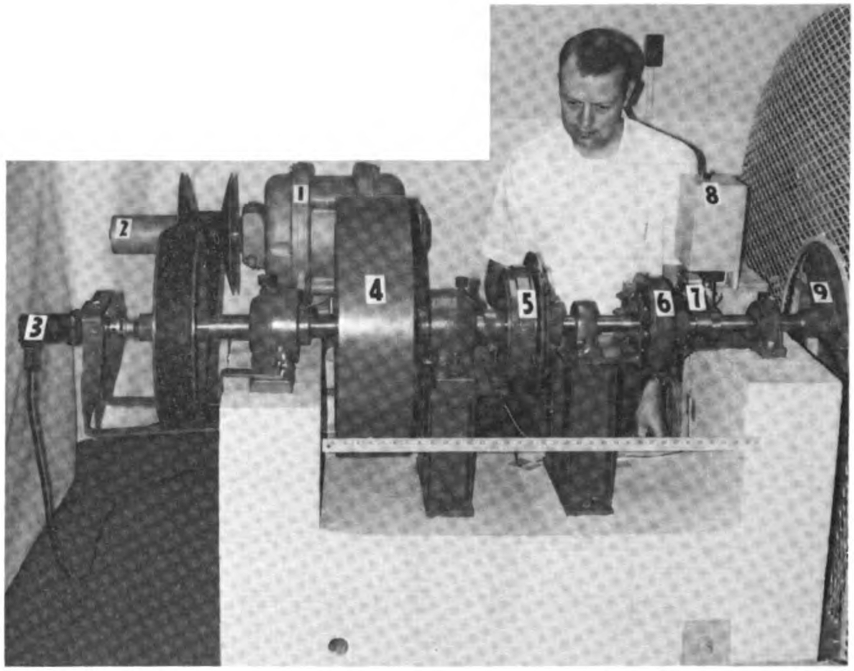

FIGURE 1. IMPACT TESTING MACHINE SHOWING THE FOLLOWING COMPONENTS: (PHOTO NO. 68-66.)

1. ELECTRIC MOTOR
2. VARIABLE SPEED PULLEY
3. TACHOMETER GENERATOR
4. FLYWHEEL
5. ELECTRIC CLUTCH
6. ELECTRIC BRAKE
7. CAM AND MICROSWITCH
8. BRAKE CONTROL SWITCH
9. IMPACT ARM

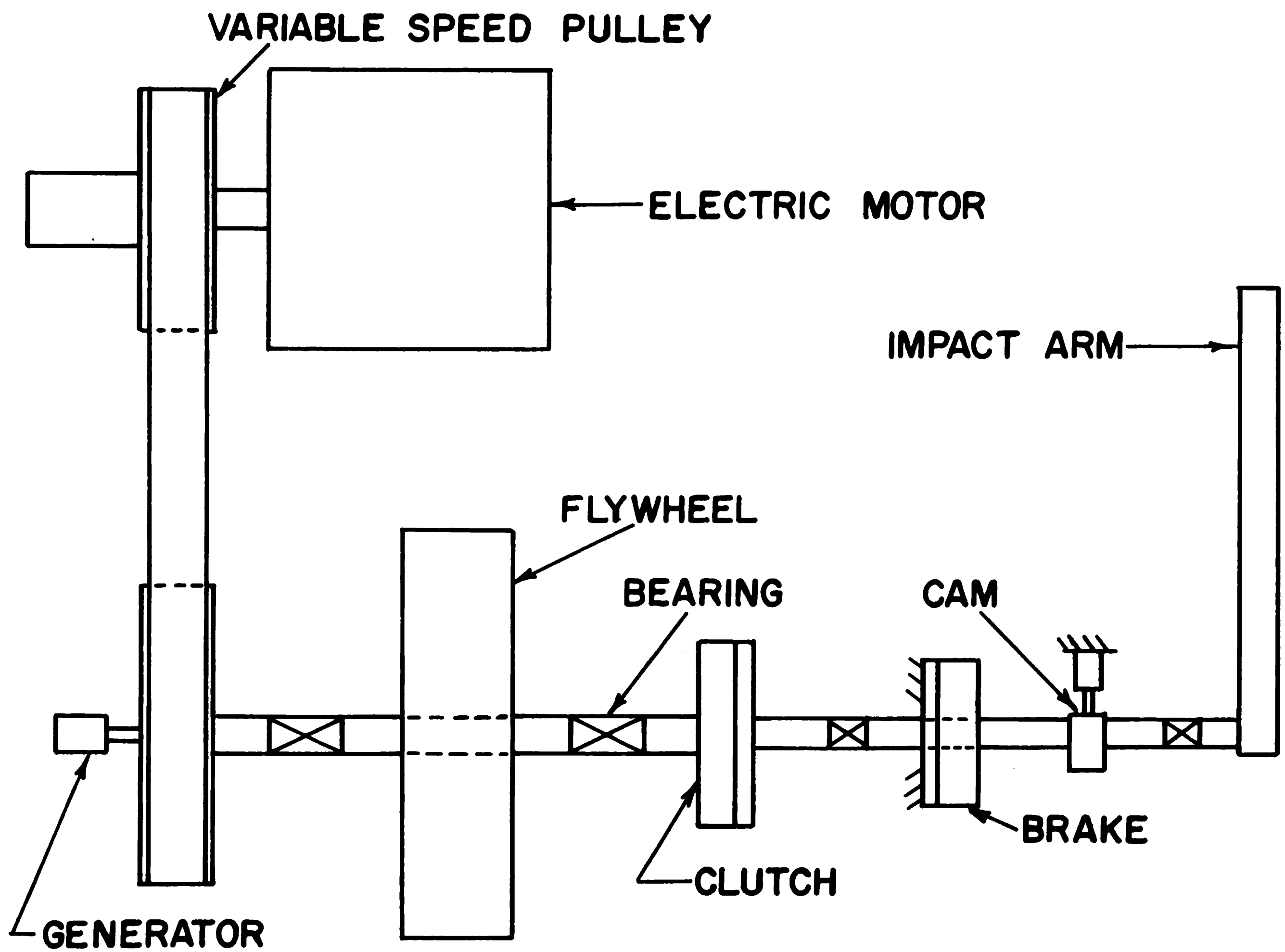

FIGURE 2. BLOCK DIAGRAM OF IMPACT TESTING MACHINE.

for the primary unit is a five horsepower, 860 rpm induction motor. The variable speed pulley and the movable base on which the motor is mounted provide a means for changing the impact velocity. To assist the motor with the acceleration of the impact arm, rotational inertia was added to the system by mounting a massive flywheel on the primary shaft. Using standard machine design techniques, it was determined that a two-inch shaft would be sufficient to bear the weight of the flywheel and withstand the repeated loading from the tension in the belt.

An analysis was made to determine the rotational inertia requirements for the flywheel. It was assumed that the torque of the motor and the rotational inertia of the drive pulley and driven sheave would be sufficient to replace the heat energy lost from the clutch. The rotational inertia of the clutch and brake was small compared to that of the impacting arm, and therefore was neglected. Assuming that only the kinetic energy of the flywheel was used to accelerate the impacting arm, the following relationship can be written:

$$I_F \omega_1^2 = (I_F + I_A)\omega_2^2 \qquad [1]$$

where:

I_F = rotational inertia of the flywheel

I_A = rotational inertia of the arm

ω_1 = rotational velocity of the flywheel before acceleration of the arm

ω_2 = rotational velocity of the flywheel after acceleration of the arm

For design purposes it was decided that the velocity of the flywheel should not be reduced more than 2 percent, i.e.:

$$\omega_2 = 0.98\omega_1 \qquad [2]$$

Combining equations [1] and [2] shows that the rotational inertia required for the flywheel is given by:

$$I_F = 24(I_A) \qquad [3]$$

The impacting arm was constructed with rectangular aluminum tubing and has a rotational inertia of 0.15 slug-ft^2 compared with 4.80 slug-ft^2 for the flywheel. This exceeds the requirements of equation [3] so it is expected the velocity of the flywheel would be lowered less than 2 percent. During actual use of the machine, there is no measurable slowing of the flywheel while accelerating the impacting arm.

It was necessary to obtain an estimate of the size of the member to be used for the impacting arm. For this design procedure the test specimen was considered to be an ear of corn weighing one-half pound. It was assumed that during impact, the surface of the specimen on the side opposite the contact area remained at rest while all the deformation occurred and was then instantaneously accelerated to the velocity of the impact arm. Under this assumption the deformation of the specimen is equal to the distance traveled by the impacter while the specimen is being accelerated.

After careful study of some corn ears, it was hypothesized that six-tenths of an inch would be a good estimate of the specimen's deformation during impact. Calculations showed that a force of 1086 pounds would be required to accelerate a one-half pound corn ear from zero to 5000 feet per minute in a distance of six-tenths of an inch. Using this value of force and design techniques, it was found that a member with a cross-sectional moment of inertia of 1.24 in^4 would be required for the arm. Rectangular aluminum tubing with a cross-sectional moment of inertia of 1.47 in^4 was chosen.

A cam (Figures 1 and 2) attached to the secondary shaft operates a micro-switch that is one of the components of the circuit for controlling the action of the clutch and brake. This cam was designed to rotate 280 degrees while the cam follower travels from zero displacement to the point where the electrical contacts are closed. The cam goes through 35 degrees of rotation while the microswitch is in the closed position leaving 45 degrees of rotation for the follower to return to zero displacement.

The safety of the operator was a prime consideration. Consequently, all of the moving parts of the impact tester were covered by shields made of expanded metal (Figures 3 and 4). The shield surrounding the impact arm was made much stronger than those used to cover the other components so that in case of failure of the arm, any broken parts would remain within the shield.

A collection box was added to gather the specimen after impact (Figure 4). The interior of the box was lined with foam rubber to prevent damage to particles hitting the box. Each specimen was suspended from the top of the collection box with small pieces of masking tape (See Figure 7, p. 24).

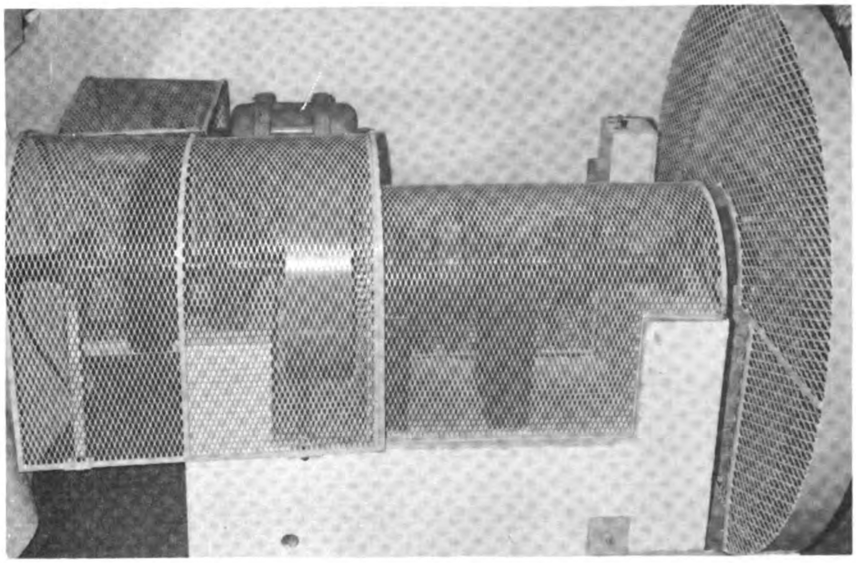

FIGURE 3. IMPACT TESTING MACHINE WITH SAFETY SHIELDS IN POSITION. (PHOTO NO. 68-85)

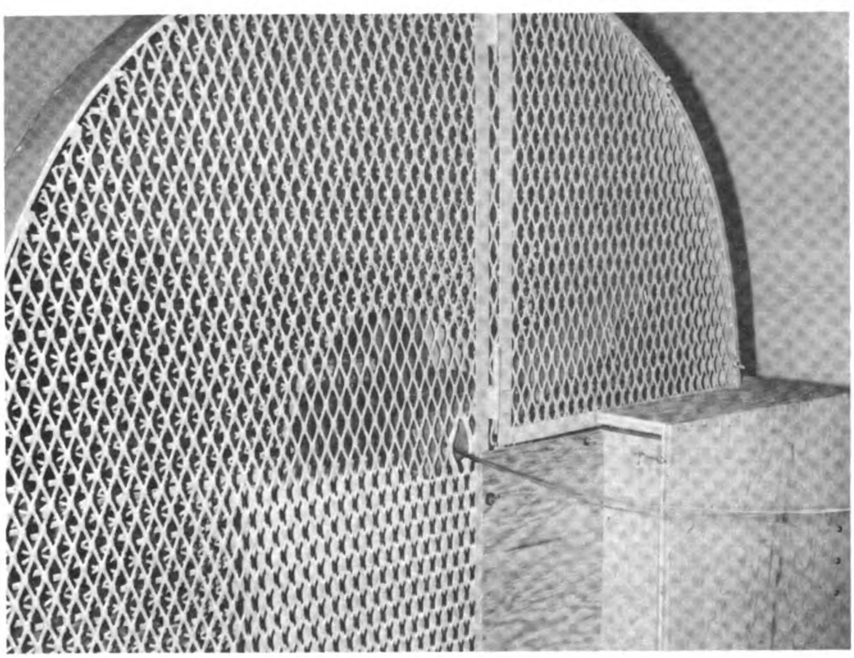

FIGURE 4. COLLECTION BOX AND SAFETY SHIELD FOR IMPACTING ARM. (PHOTO NO. 68-70.)

5. INSTRUMENTATION

The circuit shown in Figure 5 controls the operation of the clutch and brake, and hence controls the starting and stopping of the impact arm. The designations used to label the components of the circuit are those recommended by the National Association of Relay Manufacturers (1966). The electrical potential comes from an Electro Model EC-1 direct current power supply with a maximum output of five amperes at twelve volts.

Each contact is labeled by one of the numbers on the left of the positive line. The contacts controlled by a coil are identified by a sequence of numbers on the right of that particular coil. A normally closed contact is indicated by underlining its identification number. For example, the designation of the right of 1CRL indicates that contact number six is normally open, contact number ten is normally closed, and both contacts are controlled by the 1CRL coil.

The contacts numbered one, three, four and five are normally open, momentary contact switches. The contacts numbered two and nine are microswitches. Latching relays are designated by CRL, unlatching relays by CRU and control relays by CR.

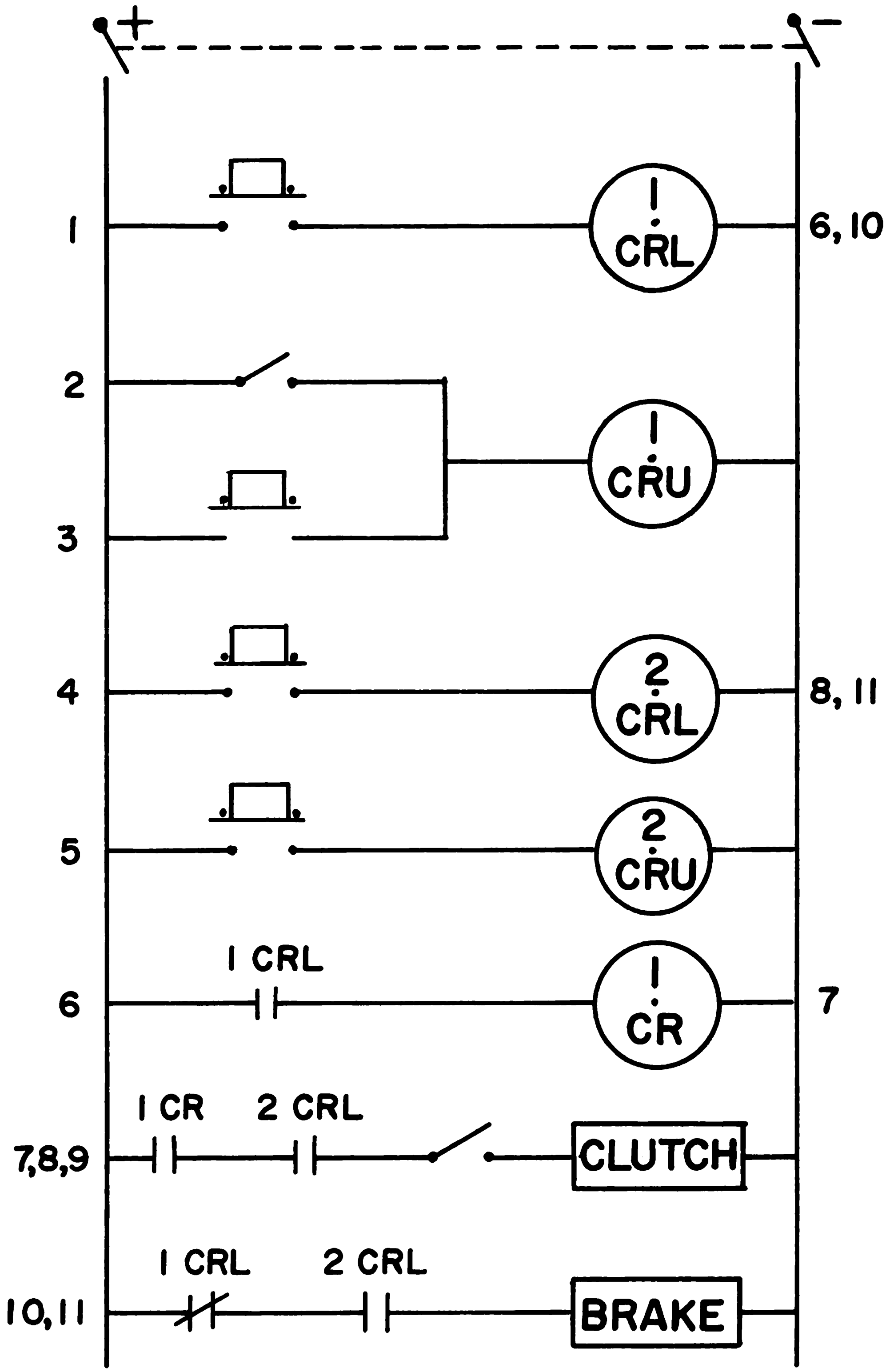

FIGURE 5. BRAKE AND CLUTCH CONTROL CIRCUIT.

Before each test the arm was placed in approximately the impact position so the specimen could be properly aligned with the impacting face. In order to manually rotate the arm to this position, the brake had to be turned off. To accomplish this a brake control switch (contact five) was closed. This procedure opened the contacts numbered eight and eleven to turn off the brake and cut the clutch out of the circuit for safety. To hold the arm in the desired position, the other brake control switch (contact four) was closed to turn on the brake and reconnect the clutch to the circuit. After the specimen was mounted in the collection box (Figure 7), the arm was rotated backward approximately 330 degrees. Further backward rotation would have caused the cam to activate the microswitch (number eight in Figure 1). Of course, the brake control switches were also used during this backward rotation.

Each test was initiated by depressing the clutch start switch (contact one) to power a latching relay (1CRL). This event opened contact number ten to turn off the brake and closed contact number six to power the control relay (1CR). As a result, contact number seven was closed to turn on the clutch. The arm then rotated through approximately 330 degrees before it impacted the specimen.

A cam-activated microswitch (contact two) was closed a few degrees of rotation after impact to power an unlatching relay (1CRU). This operation closed contact number ten to turn on the brake and opened contact number six to cut off the current to the control relay (1CR). This opened contact number seven to turn off the clutch. If there would have been a mechanical failure of the cam or microswitch, the braking procedure could have been initiated with the emergency switch (contact three).

Several safety features were added to this circuit. When the door of the specimen collection box (Figure 4) is open, the microswitch (contact nine) is open. In case the clutch start switch should accidently be closed while the door is open, the clutch will not operate.

When the brake is turned off with the brake control switch, the line to the clutch is also opened. This safety precaution prevents the operation of the clutch while the brake is unusable. If the clutch could be activated while the brake is cut out of the circuit, there would be no way to stop the rotation of the arm. During regular operation the cam-operated microswitch may fail to function properly. In this situation the arm can be stopped by closing the emergency off switch.

For design purposes it was assumed that the impact arm could be accelerated to a constant angular velocity in less than one revolution. As a means of checking the validity of this assumption, two systems for measuring impact velocity were developed and tested. The results of these tests will be reported in a later chapter. Under the previous assumption, if the rotational velocity of the flywheel and the effective length of the arm are known, the impact velocity can be calculated. The rotational velocity is measured by a Standard Electric Time Co., Model SG-6 chrono-tachometer powered by a Standard Type CM-9 generator.

The other velocity measurement system consists of two Electro Model 3030-AN magnetic pickups and a Beckman Model 6040A preset EPUT and timer. As the impact arm passes a magnetic pickup, the timer is triggered and starts counting micro-seconds. When the arm passes a second pickup, the timer stops counting. If the distance between the magnetic pickup and the time required to travel that distance are known, the velocity can be calculated. Figure 8 shows impact velocity vs. time as read directly from the electronic timer.

A Kistler Model 900 A Series quartz load washer was placed between the impacting face and the arm (Figure 7). The signal from the load washer is conditioned by a Kistler Model 503M15 charge amplifier and recorded on a Tektronix Type 549 storage oscilloscope equipped with a Type 1A4 four-channel amplifier (Figure 6). This system was utilized to measure the impact force.

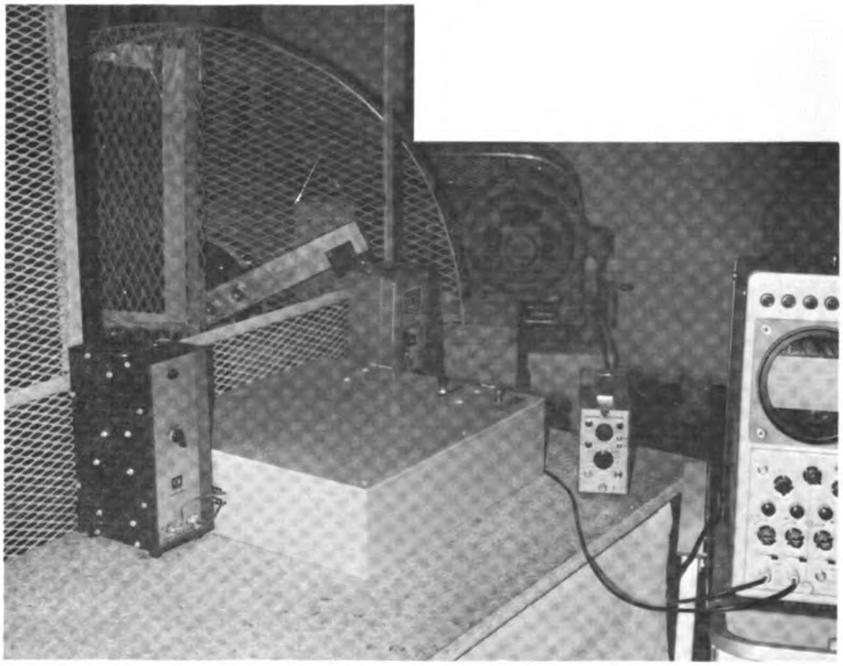

FIGURE 6. INSTRUMENTATION AS SEEN FROM THE OPERATOR'S POSITION. (PHOTO NO. 68-87.)

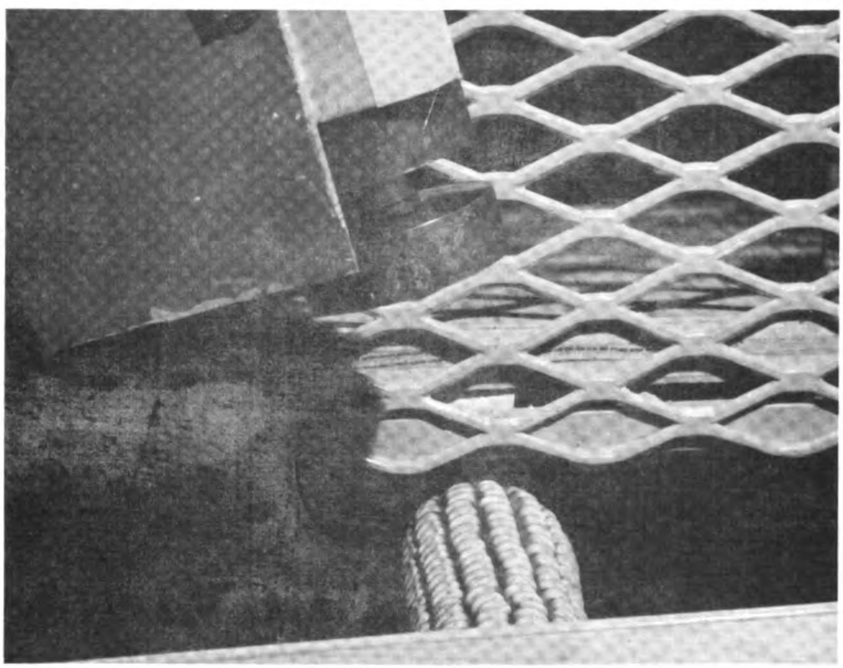

FIGURE 7. IMPACTING ARM WITH A SPECIMEN IN POSITION. (PHOTO NO. 68-94.)

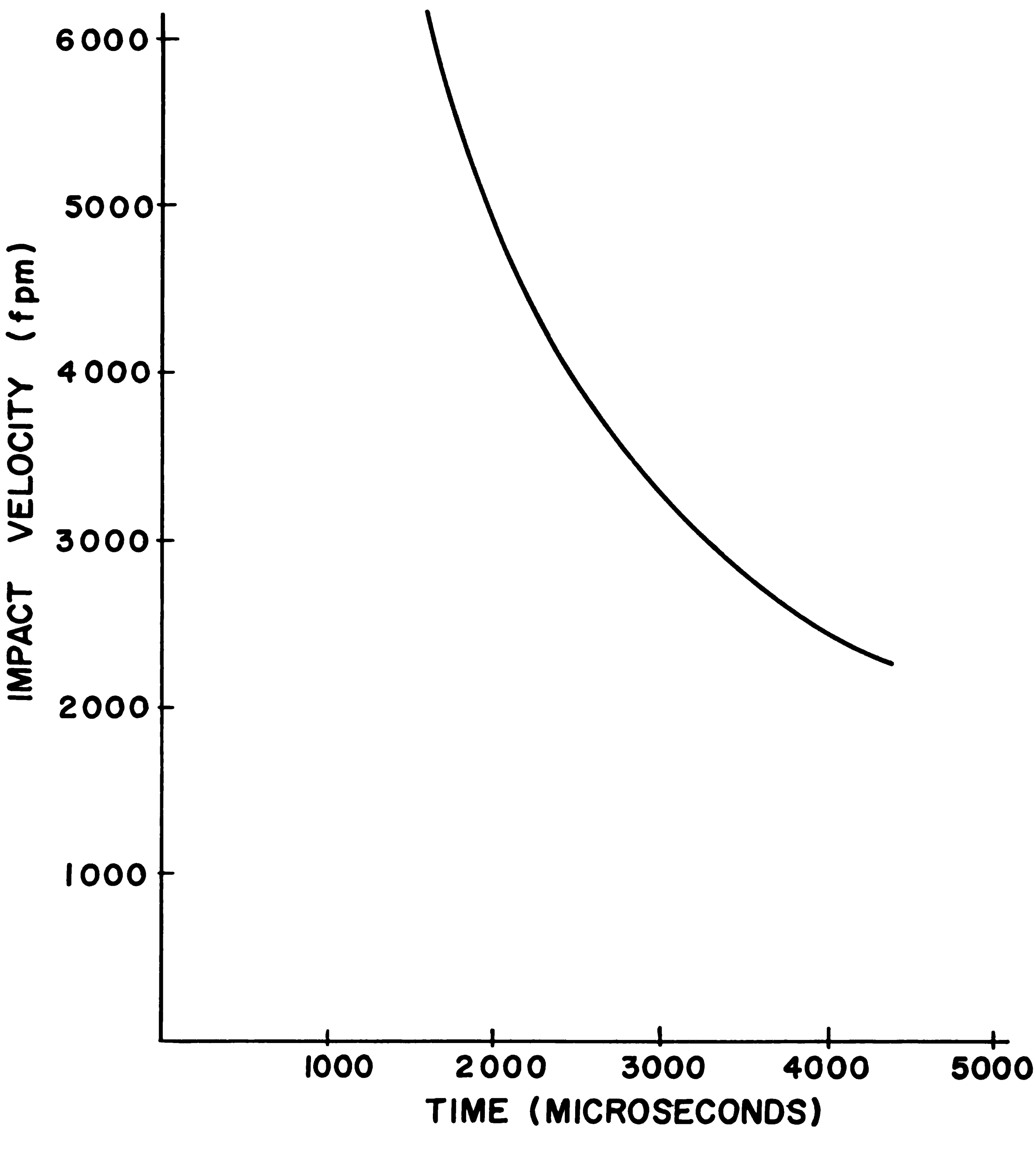

FIGURE 8. IMPACT VELOCITY VS. TIME. THE TIME IS READ DIRECTLY FROM THE ELECTRONIC TIMER. THE CURVE IS VALID FOR AN 18-INCH IMPACTING ARM.

A Kistler Piezotron Model 818 accelerometer was mounted inside the arm directly above the load washer. It was oriented so that only the tangential component of acceleration is measured. The accelerometer signal is conditioned by a Kistler Piezotron Model 548B coupler and recorded on the storage oscilloscope. The oscilloscope is triggered by the accelerometer signal.

Under stationary conditions, the conduction of a signal from a transducer to the recording equipment can be readily accomplished by using shielded cable. However, the load washer and the accelerometer which are mounted on the impacting arm rotate with respect to the oscilloscope. This relative motion complicates the signal transfer.

The rotary impact tester developed by Clark _et al._ (1967) utilized slip rings to conduct the signal from a strain gage bridge circuit. However, the signal from a piezoelectric transducer is so small that it is doubtful the signal could be successfully conducted through slip rings. In the case of continuous rotation, the noise component of the signal originating at the slip ring contacts might be negligible in comparison to the signal component originating at the piezoelectric transducer. However, the impact tester developed during this study does not have a continuous rotary motion. There is a rapid acceleration with the impacting arm reaching constant velocity only a few microseconds before conduction of the signal. It is likely that slip rings used under this condition would add a larger noise component than if they were under continuous rotation.

According to Beckwith and Buck (1961) direct connection can be used for signal conduction when the duration of the rotary motion is short enough. They suggest providing sufficient length of shielded cable and allowing the leads to wrap around the rotating shaft. A modified version of their method was utilized in this study. The leads were aligned with the center of the secondary shaft (Figure 4) so that during the rotation of the arm, the cables were subjected to a torsional deformation in lieu of bending deformation. This twist which lasts for approximately two revolutions is distributed over several feet of cable. By twisting the cables backward one revolution when getting ready for each test, the leads are unwound at the time of signal conduction.

In order to prevent damage to the leads, it is necessary to stop the arm as soon as possible after impact. A circuit (Figure 5) was developed to automatically turn on the brake shortly after impact. If it were not for the protection of the cables, the operation of the clutch and brake could have been controlled manually.

6. OPERATING PROCEDURE

Proper maintenance and caution are normally necessary for the successful use of any machine. The impact testing device certainly is no exception to this rule. This equipment should be carefully inspected before a series of tests are run. Failure of the tester during operation could injure the operator or damage expensive electronic equipment.

The concrete base, the frame joints and the impact arm should be checked for cracks or any other signs of failure. All of the nuts should be examined for proper tightness. The operation of the cam-activated micro-switch should be observed while the shaft is manually rotated. If there is evidence of faulty operation of the cam and microswitch, they should be adjusted to function properly. Failure of the cam and microswitch to activate the brake during a test will cause unnecessary deformation and possible damage of the shielded cable. All of the safety shields should be returned to their proper positions after this inspection.

To obtain the desired impact velocity the motor is simply moved relative to the primary shaft by use of the crank on the sliding motor base. The velocity adjustment should be carried out with the motor running to prevent damage to the drive belt and to use the tachometer as an aid for velocity selection. After the desired rotational velocity is reached, the motor should be stopped and started a few times to find out if the velocity setting is correct. Frequently the initial adjustment is not quite right, and minor corrections are needed to obtain the desired velocity. Apparently a few cycles are required before the drive belt reaches an equilibrium position.

The proper manual should be consulted for the operating procedure of any instrument that is not thoroughly understood. A trial and error method is frequently needed to adjust the calibration and trigger sensitivity of the oscilloscope. The same method may be needed to adjust the trigger sensitivity of the electronic timer. Consequently, several trial runs should be attempted before the actual testing is begun.

The magnitude of the signal from the magnetic pickups depends upon the velocity of the arm and upon the clearance between the arm and the pickups. When the impact velocity is adjusted to a lower value, the peak voltage of the signal from the magnetic pickups will be lowered. The trigger sensitivity of the electronic timer may need adjusting to compensate for the smaller peak voltage. If consistant triggering cannot be achieved at low impact velocities, the clearance between the pickups and the arm may be reduced. The operator's manual for the magnetic pickups gives the relationship between clearance and peak voltage.

When the power supply is turned on, the brake is activated and the impacting arm is held firmly in place. Before each test it is desirable to have the arm in the impact position to ensure proper alignment of the specimen with the impacting face. This can be accomplished by turning off the brake control switch to release the arm so that it can be rotated to the impact position. It is advisable to turn the arm backward during this positioning process in order to prevent excessive twisting of the shielded cable. The brake control switch can be reactivated to hold the arm in position while the specimen is being secured in place. The arm should then be turned backward approximately 330 degrees to the ready position.

At this point the door of the collection box should be closed. For safety purposes, all personnel in the area should be moved well away from the plane in which the arm rotates. The oscilloscope and the electronic timer should be reset so they are ready to record the data.

About 15 seconds after the electric motor has been started, the clutch start switch should be depressed and immediately released. If a delay different from 15 seconds before initiating the tests is found convenient, the test results will not be affected. However, it was found that the impact velocity is dependent on the length of the delay. Thus, it is important to keep the wait as constant as possible to reduce the variability of the impact velocity. The clutch could fail to operate for any of the following reasons:

1. The arm was turned past the ready position so that the cam activated the microswitch in the reverse direction.
2. The door of the collection box was not properly closed and the clutch was cut out of the circuit by the safety switch.
3. The brake control switch was left in the off position so that the clutch was cut out of the circuit.
4. The power supply was not turned on.
5. The power supply was not plugged in.

If the clutch fails to operate and none of the above factors is at fault, the circuit should be checked for faulty relays or loose connections. To date, all of the clutch failures have been due to an oversight by the operator.

The microswitch is moved very rapidly by the rotating cam, and this action causes small vibrations in the cam and the microswitch. As a result of this motion, either component could occasionally move out of adjustment causing the automatic braking system to fail. Normally the preliminary inspection will eliminate this problem. However, the operator should always be prepared to depress the emergency off switch to manually activate the brake.

After the specimen has been impacted, the motor should be stopped. The rotation of the flywheel should be completely ceased before the specimen is removed from the collection box. If this precaution were not taken, accidental rotation of the arm could cause serious injury. While waiting for the flywheel to become motionless, it is convenient to record the data from the electronic timer and the oscilloscope.

Summary of Operating Procedure

1. Inspect mechanical components
2. Select impact velocity
3. Connect and calibrate instruments
4. Rotate arm to impact position
5. Mount specimen
6. Rotate arm to ready position
7. Close door of collection box
8. Start electric motor
9. Activate clutch
10. Stop electric motor
11. Retrieve specimen

7. EVALUATION

A high-velocity, high-momentum impact testing device for agricultural materials has been developed during this study. The testing device has been used for more than 1000 impacting tests without any serious difficulties.

When using an 18-inch arm, the impact velocities can range from 2300 to 5600 feet per minute. Velocities as high as 7500 feet per minute can be achieved by changing to a 24-inch impacting arm. If velocities higher than these are desired, a smaller driven sheave could be attached to the primary shaft. In case this modification would not allow sufficient time to accelerate the arm to a constant angular velocity, it would be necessary to use a larger clutch and a longer arm to achieve higher velocities. By changing to a 12-inch impacting arm, the range of velocities can be lowered to 1500 feet per minute. This machine could be modified to obtain lower velocities by adding a larger driven sheave.

The use of the tachometer for velocity measurement depends on the assumption that the arm reached a constant angular velocity before impact. The other measurement system provides the velocity of the arm immediately prior to impact and is not based on the constant velocity assumption.

If the arm did not reach a constant velocity, the electronic timer would indicate an average velocity over the distance between magnetic pickups. The velocity as measured by the tachometer would be considerably larger than the value obtained from the electronic timer. However, if the arm did reach a constant velocity, the two systems should indicate the same velocity. Also, the electronic timer would indicate the actual velocity and not an average velocity.

Several tests were run using the two systems simultaneously. The results were nearly the same and supported the previous assumption. For example, on a trial of 20 tests at the same velocity setting the 95 percent confidence interval for the tachometer was 3447 ± 3 feet per minute. On the same trial the 95 percent confidence interval for the electronic timer was 3507 ± 6 feet per minute and the 95 percent confidence interval for the difference of the means was 60 ± 6 feet per minute. The confidence interval for the timer was wider than that for the tachometer because it included the variability of the clutch.

Other trials, each at a different velocity setting and each consisting of 20 tests, were run. The width of the confidence intervals for the tachometer and the electronic timer were all about the same as those listed above. However, in all cases the confidence interval for the differences of the means was smaller indicating a closer agreement between the two systems.

The accelerometer is mounted in the arm in such a manner that only the tangential component of acceleration is measured. The tangential component of acceleration is zero when there is a constant angular velocity. The signal from the accelerometer returns to zero before impact indicating a constant velocity. This evidence lends further support to the constant velocity assumption.

Movies of the impact process were taken at 6100 frames per second. No disturbance of the specimen by air currents prior to impact was detected from careful study of these movies. No significant reduction of arm velocity was observed during impact.

Some of these movies were taken while a ball bearing was being impacted. It was hoped that a frame-by-frame analysis would yield an estimate of the acceleration of the ball bearing to check the calibration of the load washer. However, by close examination of the film it was found that the impacting face and the ball bearing were in contact for only one frame. Thus, no estimate of acceleration could be obtained by this method.

The calibration of the load washer than had to be accomplished using the impulse-momentum law:

$$m(v_2 - v_1) = \int_{t_1}^{t_2} F\,dt \qquad [4]$$

where:

m = the mass of the bearing

t_1 = the time of contact

t_2 = the time of separation

v_1 = the velocity of the bearing at time t_1

v_2 = the velocity of the bearing at time t_2

Since the bearing was at rest at the time of contact, $v_1 = 0$. The value of the right side of [4] was estimated by measuring the area under the force-time curve. Then v_2 was calculated to be 3200 feet per minute. From a frame-by-frame analysis, v_2 was determined to be 3035 feet per minute. Mohsenin (1968) reported that a discrepancy such as this can be expected when the collision is not perfectly elastic.

According to Halliday and Resnick (1963) if this had been a perfectly elastic collision, the impact velocity of 2500 feet per minute would have given v_2 = 5000 feet per minute. Thus, the impact of the bearing was not an elastic collision. Since much or all of the difference in the two values of v_2 has been explained, the calibration of the load washer is apparently quite correct.

The impact testing machine which was developed during this study has a wide range of impact velocities. It has suitable instrumentation for measuring impact velocity and force and sufficient inertia to prevent significant loss of velocity during impact. The orientation of the specimen prior to impact can be controlled and a wide range of products can be tested. Throughout the development and use of the impact testing machine efforts were made to minimize possible hazards to both the operator and the machine.

REFERENCES

Arnold, R. E.
1959 The effect of harvest damage on the germination of barley. Journal of Agricultural Engineering Research 4(1):24-29.

Bainer, R., R. A. Kepner and E. L. Barger
1963 Principles of Farm Machinery. John Wiley and Sons, Inc., New York. 571 pp.

Beckwith, T. G. and N. L. Buck
1961 Mechanical Measurements. Addison-Wesley Publishing Company, Inc., Reading. 559 pp.

Bilanski, W. K.
1966 Damage resistance of seed grains. Transactions of the ASAE 9(3):360-363.

Burns, R. and W. W. Werring
1938 The impact testing of plastics. Proceedings of the Forty-First Annual Meeting of the American Society for Testing Materials 38(2): 39-52.

Charpy, G. and A. Cornu-Thenard
1917 New experiments on shock tests and on the determination of resilience. Journal, Iron and Steel Institute 96(2):61-104.

Clark, R. L., G. B. Welch and J. H. Anderson
1967 The effect of high velocity impact on the germination and damage of cottonseed. Paper No. 67-822 presented at the 1967 Winter Meeting, American Society of Agricultural Engineers, Detroit.

Hall, G. E. and W. H. Johnson
1968 Corn kernal crackage induced by mechanical shelling. Paper No. 68-112 presented at the 1968 Summer Meeting, American Society of Agricultural Engineers, Logan.

Halliday, D. and R. Resnick
1963 Physics for Students of Science and Engineering. John Wiley and Sons, Inc., New York. 1177 pp.

Halyk, R. M.
1968 A theory of corn shelling based on a quasi-static analysis. Thesis for the degree of Ph. D., Michigan State University, East Lansing. (Unpublished)

King, D. L. and A. W. Riddolls
1960 Damage to wheat seed and pea seed in threshing. Journal of Agricultural Engineering Research 5(4):387-398.

Kirk, I. W. and H. E. McLeod
1967 Cottonseed rupture from static energy and impact velocity. Transactions of the ASAE 10(2): 217-219.

Kolganov, K. G.
1956 Mechanical damage to grain during threshing. Sbornik Trudov Po Zemledyelskoi 3:321. (Translation by E. Harris in Journal of Agricultural Engineering Research, 1958, 3(1):179-184.)

Kunze, O. R., F. W. Snyder and C. W. Hall
1965 Impact effects on germination and seedling vigor of sugar beets. Journal of the American Society of Sugar Beet Technologists 13(4): 341-353.

Leonhardt, J. L., G. C. Zoerb, and D. D. Hamann
1961 Investigation of factors affecting impact damage to sorghum seeds. Paper No. 61-125 presented at the 1961 Summer Meeting, American Society of Agricultural Engineers, Ames.

Manjoine, M. and A. Nadai
1940 High-speed tension tests at elevated temperatures. Proceedings of the Forty-Third Annual Meeting of the American Society for Testing Materials 40(1):822-837.

Mann, H. C.
1936 High-velocity tension-impact tests. Proceedings of the Thirty-Ninth Annual Meeting of the American Society for Testing Materials 36(2):85-97.

Michigan Department of Agriculture
1968 Michigan Agricultural Statistics. Michigan Crop Reporting Service, Marketing Division, Lansing, Michigan. 38 pp.

Mitchell, F. S. and T. E. Rounthwaite
1964 Resistance of two varieties of wheat to mechanical damage by impact. Journal of Agricultural Engineering Research 9(4):303-306.

Mohsenin, N. N.
1968 Physical Properties of Plant and Animal Materials. Part Two, Volume One. Texture of Foods, Mechanical Damage Aero- and Hydrodynamic Characteristics and Frictional Properties. The Pennsylvania State University. 306-747.

Narayan, C. V.
1969 Mechanical checking of navy beans. Thesis for the degree of Ph. D., Michigan State University, East Lansing. (Unpublished)

National Association of Relay Manufacturers
1966 Engineers' Relay Handbook. Hayden Book Company, Inc., New York, 300 pp.

Perry, J. S. and C. W. Hall
1960 Germination of pea beans as affected by moisture and temperature at impact loading. Quarterly Bulletin of the Michigan Agricultural Experiment Station, Michigan State University, East Lansing. 43(1):33-39.

Perry, J. S. and C. W. Hall
1965 Mechanical properties of pea beans under impact loading. Transactions of the ASAE 8(2):191-193.

Saul, R. A. and J. L. Steele
1966 Why damaged shelled corn costs more to dry. Agricultural Engineering 47:326-329, 337.

Turner, W. K., C. W. Suggs and J. W. Dickens
1957 Impact damage to peanuts and its effects on germination, seedling development, and milling quality. Transactions of the ASAE 10(2): 248-251.

Wilson, T. R. C.
1922 Impact test of wood. Proceedings of the Twenty-Fifth Annual Meeting of the American Society for Testing Materials 22(2):55-73.

Zoerb, G. C. and C. W. Hall
1960 Some mechanical and rheological properties of grains. Journal of Agricultural Engineering Research 5(1):83-93.

APPENDIX I

EXPERIMENTS TO DETERMINE REPEATABILITY OF IMPACT VELOCITY

The data in the following tables summarize the results of tests run using the two velocity measurement systems simultaneously. Each trial consisted of twenty tests conducted at three minute intervals, which is typical of the time required to run an impact test. After the electric motor was started there was a fifteen second delay before the clutch was activated to accelerate the arm. The angular velocity of the primary shaft was measured by the tachometer. The time required for the arm to travel the distance between the magnetic pickups was measured by the electric timer. These values were used to calculate two values for the impact velocity in feet per minute for each test.

Table A1.1--Comparison of velocity measurement systems

Test Number	Tachometer fpm	Timer fpm	Difference fpm
1	2472	2515	43
2	2479	2505	26
3	2485	2509	24
4	2495	2513	18
5	2491	2509	18
6	2485	2513	28
7	2482	2512	30
8	2498	2514	16
9	2491	2511	20
10	2491	2506	15
11	2498	2507	9
12	2498	2515	17
13	2485	2511	26
14	2498	2512	14
15	2491	2515	24
16	2479	2508	29
17	2491	2513	22
18	2495	2517	22
19	2491	2513	22
20	2488	2515	27
95 Percent Confidence Interval	2489±5	2512±2	23±5

Table A1.2--Comparison of velocity measurement systems

Test Number	Tachometer fpm	Timer fpm	Difference fpm
1	3007	3007	0
2	3007	3016	9
3	2997	3018	21
4	3007	3008	1
5	3007	3010	3
6	3004	3004	0
7	3004	3006	2
8	3007	3004	-3
9	3004	2998	-6
10	3007	3006	-1
11	3004	3000	-4
12	3004	3007	3
13	3004	3010	6
14	3007	3011	4
15	3004	3005	1
16	3007	3005	-2
17	2997	3001	4
18	3004	3002	-2
19	3000	3010	10
20	3009	3009	0
95 Percent Confidence Interval	3004±2	3007±3	3±4

Table A1.3--Comparison of velocity measurement systems

Test Number	Tachometer fpm	Timer fpm	Difference fpm
1	3447	3517	70
2	3452	3513	61
3	3456	3522	66
4	3456	3508	52
5	3440	3505	65
6	3452	3524	72
7	3443	3502	59
8	3447	3488	41
9	3443	3504	61
10	3443	3512	69
11	3447	3507	60
12	3440	3518	78
13	3440	3523	83
14	3434	3496	62
15	3452	3503	51
16	3450	3481	31
17	3450	3509	59
18	3450	3517	67
19	3450	3509	59
20	3443	3487	44
95 Percent Confidence Interval	3447±3	3507±6	60±6

Table A1.4--Comparison of velocity measurement systems

Test Number	Tachometer fpm	Timer fpm	Difference fpm
1	5294	5243	-51
2	5294	5232	-62
3	5287	5271	-16
4	5287	5285	- 2
5	5290	5291	1
6	5290	5294	4
7	5290	5291	1
8	5290	5291	1
9	5294	5288	- 6
10	5294	5297	3
11	5297	5291	- 6
12	5294	5294	0
13	5290	5302	12
14	5294	5305	11
15	5290	5288	- 2
16	5294	5297	3
17	5290	5297	7
18	5294	5300	6
19	5290	5297	7
20	5290	5297	7
95 Percent Confidence Interval	5292±2	5287±12	-5±12