DESIGN, FABRICATION, AND CALIBRATION OF AN INSTRUMENTED DROP WEIGHT IMPACT TESTER

by

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ABSTRACT

In this thesis, the complete design, fabrication, and calibration of an instrumented drop weight impact tester is described. Included in this description are all the sketches and drawings that are needed to duplicate this project, if so desired. This impact tester was built for around $23,000 less than it would have cost to buy and modify a commercial tester for the intended research application. This tester, as designed, was intended to be used in the field of impact location detection using artificial neural networks. Even though this impact tester was built for a specific research application, the design concepts that are presented can easily be adapted to a variety of testing needs. This impact tester was built using an non-working milling machine for a base. This provides a rigid, stable base along with a moveable X-Y table. The tester itself has the capability for drop weights ranging from 3.518 lb up to 15.408 lb, and impact energy levels ranging from 0.6 ft-lb up to 45.6 ft-lb. Also, it is capable of impacting multiple locations of large plates with variable boundary condition sizes up to 12" x 24". Furthermore, it uses a computer program written using a data acquisition software package to provide output plots for the impact
event, including the force and energy applied to the specimen versus time and the force versus displacement. Finally, initial experimental results obtained from this tester agree very well with those obtained from a commercially available tester, allowing it to be used in future tests involving intelligent material systems.
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Chapter 1
Introduction and Background

1.1 Introduction

In this thesis, the complete design, fabrication, and calibration of an instrumented drop weight impact tester will be described. Throughout the extensive literature that has been published on instrumented impact testing, there were no references that could be found that went into in-depth detail describing the entire design process involved with building a complete instrumented drop weight impact tester. This thesis is an attempt to do just that. The purpose for this is that often times with specialized research applications, a reliable instrumented impact tester can be built for considerably less cost than if one was bought and modified accordingly. Furthermore, much of the time and difficulties associated with the construction of such a tester can be eliminated if the design and construction procedures are already documented, as will be done in this thesis. Each step of the construction process will be covered, so that if anyone desires to build their own impact tester in the future, the details will be all laid out. The actual impact tester that was built
was designed for a specific type of application, but the aspects of the basic design can be modified to accommodate other experimental requirements.

The idea to build an instrumented impact tester was thought up in order to conduct research associated with damage detection and alleviation in structures by using intelligent material systems. These systems, also called "smart" or "adaptive" structures, are a new class of advanced material systems which use integrated actuators, sensors, and controls to allow the structure to adaptively change and respond to external systems (Rogers, 1993). Basically, these intelligent material systems try to replace mass with energy. So instead of over designing a structure for a worst case scenario, the smart structure can release energy at critical times to prevent or minimize damage. The basic premise then is to use energy as opposed to mass to either strengthen or monitor the health of a structure. One area where this tester has a possible application is in work involving piezoelectric sensors (PZT's) as part of an artificial neural network. Artificial neural networks are known for their abilities in the area of pattern recognition. The neural network will hopefully be able to detect where an impact event occurred and possibly how severe the impact was. The neural network would first be trained by impacting the plate at several known locations and at known energy levels as provided by the instrumented portion of the impact tester. During this training, the response from the PZT sensors on the plate would be stored for each impact position. Then, when the plate is impacted at a random location, the neural network should be able to identify the general location where the impact occurred. Also, it is hoped that the neural network will be able to tell what the energy level of the impact was. This technology would have applications in such things as armored tanks. If the tank gets hit by a projectile, the neural network could identify where the impact occurred and how much energy was absorbed by the tank's armor. Then, other
damage identification techniques, or possibly an extension of this neural network approach, could be employed to determine how much damage was incurred. Initial work has already been done using PZT's and neural networks to detect delaminations between aluminum plates with bonded composite patches (Ganino, 1994).

Another application for this impact tester in the area of smart material systems is the recent work being performed on impact resistant composite plates with embedded shape memory alloys (Paine, 1994). Initial results from this work indicate a significant increase in the damage tolerance of composite plates subjected to impacts.

As can be seen by these examples, there are many applications where an impact tester would be useful in research related to intelligent material systems. It is for this reason that obtaining impact testing capabilities at the Center for Intelligent Material Systems and Structures (CIMSS) at VPI was pursued.

1.2 Background of Instrumented Impact Testing

Impact testing has been around for many years. Impact testers generally come in three forms. These are drop weight, pendulum, and gas guns. The drop weight and pendulum testers for the most part use gravity to adjust the velocity of the impact head. A pendulum tester consists of a swinging weight with an arm connected to a rotating hub. An example of this type of tester would be the Charpy testers that most engineers are familiar with. A drop weight tester consists of a weight that either falls inside a vertical tube or is guided by two vertical rods. Some drop weight testers employ a spring or pneumatic assist to get the head moving quicker (Tat Teh, 1993). Impact velocities for the free fall drop weight testers reach a maximum of around 9.9 m/s for a 15 ft industrial tester (SATEC, 1993)
down to around 4.5 m/s for a standard room sized tester (Dynatup, 1993). With pneumatic assist, the Dynatup 8250 impact tester can reach velocities of 13.41 m/s, while still being less than three meters tall. These velocities are all considered low velocity impacts. A gas gun, on the other hand, uses the release of a compressed gas to propel a projectile through a gun barrel towards the target at up to 500 m/s (Tat Teh, 1993). This is used more for high velocity impact testing, and the test is limited by the light weight of the projectile and the inability to completely instrument the system.

Instrumented impact testing technology is relatively new. Although instrumented impact tests were set-up as early as the late 1950's (Scheetz et al, 1960), there were less than five laboratories in the U.S. using this type of testing regularly in 1970. In 1972, the number of labs had grown to approximately 25, and in 1973 around 50 were using the tests (Ireland, 1973). This number has continued to grow significantly over the years, and today the instrumented impact test is the preferred method for testing how high rates of loading affect materials. The main reason for this is because of the more widespread use of advanced materials. In the plastics and composites industries, instrumented impact testers are used for material research, applications development, and process quality control. The reason for their widespread use is that this type of testing provides much more information about the material performance than was previously available. Modern instrumented impact test machines generate a complete record of 1) force applied to the specimen versus time, 2) displacement of the impactor (from the point of specimen contact) versus time, 3) velocity of the impactor versus time, and 4) energy absorbed by the specimen versus time. By using instrumented impact testing, a record of all these parameters can be examined to determine the four critical values for homogeneous
materials. These are the maximum load, the energy to maximum load, the total absorbed energy, and the deflection to maximum load (McMichael and Fischer, 1989).

The next question one might ask is, "Why use an instrumented impact tester?" The answer is because of the expanded knowledge of the impact event and accuracy of the results. Anyone familiar with materials testing knows what a Charpy impact tester looks like. It consists of a pendulum with a weight and an impact head at the end. A specimen is placed in a holder and the crosshead is raised to a certain height. This height is marked and the crosshead is released, thus impacting the specimen. After impact, the crosshead will swing back up to some new height. From measuring the difference between the two heights, the energy that the test specimen absorbed can be found. This type of test is great for general testing to determine if a material is brittle or ductile at various temperatures, but the exact mechanism of failure is not determined. By instrumenting the testing, much more can be learned.

With the advent of oscilloscopes and computers, a whole new door was opened as far as instrumentation capabilities are concerned. The impact test could now be performed in such a way that the force applied to the specimen at any given time could be determined (McMichael and Fischer, 1989). Further calculations would give the actual energy absorbed at any given time. The three basic components required for an instrumented impact test are the impact machine, the load sensor, and a signal display component such as an oscilloscope or computer (Ireland, 1973).

As has been stated, most applications for instrumented impact tests today are with plastics and composite materials. Composites are beginning to gain widespread
acceptance in the aerospace industry. This leads to a whole range of new problems where instrumented impact testing could be beneficial during design. Dropped tools, bird strikes, or rocks and debris during takeoffs and landings all can cause damage such as matrix cracking, delamination, and fiber breakage in the material. This damage significantly reduces the structural integrity of the composites. The main reason impact is an important consideration with composite materials is that they are very strong in-plane, but especially susceptible to transverse impact (Tat Teh, 1993). When isotropic polymers or metals are impacted, the energy is absorbed mostly through plastic deformation. As opposed to most metals and thermoplastics which are highly ductile and damage tolerant, composites are inherently brittle materials. This lack of ductility is a real concern when designing with composites (Wardle and Zahr, 1985). In order to dissipate the required energy, composites undergo a process called delamination, with fiber breakage occurring at higher energy levels. Delaminations are where the bonding between layers develops cracks which will propagate until the energy has been absorbed. This process may or may not be visible on the surface. One result of these delaminations is a significant reduction in strength, especially under compression loading. In this case, the load carrying plies do not have any structural support from nearby plies so the result is premature failure through crippling or buckling (Tat Teh, 1993). By using instrumentation during the impact tests, this process of crack initiation and propagation can be observed (Driscoll, 1985).

There are many standardized tests for impact testing of polymers and composites and the many commercially available testers are geared toward this type of testing; however, in research, one is not always interested in such tests. It is often desired to pursue different geometries or boundary conditions. This flexibility along with a reduction in cost are two
of the main reasons to pursue the custom building of an impact tester such as was done here.

This thesis will cover all the elements necessary to build a versatile drop weight instrumented impact tester at a very reasonable cost as compared to commercial packages. The actual cost and assembly could vary significantly from person to person depending on the availability of a machine shop like on a university campus, and the machining experience of the individual overseeing the design and construction. The author of this thesis had previous machine shop experience, which allowed the construction of almost every part of the tester without any labor costs other than the time and cost associated with a Graduate Research Assistantship.

The body of this thesis covers the four main areas associated with building the tester. In Chapter 2, the process of locating, purchasing, and preparing a suitable test bed on which to build the tester is presented. Then, in Chapter 3 the design and fabrication of the drop weight portion of the tester is discussed. In Chapter 4, the design and fabrication of the specimen clamp is presented. Next, Chapter 5 covers the instrumentation associated with the instrumented impact tester. Then, the calibration process of the completed test apparatus is examined in Chapter 6. This consisted of testing identical specimens on both the constructed impact tester and a commercial Dynaup 8200 tester and comparing results. Finally, in Chapter 7 the accomplishments and conclusions are given along with some recommendations for future work. It is hoped that by presenting not only the process that was followed for the design, fabrication, and calibration of the tester but by also presenting some of the mistakes that were made along the way that the individuals who use this thesis can avoid the same problems that were encountered by the author.
Chapter 2
Choosing a Suitable Test Bed

2.1 Reasons for Building an Instrumented Impact Tester

When this project was first proposed, the initial studies consisted of not only finding out what type of impact tester would best suit the research needs at the Center for Intelligent Material Systems and Structures, but also whether or not a suitable instrumented impact tester could just be purchased. After all, if a tester that completely suited all the Center's needs was already out there, why not just purchase it. In other words, why reinvent the wheel? The answer to this question became very clear once initial investigations into the commercial packages were completed. More about that later.

The first order of business was to develop some initial guidelines regarding the tester, in order to know what to look for. The main reason that CIMSS needed an impact tester was to be able to impact specimens at different locations without redoing the boundary conditions after each individual test. The instrumentation was required to provide some
information about the impact event itself, such as the force applied or the energy absorbed throughout the event.

One thing that had to be determined for this tester was the maximum specimen plate size that it should be required to accommodate. From discussions with Dr. Jim Sirkis, a professor in mechanical engineering at the University of Maryland (UMD) performing similar work, and Dr. Harry Robertshaw, a professor in mechanical engineering at Virginia Tech, it was decided that rather large plates would be best because the large boundary condition would be closer to actual real life situations. Additionally, the larger specimens would provide a more distinct signal if various sensors were going to be used. This is because the vibrational patterns would have larger magnitudes. It was also decided that the plates should not be perfectly square. With a rectangular plate, the situation would again be more like real life. Furthermore, the first two modes of vibration are at the same frequency with a square plate. The initial estimate of a suitable plate size was decided to be around 18" x 26". This of course was just a rough guideline. It almost goes without saying that since the whole purpose of having an impact tester for CIMSS was to test multiple locations of the plates. The tester must therefore have some means to move the plate around with as large a range as possible.

Next, the appropriate type of impact tester needed to be decided upon. As described in the introduction, there are three main types of impact testers. The drop weight tester was the one chosen for this project. The reason for this is that it was felt that this type of tester was the best for the research that was to be conducted. The drop weight was chosen over the pendulum tester because of the restrictions that the vertical plate orientation had on specimen mobility. Recall that rather large plates were going to be
used and they needed to move as much as possible to test the majority of the plate. The drop weight type was chosen over the gas gun because most of the time the specimen would not be severely impacted at high velocities resulting in through penetration and high damage levels. For the applications at CIMSS, most of the tests would require the impactor to undergo rebound, where the specimen bounces back off the specimen with minimal or no penetration. This type of testing is also called "low blow" testing (Ireland, 1973). Also, more thorough data from the instrumentation can be obtained from a drop weight type of tester.

2.2 Investigation into Commercial Packages

This project was specifically geared from the beginning towards research applications, not standardized tests. For this reason, a non-moveable specimen would not be acceptable. The first thing that was pursued once the initial system requirements were decided upon was the possibility of just purchasing a commercial instrumented drop weight tester. Many companies that manufacture and sell the testers were contacted. A complete listing of impact tester makers with phone numbers was found in the Advanced Composites magazine's 1993 Bluebook. There were nineteen companies listed, of which about five sold drop weight testers, and only three (Dynatup, SATEC, and Kayeness) fit most of the established size and energy range requirements. The big issue with all the companies was that their testers were again all built for standardized tests and extensive modifications would have been required to make the testers compatible with the research needs at CIMSS. The Engineering Science and Mechanics Department (ESM) owns a Dynatup model 8200 drop weight tester, so it was very beneficial to physically look at it and see if it would be acceptable. The major modification that these testers would have needed would have been to incorporate some sort of set-up to move the specimens. For
example, an X-Y table would have worked well, but this would have been very expensive. Also, to make this table an integral part of the tester would have required extensive modifications.

That brings us to the next issue associated with purchasing an instrumented impact tester, the cost. Not only would the tester need to be purchased, but so would the X-Y table. Also, extensive modifications would have been required, which would have added even more cost. Quotations were obtained from the three companies that produce suitable impact testers. Costs from the different companies for the machines and electronics packages are as follows:

- SATEC $21,245.00
- Kayeness $21,000.00
- Dynatup $24,500.00.

These systems all offered similar features; however, the Dynatup matched the project's requirements the closest and would have required the least modifications. When the estimated cost of modifications ($1,000) and the cost of an X-Y table ($3,000) were added in, the total cost for the commercial impact testers ranged from $25,000 to $28,500. Because of this high cost, it was decided to pursue other options. Specifically, building an impact tester from scratch.

2.3 Choice of a Test Bed

Once the decision was made to build an impact tester, there were many factors to consider. Dr. Sirkis and some students at UMD built a drop weight impact tester for work similar to what was planned for this project. In fact, both of these groups (UMD and VPI) are funded by the same source. The goal was to take different approaches
toward achieving the same goal of impact detection. Their tester was copied from a
similar tester at the Naval Research Lab. It was decided to pursue a design similar to the
tester at UMD, but changes were desired to make the tester at CIMSS more versatile. A
trip was taken by the author to UMD to see their tester and gather some preliminary
design data. This was just an attempt to gain an overall picture of how the tester would
look. The tester was not completely finished at the time of the visit, but it was completed
sufficiently to observe the basic layout. Their tester was built off of a wall, and they were
going to use an X-Y table that they had purchased for a previous project to move the
specimens. A picture of the UMD tester is shown in Figure 1.

The actual design process of the tester that was built at VPI will be discussed in
Chapter 3. The first thing that needed to be done was to find a suitable test bed, which
could also move the plates. Once this was found, the rest of the tester could be designed
and built around it. As stated previously, a suitably sized X-Y table costs around $3,000.
One additional requirement for the base platform was that it be rigid. The reason for this
is to minimize the energy dissipated by the base. In other words, the amount of energy
that is lost during the impact should be mostly absorbed by the specimen plate, not the
surrounding structure. An X-Y table would accomplish this adequately, but again the cost
was too high. The solution to this problem came to the author one day. The impact tester
could be built using a milling machine with the head removed. This would provide a very
rigid base, a precise X-Y table, and would be relatively inexpensive if a suitable milling
machine could be found.

Now that the base structure had been determined, an old milling machine had to be
found. It was believed that this search process would not take too long, but how wrong
Figure 1: Picture of the UMD Impact Tester.
that thought was. The requirements for the milling machine were that it had to have a large enough table to handle the pre-described plate size, or close to it. Also, it had to have sufficient table mobility to test the majority of locations on the plates. Finally, it had to be inexpensive. Preferably, the milling machine head should not even have worked since this would bring the cost down.

A milling machine was first located at a federal surplus yard in Richmond, VA, but the table was too small. After much extensive searching, a suitable milling machine was finally located in Salem, VA. There were actually two suitable machines at the one location. One machine had a table that was 8 3/8" x 36 3/4" and had a mobility of 7 5/8" x 24". It was a very old machine from the days when machine shops were powered by central belts running along the ceilings. This was good because it did not have a real working head. This was just what was desired. It cost $300. The other machine was a little bigger and had a table size of 12" x 42" with approximately 13" x 22" mobility. Its cost was $400. Both these machines were very well suited to the application, but it was decided to purchase the slightly smaller and cheaper of the two. Its exact weight was not known, but it was estimated to be around 1000 pounds. The fact that it appeared to weigh a little less than the larger machine was another reason the smaller machine was chosen. This weight concern will be discussed later in this chapter. A picture of the milling machine is shown in Figure 2.

2.4 Preparation of the Milling Machine Base

A heavy truck was used to go pick up the milling machine. The machine was taken to the ME shop so it could be worked on. The machine had always been indoors, but it was extremely dirty, greasy, and it had what appeared to be sawdust all in its gears. The first
Figure 2: Picture of Milling Machine in Initial As-Purchased State.
order of business was to remove all unnecessary parts, in an attempt to reduce weight. An external gearbox was removed as were a large rod for tooling and some other gearing. The total weight taken off of the machine was estimated at 300 pounds. At this point the actual weight still was not known, so the main motivating factor in the material selection and design was to keep the weight down.

At this time, it would be best to present a overhead schematic of the milling machine which shows the table mobility, table size, and modifications that were made. It also shows what is defined as the front, back, left, and right sides of the X-Y Table. This picture is shown in Figure 3. This picture will come in handy when describing the individual preparation processes.

The next step once the external parts were removed was to begin modifying the actual structure itself. To proceed with the building of the impact tester, a stable platform from which to build off of was needed. The best way to accomplish this was to cut off the cast iron head at two small "neck" locations. These "necks" were oval shapes about 7 1/8" x 2 5/8". An appropriate location was found and marked, making sure the two cuts would be made at about the same height in order to reduce the amount of finishing work that would be needed after the cutting. A portable bandsaw was used to make the cuts.

Having accomplished this task, the next job was to grind flat the two locations where the cuts had been made so a steel plate could be mounted. This would provide a stable platform on which the rest of the machine could be built. This operation consisted of first leveling the machine itself. Then, a hand disk grinder was used to not only flatten and level each cut "neck" piece, but also to make sure that the tops of the two pieces were
Figure 3: Top View Schematic of Prepared Milling Machine Base.
level with each other. Having completed this, it was time to mount the steel plate. This plate had to be rigid enough to support all the loads that were going to be put on top of it, but at the same time light enough to keep the overall weight of the tester to a minimum. Steel was chosen because of its rigidity, and a thickness of 1/2" was chosen because it practically seemed to be the best thickness. As it turned out, this was a good choice. The plate had to be long enough to cover both of the cut off "neck" posts. This was so it would have something to bolt into to secure it. Also, by knowing the approximate dimensions of the rest of the tester from initial design work, the appropriate width was purchased. A 32" x 23" x 1/2" low carbon steel plate was chosen. Another option would have been to use a thinner plate with stiffeners. This option was not chosen because it would not have been quite as rigid and not as easy to bolt other things into it.

The next thing done was the attachment of the steel plate. The two "necks" were the only two place that the plate could be attached to so the options were limited. The decision was made to use two 1/2" diameter bolts on each "neck". The plate was first set on the two prepared supports, and the locations of the holes were carefully determined. Then, holes were drilled using a portable drill press with a magnetic base to keep the drill press, and likewise the holes, straight. These holes were drilled to a diameter slightly less than the 1/2" bolts would require. This was to allow for a Normally Coarse (NC) tap to be used to produce the 1/2" threaded holes in the two "necks." Then, the holes in the plate were drilled out to 1/2", and the plate was mounted.

One additional step in preparing the milling machine base for use as a test bed was to clean the entire tester. This consisted of just good old fashion hard work. A degreaser was used at first. Then, a substance similar to WD-40 was used to remove all the sawdust
and old grease from the gears. The X-Y table did not move as freely as it should have because of these contaminants in the gearing. The whole table should have been removed for cleaning and inspection. The problem was that the machine was so old that the proper way to remove the table was all corroded solid. As it turns out, the gearing was cleaned very well. Once everything was clean, new grease was applied to all the gears. After this, the table moved very easily in the front to back direction, but it did still not move quite as easily as it should have in the side to side direction. It did move much better than it did before cleaning all the debris out of its gears. Also, the table began to move more freely the more it was used. This indicates that there was just a lot of particulate in the gears due to sitting for a long time without being used. The table should continue to loosen up in this side to side direction with continued use and should not be a problem in the future. Finally, the gearing should be lightly greased periodically to further aid the table's movement.

One more thing done to the base to prepare it was to remove a piece of metal that was under the X-Y table. The metal was supposed to cover a slot over the gearing that moved the table front and back and up and down. This plate, however, was not staying in place. Instead, it stayed in its correct place when the table was adjusted inward; but, when it was cranked back out, it would move with the table. This caused about 4" of the plate to stick out when the table was being adjusted outward toward the operator. This metal would then hit the handle used for this adjustment. This in turn would not allow the table to move any further outward. Attempts to pull this piece of metal out were unsuccessful, so the piece was just broken off right at the edge of the table. This did the trick, and now the table can move its full 7 5/8" in that direction.
The next step in the preparation process was to build a stable structure on top of the steel plate from which to build the rest of the impact tester. Two I-beams were chosen because of their stiffness in all directions. Again, to keep weight to a minimum, aluminum I-beams were used. The I-beams were pre-cut to allow for easier shipping. This saved transportation costs and speeded up delivery. The length of the I-Beams was dictated by the height of the ceilings where the machine would be located. Obviously, the higher the I-beams, the higher the drop height could be made. The ceiling at CIMSS is 9 feet, and it was 44" from the floor to the top of the steel plate. So, the I-beams were chosen to be 70" for a total height of 8'8". Considering future design dimensions, the face had to be at least 6" wide to attach some cantilever members which will be discussed in Chapter 3. The only I-Beams that could be obtained with this face width were 6" x 6".

These I-Beams were mounted vertically using 2" wide pieces of 2" aluminum angle and one 6" long piece of 3" aluminum angle. The 3" piece was placed along the side opposite the X-Y table because the cantilever members were going to be attached onto the front side. The rest of the tester was going to be built off of these cantilever arms, which would create a moment at the rear base of the I-beams. So, the more rigid 3" angle was used there. Then, two of the 2" angle pieces were mounted at the front base of the I-beams. Two were also mounted at the sides with one common bolt going through the tops of both pieces and through the web of the I-beam. All the aluminum angle pieces were held in place by drilling through the steel base plate and using 3/8" bolts with lock washers and nuts to anchor them. The I-beams were located 9 1/8" back from the edge of the plate closest to the X-Y table, and the outside edges lined up with the side edges of the steel plate. This distance was arbitrarily chosen in order to allow enough room for the test
plates to move under the soon to be installed cantilever arms without hitting the I-beams. This distance just turned out to be 9 1/8".

The installation of the I-beams was not as simple as just bolting them onto the steel plate. They had to be relatively perpendicular to the plate (which was already level with the X-Y table) on all sides, and they also had to be close to parallel with each other. This alignment process consisted of two different processes. First, the I-beams were cut by the distributor to the correct length, so their ends were already pretty square. The steel plate was also relatively flat. It would appear then that the I-beams could just be mounted on the plate and they would automatically be vertical. This was not the case. Since the I-beams were five feet long, even a slight variation in either of the two surfaces caused a pretty significant tilt at the top of the beams.

To make sure the beams were perpendicular to the milling machine table, a square was used along the front side of the I-beams using the X-Y table as a reference surface. At this point, the aluminum angle pieces described previously were already attached to the I-beams and holes had been drilled in the plate. Then, by tightening the angle pieces down to the plate in different order, the beams could be tilted slightly forwards or backwards. For example, by tightening the front angles first, the I-beams would tilt forward. Conversely, if the back piece of angle was tightened first, the beams would tilt backwards. Of course, after one side is tightened, the I-beams will pull back slightly when the opposite side is tightened. Therefore, one must overcompensate somewhat when trying to find the correct adjustment. This same adjustment procedure works to a lesser extent on the side to side tilt. The I-beams were already pretty close to perfectly vertical in this direction when they were installed, so not much was required for this adjustment.
Finally, to ensure that the I-beams were perfectly parallel with each other, a 15" long piece of 2" aluminum angle was bolted to the top inside back edge of the beams. The distance that the bases of the beams were apart was measured, and the 2" angle piece was bolted into place with the two I-beams this exact same distance apart at the top. This ensured that the I-beams were nearly perfectly parallel.

The final step in the preparation of the milling machine was to chip off all the old paint, apply primer, and paint the whole milling machine base and steel base plate. Over the years, coat after coat of paint had been applied to the structure. This resulted in a ugly, chipped appearance. A putty knife was used to remove nearly all of the old paint. The surface was then sprayed with the same WD-40 substance used earlier in order to remove any remaining oil on the surface. Then, a coat of primer was applied to both the base and the attached plate. This tester sat outside for most of the construction phase because there was not sufficient space in the machine shop for it. Because of this, a rust inhibitor primer and paint was used to prevent the structure or the steel plate from rusting. Even though the machine was always covered with plastic when it was outside, water still got under the plastic and caused some slight rusting before the paint was applied. This rust was removed with emery cloth and the WD-40 like substance. The milling machine was now ready to accommodate the rest of the impact tester. Pictures of the milling machine base with the steel plate and I-Beams installed, but without the new paint added or the top cross member installed, can be seen in Figures 4 and 5.

2.5 Weight Concerns

In an effort to remove all excess weight, a torch was used to cut off extra steel from the plate behind the I-beams. The shape of this cut could be seen in Figure 3. This steel
served no purpose and just contributed to the tester’s weight. This issue of weight was of
great concern. In fact, one reason that the smaller milling machine was chosen was
because it weighed less. The research lab at CIMSS is on the second floor of a leased
building. The building was not designed or zoned for industrial use, so the floor was
potentially not strong enough to support a large milling machine. The owner of the
building was contacted regarding the maximum weight that could be placed on the floor.
The owner referred the concern to the building’s designers. It was determined that the
maximum weight that could be placed on the lab floor was 70 lb/ft². A safety factor was
probably included in this design limitation, but still the maximum load should probably not
have exceeded 100 lb/ft². The milling machine base was 28 1/2” x 42” or 8.07 ft². Using
the initial estimated weight of 1000 pounds, the floor would undergo a load of 123.9
lb/ft². Remember that this is just an estimate, so the decision on whether or not to attempt
to put the tester in the lab could not be made until the exact weight was known.

Because of this weight uncertainty, it was necessary to find a suitable scale to weigh
the machine. The only scale that was readily available only read up to 600 pounds. The
milling machine pegged this scale at its maximum, so again only an estimate of its actual
weight could be made. The project progressed for about three more weeks before a scale
was located that could handle the machine’s weight. This scale was located at the power
plant at VPI. This scale was used to weight trucks with loads of coal. The machine was
transported by forklift and placed on the scale. The machine already had the steel base
plate and I-beams installed. It was determined at this point that the machine weighed
1580 pounds. This works out to 195.8 lb/ft², or more than twice the rating of the lab
floor. Furthermore, there was more construction to come. It was estimated that another
60 pounds would be added before the project was completed.
There were some options that could have been attempted at this point to distribute the weight, but it was felt that none would perform this task adequately. Therefore, it was decided that the tester would not be put in the lab at CIMSS and other options had to be pursued. It should be noted that much of the design and some of the fabrication had already been completed by the time the actual weight was determined. Additionally, most of the material had already been ordered. So, for aesthetic and cost reasons, it was decided that the tester would continue to be built out of aluminum. Some cost savings could have been had if the actual weight had been determined earlier, because steel, which is less expensive than aluminum, could have been used for most of the parts.
Figure 4: Angled View of Prepared Milling Machine Base with Steel Plate and I-beams Installed.

Figure 5: Side View of Prepared Milling Machine Base with Steel Plate and I-beams Installed.
Chapter 3
Design and Fabrication of Tester Components

3.1 Material Acquisition

As with any design project, one must always plan ahead and order any parts that it is felt will be needed in the near future. It is often difficult to know exactly how much material will be needed, and as a result, there is usually material left over after the project is completed. The amount of material that would be needed for this project was determined from initial design ideas and sketches along with some of the sketches and dimensions from the UMD design, and the aluminum was ordered. This order was placed before the milling machine base preparation was complete, so it included some parts already described, like the I-beams and structural angle. The order consisted of the following:

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Length</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>5'</td>
<td>6&quot; x 6&quot; wide flange beams</td>
</tr>
<tr>
<td>2</td>
<td>4'</td>
<td>2&quot; structural square tubing w/ 0.125&quot; thick wall</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(type 6063-T5)</td>
</tr>
<tr>
<td>Quantity</td>
<td>Length</td>
<td>Description</td>
</tr>
<tr>
<td>----------</td>
<td>--------</td>
<td>------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>1</td>
<td>2'</td>
<td>2&quot; x 2&quot; x 1/4&quot; structural angle (type 6061-T6)</td>
</tr>
<tr>
<td>1</td>
<td>3'</td>
<td>3&quot; x 3&quot; x 1/4&quot; structural angle (type 6061-T6)</td>
</tr>
<tr>
<td>1</td>
<td>1'</td>
<td>2&quot; square bar (type 2024-T351)</td>
</tr>
<tr>
<td>1</td>
<td>2'</td>
<td>3&quot; x 1/2&quot; rectangular bar (type 2024-T351)</td>
</tr>
<tr>
<td>1</td>
<td>2'</td>
<td>1 1/2&quot; x 1&quot; rectangular bar (type 2024-T351)</td>
</tr>
</tbody>
</table>

As it turned out, this was not enough material and more aluminum was ordered. Also, not all of this aluminum was used in the final design. The 2" square bar and the 3" x 1/2" rectangular bar were not used. Another order was placed at a later date once the design was finalized. This order consisted of the following:

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Length</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>25'</td>
<td>2&quot; x 2&quot; x 1/4&quot; structural angle (type 6061-T6)</td>
</tr>
<tr>
<td>1</td>
<td>12'</td>
<td>2&quot; x 1&quot; rectangular bar (type 2024-T351)</td>
</tr>
</tbody>
</table>

Full pieces had to be ordered this time as a different salesperson handled this order. He indicated that the standard length pieces could not be cut up and sold separately, as had been done previously. This was fine since a real good deal was worked out which actually made the bulk aluminum cheaper than if it had been ordered in smaller sections. The angle was cut into three 8' sections and the bar was cut into two 6' sections for ease of shipping. Because so much material had to be ordered, there was some left over scrap after the project was completed. The left over parts consisted of one 8' section of angle and about 9' of the rectangular bar. There were also some other small scrap pieces left over. From experience, having leftover parts from design projects has usually been found to be the norm. After all, having extra is always better than not having enough material.
and having to delay the fabrication schedule. Plus, this extra material will likely be used by the lab in the future. Even if it is not, it can simply be recycled.

3.2 Installation of the Cantilever Arms

Once the base was prepared and the material had been ordered, it was time to design and build the rest of the mechanical portions of the impact tester. As was mentioned previously, the basic design was similar to the tester built at the University of Maryland; however, many of the components were redesigned to add flexibility. Specific concerns of the UMD design were that it did not have a wide enough impact energy range and that it could not be moved because it was built onto a wall. Also, the UMD tester's impact energy range only went up to 18 Joules or 13.3 ft-lb. The tester that is going to be described here is also loosely modeled after the commercially available Dynatup 8200 instrumented impact tester. As was found out during the design process, one advantage of building a tester from scratch is the ability to pick and choose which features to incorporate into it. One additional design consideration was that the tester had to be built with regard to the dimensional limitations of the milling machine base. Much of the design work was actually done concurrently with the base preparation. Therefore, the basic dimensions of the tester, such as where the I-beams were positioned, etc., were known.

The first step was to mount four cantilever arm assemblies onto which the guide rails would be mounted. For ease of description, it would probably be best at this time to show the basic layout the tester. This will give a clearer picture of what is being described in this chapter. The front, side, and top views can be seen in Figures 6, 7, and 8, respectively. These are just general overviews with the design features, hence they are not drawn to scale. Details of the various components will be given in the remaining sections.
of this chapter. As stated in Chapter 2, the front of the I-beams were set back 9 1/8" from the front of the steel base plate. This was an arbitrary distance that was believed to give enough room to accommodate the specimen clamp, which will be described in Chapter 4. Both of the I-beams were mounted pretty much vertically and at about the same distance back from the front of the plate; however, by taking measurements from the I-beams to the table, it was learned that one beam was 1/16" closer. This was no problem, but it was beneficial in making sure the tester was uniformly made. The size and mobility of the milling machine table could also be measured. Furthermore, the length from the flat mounting surface of the shaft support blocks to the centerline of the shaft was also known from the manufacturer's drawings to be 1 1/4". By knowing all these dimensions, the exact length that the cantilever support arms needed to be could be found. This distance had to be chosen to put the guide rails exactly over the center of the X-Y table when it was in its exact middle position. This was required in order to take full advantage of the table's mobility when testing the plate specimens. From the calculations, the arms had to be 17 1/8" long.

Two inch hollow square tubing was used for these arms. This was chosen because it is rigid in all directions and other parts could easily be bolted onto it. Also, this is the same type of material that was chosen for the UMD tester. Since there would be some weight on the ends of these cantilever arms, some basic calculations were performed to find out if there would be excess downward deflection of the free ends. Using elementary mechanics of materials for a fixed-free beam and a weight of approximately 25 pounds distributed over four beams, the deflection was found from the formula (Gere and Timoshenko, 1984)

\[ \delta_{end} = \frac{PL^3}{3EI}. \]  

(1)
Figure 6: Front View Schematic of the Impact Tester.
Figure 7: Side View Schematic of the Impact Tester.
Figure 8: Top View Schematic of the Impact Tester.
The moment of inertia for the 2" square tubing with 0.125" walls was found to be $I=0.303$ in$^4$. The material was 6063-T5 aluminum which has a modulus of $E=10 \times 10^6$ psi. Therefore, the deflection of the free ends of the tubing would be only 0.0034". This was clearly an insignificant amount over the 17 1/8 " lengths of the pieces.

The next step was to mount the square tubing. The tubing was first cut to the proper lengths. Then, the mounting support pieces were cut. For this, 2" long pieces of 2" aluminum angle were used. The mounting and adjustment procedures were very similar to those used when the I-beams were added. Four pieces of angle were cut for each piece of tubing. Then, holes were drilled through these and the tubing and the mounting pieces were attached flush with the ends of the tubing using 3/8" bolts with nuts and lock washers.

Next, the arm assemblies were put at their correct locations and attached to the I-beams using C-clamps. Holes were then drilled through the mounting pieces and the I-beams. Note from the front view in Figure 6 that only one 3/8" bolt was used for each of the side holes, while two 1/4" bolts were used on each of the top and bottom support pieces. This was done because the web of the I-beam would have been in the way if a single hole had been used directly in the center. Again as was the case with the I-beams, there were some slight adjustments in the side to side and up and down tilt of the cantilever arms that could be made by tightening the supports in different orders. The arms all had to be parallel with each other as well as being parallel with the top surface of the milling machine table. To make sure that the arms were parallel to the milling machine table, a square was used to measure the height at both the fixed and free ends of the arms. This height should have been the same. To ensure that the arms were parallel with each
other, the distances between the fixed ends of the arms at the same level (i.e., the lower arms or upper arms) were checked to make sure they were the same as those at the free ends. The cantilever arms were then all inspected visually as a final alignment check.

The next step was to connect the top two cantilever support arms together as well as the bottom two. This was done using two cross support bars made out of 2" aluminum angle. The lengths of these pieces were determined by simply measuring the outside edge to outside edge distance of the tubing at the fixed ends. This turned out to be 19". The cross support bars were then cut to this length and attached using 3/8" bolts and nuts, making sure the free ends were the same 19" apart. Since care had been taken to assure the proper alignment of the cantilever support tubing, attaching these cross support pieces did not require much effort. These pieces really tied the framework together, making it good and rigid.

From looking at the figures, a few important things can be seen regarding the cantilever support arms. The first is that the tops of the upper arms were mounted 4" from the tops of the I-beams. Thus, with the addition of the 2" tall cross support bar, there would be 2" from the top of the I-beams to the rest of the tester. This was done for a reason. At this point in the project, the exact weight of the tester was still not known, and one of the options for distributing the weight was to put the tester up on something. Therefore, if this had to be done, the top 2" of the I-beams could be cut off. This would then allow for the tester to be raised up to 6" and still fit in a room with a 9' ceiling. The arms could have been set up an additional 2" thus increasing the maximum tower drop height accordingly had it been known at the time that the tester weighed too much for the
lab. Also, the I-beams and rails could have been made longer, thus further increasing the drop height.

The other design consideration that should be noted is that the lower cantilever support arms were mounted 6" above the steel base plate. This was a distance chosen in order to allow the specimen clamp to fit under these arms without any interference. At this point the specimen clamp had not been designed yet, so this was a judgment call. As it turned out this was just enough distance.

3.3 Installation of the Guide Rails

The next step in the design and fabrication process was to determine what to use with the falling weight to provide as close to free fall as possible. In Dynatup's model 8200 drop weight impact test machine, brass bushings are used with guide rails. The University of Maryland used linear ball bushings to guide their falling head down rails after they had determined that non-ball bushings caused binding problems. One cause of this binding might have been the fact that they were only using one block for a falling weight. Through conversations with a Dynatup representative, it was learned that the length between the top pair of guide bushings and the bottom pair should be greater than the distance that the two bushings in each pair are apart. In other words, if there are four bushings in a rectangular configuration, the vertical sides of the rectangle should be longer than the horizontal sides. This would keep the binding to a minimum. This would explain the binding problems encountered by UMD, since they were only using two bushings with a single falling head. Because one of the optional drop configurations of this tester would be a single falling block with only two guide bushings, it was decided to also go with the ball bushings to reduce binding.
The main determining factor in this selection was that it was felt that the ball bushings would produce less friction; however, Dr. Sirkis at Maryland had mentioned that their group had difficulty aligning the guide rails. While perusing the bearing catalog, the idea to use self-aligning ball bushings came to the author. This solution would solve two problems. It would assure low friction during the drop and it would assure proper rail alignment. With self-aligning ball bearings, the rails do not have to be perfectly aligned, just relatively close. The bearings will adjust for any slight alignment variances.

Knowing now what was going to be used, four Thompson model Super 12 self-aligning ball bushings were ordered. These bearings have an inner diameter (ID) of 0.75" and an outer diameter (OD) of 1.250". Also, using these dimensions, the proper guide shafts were ordered. These were 3/4" Thompson 60 Case Shafts. These shafts have a Rockwell hardness of 60-65C, and have a straightness within 0.001/0.002" per foot. Two of these shafts were factory ordered in 49" lengths. This length was dictated by the distance between the two existing cantilever support assemblies. This length was 48 3/4" and an additional 1/4" was added for good measure.

These shafts had to be held to the support structure somehow. The contingent at UMD designed and built their own shaft support blocks, and the same was going to be done here; however, while browsing through the Thompson Linear Motion Technology Guide, it was determined that these shaft support blocks were available pre-made by the company for only $16.80 each. There was no way that they could be made that cheaply, and buying them would save time. Therefore, four model ASB shaft support blocks were purchased. These blocks basically just hold the shaft in place and allow it to be anchored to the surrounding structure.
The next step was to find a way to attach these shaft support blocks (and rails) to the cantilever support structure. The key to this would be to come up with a way to be able to adjust the side to side position of the blocks to facilitate easy alignment of the rails. The solution was to use a 1/4" end mill and cut four slots in the cross support bars, two in the top one and two in the bottom one. The purpose of each slot was to allow one shaft support block to move a certain side to side distance. These slots were cut 2.5" long. The two mounting holes on the shaft support blocks were 2" apart, so this allowed 1/2" of side to side adjustment of the blocks. By knowing that the centerlines of the rails were going to be 5" apart, the correct locations for these slots could be determined. The two slots were to be equidistance from the centerline of the cross support bar. Furthermore, the top shaft support blocks were designed to be flush with the top of the top cross support bar and the tops of the two bottom shaft support blocks were designed to be flush with the top of the bottom bar. In actuality, these bottom blocks could have been mounted part way down the bottom cross support bars, which would have increased the drop height by 3/4". This was not realized until after it had already been done and the guide shafts had already been received. Also, a math error resulted in the misplacement of the slots in the first two cross support bars. Again, these little mistakes are all part of the design process. This problem was corrected, and new cross support bars were made. A drawing of one of the cross support bars with the machined slots can be seen in Figure 9. Also, Figure 10 shows a picture of the shaft support block that was purchased from Thompson.

3.4 Design of the Quick Release Mechanism

The first item that was designed and built that went on the guide rails was the quick release support. The purpose of the quick release is to hold the falling weight at a certain
Figure 9: Cross Support Bar with Machined Slots.

Figure 10: Shaft Support Block Purchased from Thompson.
height. Then, when testing is ready to proceed, it releases the weight as smoothly as possible.

The UMD design was used for this part with a few modifications. The main change was the use of an electromagnet instead of a mechanical latch to release the falling weight. The UMD designed mechanical latch could have been used, but it was not for a few reasons. As the design of this quick release was in its initial stages, the exact same design was going to be used with the exception that it would be made out of steel as opposed to aluminum as theirs was. The reason for this change was that even in the initial stages of use, the UMD quick release latch was showing signs of wear.

One day while going over the process that would occur during testing, it was decided that the mechanical latch would not be the best thing to use to hold and release the weight. Instead, an electromagnet should be used. This would allow for easier testing, and it would allow the weight to free fall right from the very start. The idea for this came from a paper in which a similar electromagnet release is described (Jermian et al, 1985). The main reason that an electromagnet would allow for easier testing is because of the height of the rails on this particular tester. Since the tester was built on an old milling machine base, the rails are a good distance above the ground. In fact, the top of the rails are nearly 9 feet off the ground. Because of this, a step ladder needs to be used to raise the weight to the quick release if it is to be dropped from the higher levels. Then, since at this time there is no mechanical rebound catch on this tester to catch the weight after it has bounced off the specimen, the operator would have to reach up and release the latch then hurry to down to the specimen and try to catch the weight on the rebound. This would be very difficult if not impossible. By using an electromagnet, the weight can be raised to the
proper level using the step ladder and held in place by the attractive pull of the magnet. Then, the operator can descend the ladder and push a button to release the weight. Since the person is already at the same level as the specimen, the falling weight can easily be caught on the rebound.

As stated, no mechanical rebound catch was designed for use with this tester. A Dynatup representative indicated that they offer a mechanical catch on one of their testers, but there were numerous problems associated with it. If this catch mechanism was accidentally not reset before a test, the force of the falling weight would break the part. This catch is the same one that is on the Dynatup model 8200 in the ESM Department. The company now offer a more reliable pneumatic catch, which uses a compressed air powered mechanism to catch the rebounding weight. It was decided that for now the weight would just be caught by hand. This is how the UMD is doing it. Because the electromagnet allows the operator to be at specimen level during testing, having to catch the rebounding weight does not seem to be too difficult a task.

Having to catch the rebounding weight by hand does, however, become more difficult at the heavier drop weights. With the heavier weights and higher drop heights, isotropic materials like aluminum will tend to deform mostly plastically. This causes problems because the falling weight does not bounce off the specimen very far. This really would not be a problem with composites at these higher energy levels, because the specimens would undergo brittle fracture with delaminations and fiber breakage occurring. In other words, the impact tup would penetrate the specimen as opposed to bouncing off. There would be a certain range at the lower energy levels where the impact head would only slightly bounce off the specimen. In this case, the rebound catch would be a worthwhile
addition. Since most of the initial work that will be performed with this tester will be low energy, low force events, catching the rebounding weight by hand will be acceptable for now.

One simple solution to catching the rebounding weight was to tie a small cord to the falling weight. Then, this cord was run up over the top cross bar support. The operator then holds one end of this chord in his or her hand with just enough slack to allow the specimen to impact the specimen. Then, the chord is pulled taught when the weight rebounds, thus preventing the weight from hitting the specimen again. This turns out to be a very effective and simple way to catch the weight, although it does require a couple of drops to get the timing down. The main benefit of this cord is that it does not require the operator to actually physically grab the rebounding weight, so there is no chance that it could be missed on the rebound. Also, there is no physical contact with the rebounding crosshead, so there is no possibility that a corner or sharp edge could be grabbed. This could potentially cause minor injuries. There are many other more complex, automated rebound catch ideas that could be pursued, but for now this cord seems to work very effectively.

Now, back to the quick release. The mechanical drawing of the quick release support is shown in Figure 11. This part was made by the Mechanical Engineering Machine Shop. By looking at the drawing, two 0.165" diameter holes drilled 1" apart can be seen. These were the dimensions required to mount the EMR 622 Electromagnet that was purchased from Jobmaster Corporation. This magnet has an attractive force of 100 pounds. This was believed to be sufficient, since the falling weight would, at its maximum, only be around 12-15 pounds. This magnet cost $52.50. Other features of the quick release
Figure 11: Mechanical Drawing of the Quick Release Support.
support include the two holes that are 5" apart. These are where the guide rail shafts go. The quick release support is then tightened to the shafts at the desired location using the slits at each end. By tightening the 3/8" bolts which span the slits, the diameter of the shaft holes is reduced. This secures the quick release support to the shafts. The quick release is then ready to be installed on the rails.

The next item that had to be built was something to power the magnet and turn it off at the appropriate time to release the falling weight. The magnet requires 12V DC and 5 watts of power. A battery could have been used to power this electromagnet, but it was decided that an AC to DC converter would be better. This way, an electrical cord could just be plugged into the wall instead of having to rely on batteries. If batteries had been used, they could have gone dead at any time. This would have caused the weight to be dropped unexpectedly, which would have been a serious safety risk.

For the converter, a 12.6 volt AC, 3-amp power transformer was used. This reduced the voltage from 110V AC to 12.6V AC. The output from this was fed into a full wave rectifier to give approximately 12.6V DC. A 2000 μF capacitor was connected in parallel with the output from the rectifier to smooth any ripples caused by the rectifying. The negative wire was then connected to one lead on the electromagnet and the positive lead passed through three switches before connecting to the other lead of the magnet.

The three switches were included not only to release the falling weight, but to provide some safety measures as well. The first switch is a two position toggle switch. This turns the main power on and off. The second switch was connected in series with the power on side of the first switch and is also a two position toggle switch. This one can be switched
to either a "lock" mode for when the operator is preparing the test apparatus or to a "test" mode for when testing is ready to commence. The third switch is a normally closed push button switch connected in series with the test mode side of the second switch. Should this button be accidentally hit while the second switch is in the "lock" position, the falling weight would not be unexpectedly released. This is because the wire from the "lock" mode side connects directly to the electromagnet to keep it on full time, while the wire from the "test" mode side goes through the push button switch before connecting to the magnet. In other words, if the first switch is on and the second switch is in the "test" position, the electromagnet is turned off when the push button is depressed. Conversely, when the second switch is in the "lock" position and the push button is depressed, nothing will happen and the electromagnet will remain on. The transformer, rectifier, and capacitor were all put in an aluminum enclosure box, and the switches were mounted on one side. This box was then mounted on the left end of the X-Y table for easy access during testing. By locating it near the impact point, the falling weight can be easily caught when it rebounds off the specimen. Finally, the switches on the box were labeled for their respective purposes. The quick release apparatus was now complete.

3.5 Design of the Falling Head

Up to this point, no real mechanical drawings had been needed because all of the work had been done by the author or the machine shop using sketches. One exception was the UMD designed quick release support. For this, the required changes were just sketched on the existing drawing for fabrication and formal drawings were made later. The fabrication of the falling head would again require the help of the Mechanical Engineering Machine Shop. These parts could have been done by the author, but for time savings reasons the shop was used. Another concern was the fact that tight tolerances were
involved and the work needed to be right the first time. The use of mechanical drawings also helped to visualize the layout of the various components of the design by retaining dimensional accuracy. Therefore, it was necessary to become reacquainted with computerized drawing programs. Here at Virginia Tech, a computerized drawing program called Cadam is used. After about a day of learning the program, the drawings were ready to be made.

For the UMD design, only one 7.2" x 2" x 2" piece of solid aluminum bar was used (Vernekar et al, 1993). It was not felt that this would give an acceptable range of impact energies. In fact, their tester had an impact energy range only up to 18 Joules (or 13.3 ft-lb). It should be noted here what is meant by impact energy. It is simply the total kinetic energy that the impactor has immediately before impact (or 1/2mv²). This maximum energy level could still inflict significant damage, but some design changes were thought up in an effort to increase the versatility of this machine. The main reason for these enhancements was to be able to pursue thorough testing on many different materials. For example, composites behave differently under low or high velocities or low or high mass impacts, even though the impact energy levels may be the same. Because of this, an attempt was made to have the best of both worlds. Taking design ideas from both the UMD design and the Dynatup 8200 tester in the ESM department at VPI, a flexible design capable of both very low and relatively high weight impacts was devised.

The design consists of a three part falling head. Tests can be performed with either a single falling mass or a frame like structure which weighs more. Additionally, when the frame configuration is used, additional weights can be added to further increase the weight of the impactor. A sketch of the complete two head "frame" configuration can be seen in
Figure 12. Furthermore, a schematic of the single head configuration can be seen in Figure 13. Then, the mechanical drawings of the top and bottom impact head portions can be seen in Figures 14 and 15, respectively.

Next, the features shown in Figure 14 will be described. To begin, the crosshead needed to be as light as possible in order to perform low weight impacts. For this reason, a 2" x 1" piece of aluminum was used. The bearings that were ordered could not be press fit according to the manufacturer, so retainer rings to hold them in place. These were ordered from Thompson to fit the Super 12 bearings. A retainer ring is just an expandable circular ring with a slot in it. These rings are stretched using a special tool and fit in retainer ring slots in the bearings. There are two of these slots in each bearing, one at each end. For the Super 12 bearing, these slots were 1.16" apart. Therefore, to hold the bearings in place, a weld bead was built up on the outside of the crosshead around where the bearings were going to be located. Because the beads were on the outside edge of the frame, they would not interfere in any way with the add-on weights which go on the inside of the frame. The weld beads were then machined so the total thickness of the part was 1.16" in the bearing regions.

An additional design feature of the crosshead includes the two holes that were made to hold the bearings. These holes are 5" apart at their centerlines. The tolerance on this dimension had to be very small in order to ensure that the holes on the top and bottom crossheads would line up, thus aiding alignment. These holes were first drilled, then reamed to produce a smooth interior finish for a good bearing fit.
Figure 12: Two Head "Frame" Configuration of the Falling Weight.

Figure 13: Single Head Configuration of the Falling Weight.
Figure 14: Mechanical Drawing of the Top Crosshead.
Figure 15: Mechanical Drawing of the Bottom Crosshead.
Other important features of the top crosshead include the two holes on one side and the single hole on the opposite surface towards the center. The two holes were put there to hold the mechanical latch catch that was originally going to be used. Instead, an electromagnet was used, as was discussed in the previous section. The single hole is where the tup cylinder is attached when only one crosshead is used.

Analyzing Figure 15 shows the same built up and machined weld bead and tup cylinder screw in location. This location is used when the frame configuration with both crossheads is used. When in this configuration, two 1/4" x 2" x 7 5/8" steel end pieces are bolted onto the crossheads using the 1/4" tapped holes at the ends of both crossheads. Since lower weight impacts are performed using just one crosshead, steel was used for these end pieces to provide a little more weight and rigidity when using the frame configuration. Additionally, the two 1/4" tapped holes located 2" apart on opposite sides of the center are for the threaded rods used to hold the add on weights. Finally, both the top and bottom crossheads were sandblasted for aesthetic purposes. This gave them a textured appearance and also provided a slightly better grip when handling the falling heads.

The add-on weights were to be made two pounds each. Using the designed capacity of four weights, this adds up to eight pounds of additional weight. For this design, steel was chosen for the material. If additional weight is desired in the future, these steel weights could either be replaced with lead ones, or more weights could be added by simply changing the length of the steel end pieces and making longer threaded rods to hold the weights. To choose the initial length of the steel end pieces, the height of each weight had to first be determined. For this, the length of the weights was picked to be 3 1/2".
This was a dimensional limitation due to the position of the bearings. In fact, if the
distance between the rails had been increased, more weight could have been put in a
smaller vertical distance because each weight could have been longer. Anyway, by
knowing the density of steel (approximately 0.28 lb/in$^3$), the mounting hole size (1/4"),
and the length of the weights (3.5"), the theoretical height of each weight could be
determined. This worked out to 1.035".

The weights were produced using these dimensions. First, pieces of steel slightly
longer than 3.5" and slightly taller than 1.035" were cut from a 2" thick plate of steel.
Then, the surfaces were machined to the proper thickness using an end mill. Opposite
sides were machined in such a way that they were perfectly parallel to each other. This
insured that the weights would sit flush with each other when they were stacked on top of
one another. With this being the case, the they would not vibrate or shift much during
testing. The weights were then stamped with their exact weights. The four weights
turned out to weigh 2.012 lb, 2.015 lb, 2.016 lb, and 2.017 lb. As can be seen, these are
all close to the designed weight. The weights slide over the 1/4" threaded rods and are
secured in place by tightening a wing nut on each rod.

Two final parts had to be added to complete the falling weight. First, since an
electromagnet was going to be used and the crossheads were aluminum, something had to
be added to the top of the upper crosshead for the magnet to be attracted to. The main
concern was residual magnetism once the power to the electromagnet was cut. The
magnet manufacturer recommended using cold rolled steel. They indicated that cast iron
could be used with the least residual magnetism; however, the attractive strength of the
magnet would decrease by about 30%. With steel, the pull would be the greatest possible,
but there would be some residual magnetism. If this became a problem, something like a piece of cloth, paper, or Teflon fabric could be put between the magnet and the steel. Another option would be to reverse the polarity of the magnet when the release button was pressed. This would alleviate the problem of residual magnetism. The decision was made to use a small piece of steel plate attached to the falling weight. Then, a piece of Teflon cloth would be put over the electromagnet to help alleviate the problem of residual magnetism. This was the least complex solution to implement. The holes for the quick release latch had already been drilled slightly off center, so these holes were just used instead of drilling new ones. A 1 3/4" x 2 1/4" x 1/4" piece of steel was cut and two holes were drilled in the proper place to allow the piece to be centered on the top crosshead. The top surface of this piece of steel had to be flat, so the holes were countersunk and flat head screws were used to attach it to the crosshead. This solution worked very well, and completed the falling weight fabrication with the exception of the force transducer attachment.

3.6 Choice of a Force Transducer Tup

The tup is the part of the falling weight that actually impacts the specimen. It has a force transducer incorporated in it. This provides the force versus time data which must be obtained as part of the instrumentation of the impact tester. The initial instrumented impact testers used strain gages attached to a cylinder to indirectly measure the force. The Dynatup 8200 uses this technology. However, with the increasingly more widespread use of piezoelectric materials, high impedance piezoelectric force transducers are now available. These are much simpler to use because they do not require calibration before each use. They are also completely self-contained with built in amplifiers. For these reasons, it was decided to use a piezoelectric force transducer.
CIMSS has many force transducers made by PCB Piezoelectronics, Inc. This company was contacted regarding force transducers with 1/2" hemispherical impact heads. The 1/2" head is a standard used with many ASTM tests. It is also the same size used on the Dynatup 8200. A representative from the company mentioned that their standard model 208 force transducer was available with a screw in impact cap. The problem with this design was that the instrumentation wire protruded from the side of the transducer. On through specimen impacts where the tup breaks through the material, this connection and wire could potentially be damaged. The representative then mentioned that PCB also manufactures a force transducer with an integrated impact head. In addition, the connecting wire comes out of the top of the force transducer, so it will not get in the way during through impacts. The company actually makes this transducer for a manufacturer of instrumented impact testers. This force transducer had a non-standard metric thread, which caused some problems. The design of the part was also somewhat questionable. More about this in a moment.

The next thing that needed to be decided was the load range of the force transducer. There is no way to truly calculate a priori the amount of force that will occur during all impacts. This depends in the plate stiffness, the boundary conditions, the impactor mass, the impact velocity, and where on the plate the impact occurs. For example, if a plate is impacted with a certain impact energy at its center, the maximum force will be less than if the same plate is impacted with the same impact energy near its edge. This is because the deflection will be less near the edge. The force is related to this deflection along with the velocity. The higher the velocity, the more inertia the falling weight will have. Therefore, the acceleration will be much higher and Newton's Law relates force to mass and acceleration. Furthermore, the deflection is difficult to predict because the impact is an
impulse force and the plate is a three dimensional specimen, with thickness also playing an important role in the transverse deflection. The best way to determine the force range is just to look at previous experimental data and make an educated guess of the range for the specific application.

The specimen plate was to have a clamped-clamped boundary condition with dimensions of around 12" x 24". Since this boundary condition is relatively large, it was difficult to find experimental data in the literature. One other factor to consider was that the maximum impact velocity for this tester is around 15 ft/s. One reference listed experimental results with eight ply composite plates, with only the two opposite edges clamped. The impactor weighed 3.4 lb and the plate had a non-supported area of 3.5" x 9.9". The literature indicated is that a maximum force of around 500 lb for a 16.4 ft/s impact occurred (Lagace and Wolf, 1993). Since the boundary conditions for this tester would be clamped-clamped, the deflection would decrease causing the force to increase. However, the boundary condition would be larger, so the deflections would be higher causing the force levels to go down. It was thought that these two factors would, for the most part, offset one another. Therefore, it was felt that a 1000 lb load range would be acceptable. Of course this was just speculation. The actual load range could be higher or lower depending on many factors, such as specimen type, specimen thickness, impactor mass, etc.

PCB sells the pre-described force transducer, the model 208M30, with load ranges of 0-500 lb, 0-1000 lb, and 0-5000 lb. The only difference between these are their sensitivities. These force transducers operate in a linear range up to 5 volts. The maximum voltage at which data acquisition systems can gather data is 10 volts. Since
1000 lb was decided on as the maximum force, the 208M30 force transducer with the 0-1000 lb range was purchased. This transducer has a manufacturer rated sensitivity of 5 mV/lb, or 5 volts for 1000 pounds of force. The calibration curve for the actual transducer that was purchased indicates a sensitivity of 5.30 mV/lb. In actuality, this transducer can be operated up to 1887 lb, which would be the 10 volts maximum that data acquisition systems can handle; however, above 5 volts, the force transducer will begin to lose some of its linearity. So, even if the initial load range assumption was way off or different specimen types or geometries are tested in the future, there would be some flexibility to handle larger load ranges if needed. Furthermore, the maximum load that this transducer can endure without actually breaking is 5000 lb. Obviously, though, it cannot gather data this high. The transducer cost $595. A drawing of it can be seen in Figure 17.

3.7 Design of the Tup Cylinder

Once the force transducer had been chosen and purchased, the next step was to design and make something to connect it to the falling weight. This connection also serves as a way to get the force transducer at the correct height to impact the specimen plate, since the bottom of the rails were more than 6" above the specimen plate. The drawing of the force transducer was shown in Figure 17. This provided the dimensional parameters around which the tup cylinder had to be designed.

The first tup cylinder that was made actual bent during the commissioning of the tester. This failure occurred when all the weights were added to the falling weight and the weight was dropped from the maximum possible height. This was the maximum impact energy that could be achieved from this tester. It was good that this happened at the time
Figure 17: Manufacturer's Drawing of Model 208M30 Force Transducer.
that it did, instead of later on down the road. This first design will be described and shown. Then, the modified design will be discussed.

The exterior of the force transducer had a diameter of 1/2". So in order to have a smooth transition from the cylinder to the force transducer, the outside diameter (OD) of the cylinder at the force transducer end should be 1/2" as well. The purpose of having a smooth transition is so nothing will get caught up should the tup break through the specimen.

The next design consideration was how to connect the force transducer to the tup cylinder. It was decided that the best way would be to first attach the 10-32 coaxial cable connector to the force transducer, then screw the transducer into the end of the cylinder. The problem with this was that the OD of the coaxial connector is larger than the ID of the threaded hole that was required to attach the transducer to the cylinder. In other words, the coaxial connector would not fit through a M7 x .75 threaded hole. This design did not seem to make much sense. It also caused the tup cylinder design to be more complex than it needed to be.

For the first design, a hollow pipe with an ID of 0.35" was used. The OD of this pipe was turned down on a lathe to 1/2" along its entire length. The transducer wire was then run up inside this pipe. Then, two end caps were used to attach both the transducer to the cylinder and the cylinder to the falling weight. These end caps were removable, so the cylinder could be assembled correctly. The complete cylinder that was initially built is shown in Figure 18 with the end cap details. As can be seen, the OD's of the end caps were 0.35" so they would fit snugly inside the pipe. The OD of the coaxial connector is
Drill and Tap for M7 x .75 Force Transducer

Drill For 6-32 Set Screw

Detail of Force Transducer End Cap

Drill for 6-32 Set Screw Approx. .1" Deep

.75"
1.1"

Detail of Attachment End Cap

Drill and Tap for 1/4-20

Drill and Tap for 6-32 Allen Set Screws

Note: These parts are all circular in shape

Figure 18: Initial Tip Cylinder that Bent Under High Load.
0.28", so it easily fit inside the pipe. The bottom end cap was 0.48" long. This was long enough for the threads on the force transducer along with a little extra to anchor the end caps to the cylinder with the set screws. The cap was drilled and tapped in the exact center for the M7 x .75 threads that were on the force transducer. The 0.75 pitch is non-standard, so expensive special taps had to be ordered. The purpose of the top end cap was to connect the cylinder to the falling weight. On the falling weight, a 1/4" hole had already been drilled and tapped. A 1 1/8" piece of 1/4"-20 threaded rod was used for this connection. The 1.1" long end cap was made with a 3/4" deep hole that was drilled and tapped for the 1/4"-20 threaded rod.

Then, anchor holes were drilled to secure the end caps. Two holes at 90° to each other were made for each end cap. These holes were drilled and tapped through the tup cylinder and into the end caps for 6-32 Allen cap screws. The purpose of these screws was to hold the end caps tightly in place. Finally, a 0.11" diameter hole was drilled through the wall just below the bottom of the top end cap. This is where the coaxial wire exits the tup cylinder. Recall that the wire runs up inside the cylinder after attaching to the end of the force transducer. This hole was just big enough for the 0.07" diameter coaxial wire to fit through. One end of the wire already had the coaxial connector attached and the other end just had a bare wire. It was ordered this way to allow for the wire to exit the tup cylinder with the smallest hole possible. Once the wire was through the hole, a coaxial connector was soldered onto the loose wires in order to allow the wire to be connected to other electronic equipment for data acquisition.

It was necessary to go back to the drawing board to redesign the tup cylinder once the first one failed. It was first necessary to determine why the first one failed. The initial
cylinder had a buckling limit calculated at 2200 lb; however, the structural weaknesses introduced by drilling the holes through the tube walls for the set screws and wire were not accounted for. Also, buckling calculations are for the ideal case. There are always material flaws which will lower this limit. A visual analysis of the tube indicated that the tube bent the most right at the set screws at the force transducer end, and slightly at the holes at the other end. These locations were clearly the weak spots. One other item that needed improvement was the way the cylinder was tightened to the falling weight. Since the tube was round, there was no really good way to perform this operation sufficiently without scratching the outer surface. Finally, as was noted earlier, the specimen plate actually deflected to the point where it hit the solid steel base plate. This clearly caused much higher loads than would have been experienced if the specimen plate had not hit something of such a high stiffness. This was not realized until after a new tup tube had already been designed. If a new tup cylinder had not been built, there would always have been a risk of failure because of the inherent weaknesses. With a new design, this part should last for the life of the machine.

The new design consists of just two parts which can be seen in Figures 19 and 20. The first thing that can be seen is the elimination of the end caps. By using two main cylinder parts instead of a single tube with end caps, the structural integrity is increased. These two parts are connected by a 9/16"-12 threaded section that was 1/2" deep. This was an arbitrary distance that was believed would sufficiently hold the two parts together. The length of the threaded portion on the top part is 0.49" while the threaded hole in the bottom section is 0.5" deep. This way, there is a little clearance to allow the two sections to tighten together securely with a smooth outside surface. This is important because the top of the bottom section needs to butt up with the flat portion of the top part to help
Drill & Bottom Tap for $\frac{9}{16}$"-12

Drill and Tap for M7x.75

Note: This Part is Circular in Shape and is to be Made of Steel Bar Stock.

Figure 19: Bottom Section of the Redesigned Tap Cylinder.
Note: This Part is Circular in Shape and is to be Made of Steel Bar Stock.

Front View

Side View

Figure 20: Top Section of the Redesigned Tap Cylinder.
transmit the load from the impact. If it does not butt up, the threads would carry all the load, which could potentially cause damage to the threads. Further main design modifications include increasing the OD to 1" to increase the cylinder's buckling load capacity. Also, the part was made out of solid round bar instead of the tubing used for the first cylinder. The important factor when trying to increase the buckling load capacity is to increase the cross sectional moment of inertia. This is done by moving material farther away from the neutral axis. Increasing the diameter to 1" does this. This obviously increased the weight, but it was necessary to make the tester suitable for its whole range of impact levels. An added benefit of having a greater axial stiffness is that it reduces the strain energy absorbed by the cylinder during the impact. This will be discussed in Chapter 5.

The total length of this tup cylinder was chosen based on design limitations of the tester. The specimen clamp had already been fabricated when this redesign was done, so the distance from the bottom of the falling weight to the top of the specimen plate could be determined. This was done with the falling weight sitting on top of two 1/2" rubber stops that rested on top of the shaft support blocks. These stops were put there to catch and cushion the falling weight should the tup ever break through the specimens instead of bounce off. The bottom of the specimen plate was 3/4" above the top of the solid steel specimen clamp base. To prevent damage to the force transducer, the tup should never hit this solid steel plate, even on through specimen impacts. So the tup should be just above this steel plate when the falling weight was resting on the rubber stops. By knowing the distance from the falling weight to the specimen plate (8 5/8"), adding on 3/4" (the distance from the specimen to the steel base), subtracting 1/32" (an arbitrary clearance so the tup does not actually hit the base plate), and subtracting the length of the portion of
the force transducer that would be exposed when it was screwed into the cylinder (0.8"), a total length of 8.544" was obtained.

First, the design of the bottom section shown in Figure 19 will be described. This is the part that holds the 208M30 force transducer. The OD of the transducer was 1/2"; so, to make a smooth transition to the tup cylinder, the 1" diameter round rod was tapered down to 1/2" at a 30° angle. This way if the tup should break through a specimen and penetrate greater than the 0.8" length of the exposed portion of the force transducer, the cylinder will not have any blunt edges that could further damage the specimen. The threads on this transducer are M7 x .75, so the hole with these threads was tapped in the center of the tapered end. The force transducer then screws into this hole. When it is screwed all the way in, the coaxial connection protrudes from the other side of the hole into the 9/16" threaded hole cut in the other end. The 10-32 coaxial connector can then be tightened to the force transducer by reaching down into this 9/16" hole with a small pair of needle nose pliers. The only remaining design feature of this bottom section is the two flat spots on opposite sides of one another cut just above the tapered section. These are just big enough to get a wrench on to allow this bottom section to be tightened sufficiently to the top section. As indicated earlier, it is difficult to put a lot of torque on a round object without grabbing it with something like vise grips or pliers. These would scratch the surface. By cutting these shallow flat spots, a wrench can be used to tighten the two parts together without the cylinder really losing any structural strength.

The top section of the tup cylinder was shown in Figure 20. At the top of this part is where the cylinder is attached to the bottom of the falling weight. This is done using the same 1/4"-20 threaded rod that was used in the initial cylinder, but this time the hole is
drilled and tapped directly into the cylinder instead of an end cap. Also, another pair of flat spots can be seen near the top of this section. This is used to help tighten the tup cylinder to the falling weight. It also is used to keep the top section from turning when the bottom section is tightened to it.

The only remaining features of this top section are the two holes drilled in it. The 5.843" long, 9/32" diameter hole is where the instrumentation wire runs up the inside the cylinder. This diameter was chosen because it would be just big enough for the 10-32 coaxial connector to fit in. Then, the wire exits the cylinder through the 0.11" diameter hole drilled at a 60° angle. Because it was drilled at an angle, the wire can be pushed up through the larger hole and out the smaller one much easier. Stringing this wire inside the cylinder is further aided by the fact that the smaller hole intersects the larger hole right at the very top of the larger hole. In other words, everything is sloping towards the angled smaller hole. A shallower angle would have been better, but the shallower the angle, the larger the hole on the surface. This circular hole becomes an elongated ellipse when it is drilled at an angle. As was seen with the previous design, every hole through the walls of the cylinder are points of weaknesses. So, the angle of the small hole was limited to 60° to keep its profile on the outside surface to a minimum. This completes the description of the tup cylinder.

3.8 Alignment of the Guide Rails

The falling weight with all its components were assembled, as were the quick release and electromagnet. Next, it was necessary to make sure the rails were aligned and that they were perfectly vertical. Recall that the alignment problem that UMD had was solved somewhat by using self-aligning bearings. The alignment procedure that was followed
was relatively simple. First, the quick release was put on the rails. Recall that the holes on this part are exactly 5" apart. Therefore, the alignment could be done by just moving the quick release to the top of the rails and tightening the bolts that hold the shaft support blocks to the cross bar. Then, the same procedure had to be done with the quick release at the bottom of the rails. It took a couple of tries to obtain proper alignment because the rails not only had to be the correct distance apart, they had to be perfectly vertical as well to allow for a truer free fall. To ensure that the rails were level, the machine itself first had to be level. This was done by putting a level on the X-Y table and using a wedge to raise the appropriate side or sides of the machine. Then, a level was placed on the sides of the rails to make sure they were level too. Having done this, the rails were exactly perpendicular to the X-Y table and perfectly vertical. From this point on, the shaft support blocks should not be moved. If the shafts need to be removed for some reason, such as to change from the single falling weight configuration to the frame configuration, all that is required is to loosen the Allen bolts that tighten the shaft support blocks around the rails. Then simply slide the rails upward until they are high enough above the bottom shaft support blocks to perform the desired work.

3.9 Final Mechanical Components

Now that the rails had been mounted and the falling weight and tup had been completed, the only remaining mechanical work remaining required for the tester was to mount the velocity measurement device and the height measurement ruler. These two components were added last because the rest of the structure needed to be assembled in order to determine where to locate them.
3.9.1 Installation of the Velocity Measurement Device

The purpose of the velocity measurement device is to determine the velocity of the crosshead just before impact. This is a required parameter for determining the energy absorbed by the specimen. For this, an Optoschmitt model HOA2001 infrared emitter/detector sold by the Honeywell Micro Switch Division was used. This was the same electronic device used by UMD. It consists of an infrared emitting light source on one side, and a sensor on the other side. Motion is sensed by breaking the radiation path between the two discrete devices (Honeywell Catalog, 1993). Some sort of object with a known width was required to break the signal for a certain time duration when it passed through the emitter/detector. By measuring this duration and because the object's width would be known, the velocity can be obtained. The specifications of the HOA2001 emitter/detector can be seen in Figure 21.

The first order of business was to make a device to hold the emitter/detector. Since the specimens could be tested at different heights or be of varying thicknesses, it was essential that the velocity detector be at the correct height. This occluding object should pass through the detector immediately before the specimen is impacted. To hold the emitter/detector, a 5" long by 1/16" thick piece of steel was used. The important dimensions required were the two mounting holes on the emitter/detector and the position of the two sets of leads. These respective dimensions can be seen in Figure 21. Three 1" long slots were milled in the steel, two were 1/8" diameter slots that correspond to the mounting holes and one was a 1/2" slot that allows for the two sets of leads to move.

The next step was to mount the sensor on the tester. An appropriate distance away from the crosshead was chosen. This turned out to be approximately 1 1/2". Then, all
HOA2001 — Optoschmitt Assembly

ABSOLUTE MAXIMUM RATINGS
(25°C Free-Air Temperature unless otherwise noted)
Storage Temperature .................. -40°C to 100°C
Operating Temperature ................. -40°C to 100°C
Lead Soldering Temperature (9 sec) ...... 240°C

INPUT DIODE
Forward DC Current .................. 50mA
Reverse DC Voltage .................. 3V
Power Dissipation .................. 100mW(1)

OUTPUT SENSOR
Maximum allowable Vcc .................. 20V
Output sink (–40°C to +100°C) ........ 18mA

Notes:
1. Derate lineally 1.25mW/°C above 25°C.
2. See Data Sheet for 83P 0000 for more complete specifications on Schmitt detector.

OUTLINE DIMENSIONS
Tolerance 3 plf decimals ± 0.010 (0.25)
2 place decimals ± 0.003 (0.075)
Unless specified
ALL DIMENSIONS IN INCHES (MILLIMETERS)

TYPICAL PERFORMANCE CURVES

Figure 21: Specifications of the Optoschmitt HOA2001 Infrared Emitter/Detector.
that had to be done was to secure the steel piece that holds the emitter/detector vertically at this location. For this, a 1" x 1 1/4" x 1 1/2" rectangular block of steel was used as a spacer from the lower cross support bar. This block was machined to this size and two 3/16" holes were drilled through it. Then, two holes were simply drilled in the cross support bar at about the center of the crossheads, and the velocity measurement device was mounted.

The next step was to hook up the electronics to the emitter/detector. It has five input wires. The correct way to hook up the emitter/detector was determined from looking at the specifications shown in Figure 21. Two AC to DC converters were purchased from Radio Shack. These are the same converters that are used to power things like little hand held games. One converter produced 1.5V DC, and the two wires from this were soldered onto the infrared emitter terminals. The other converter provided an output of 6V DC. The positive wire from this converter was soldered onto the $V_{ce}$ lead, and the negative wire was connected to the ground. Finally, a two wire coaxial cable was used to provide the output from the detector. One wire was soldered to the output pin, $V_o$, and the other wire was just soldered to the cable's shielding. This shielding was also connected to the ground. All these wires connected to the ground were not all soldered to the little pin. The cable shielding was soldered to the pin, and the wire from the voltage supply and the other wire in the coaxial cable were both soldered to the shielding. The second wire in the coaxial cable was really not needed, but it was connected to the ground because it would be easier to attach to the screw terminal panel that interfaces with the computer than the larger shielding wire. The output signal from this emitter/detector was checked on an oscilloscope, which indicated that it did indeed work correctly. The output voltage is approximately 5 V when the signal is not occluded, and 0 V when it is.
The final procedure that had to be done was to come up with something to use to occlude the emitter/detector signal, and a way to attach this object to the crosshead. UMD used a transparency sheet marked with a blackened in rectangle of a certain width. This was the first thing that was tried. A device to hold the transparency card was first built using a 3/4” tall piece of aluminum T-channel. Two size 6 holes were drilled in the T-section to attach it to the crosshead, and a size 6 slot was cut into the web section for two small Allen bolts which would hold the transparency card. Then, holes were drilled and tapped 1/2” apart at the center of the 7 1/4” side of both crossheads for 6-32 bolts which would hold the t-section to the crosshead. Since holes were made in both crossheads, the velocity card holder could be attached to either one depending on what falling weight configuration is used. As it turned out, this transparency card did not work real well. The problem was that the card would not remain straight. Instead, it would bow and warp. It was decided that this was unacceptable, and that something else needed to be done.

The Dynatup 8200 uses a small metal strip to occlude the signal. Therefore, this was the next route to pursue. A 3.5” x 0.5” strip of 0.03” thick aluminum was cut. This strip was first bent in half, and a hammer was used to completely flatten the two halves over on each other. Then, since the emitter/detector had already been mounted, the distance from it to the edge of the falling weight could be found. This was about 1.5”. The ends of the strip were bent 90° to form an elongated T-shaped piece. Holes were drilled 1/2” apart in each of the bent up pieces to match the 6-32 holes already drilled and tapped in the crossheads for the previous velocity card holder. The two edges of the piece were machined flat and parallel to each other. This gave the flag a uniform thickness
throughout the part. This thickness was then measured with a set of calipers. For the particular piece that was used, this distance turned out to be 0.5850". This width can really be any value within reason. It is just input into the data acquisition system to determine the velocity. This redesigned piece works very well. Now, the velocity detection device was complete, and the emitter/detector could be fully adjusted to the proper height. If new velocity pieces are needed in the future, they can be made in about twenty minutes using only a power shearing machine, a press break to bend the material, a milling machine with a small end mill, and a drill press to drill the two holes. After the velocity card is installed, the emitter/detector needs to be aligned so that the flag passes through its slot. To do this, the emitter/detector holder can be loosened, rotated slightly, and tightened back down to line the emitter/detector up with the attached velocity card.

3.9.2 Installation of the Height Measurement Device

The final assembly piece required was a measuring device that could be used to determine the drop height. This is important because the drop height needs to be known in order to get some sort of idea as to what the impact energy will be. The impact energy is just the kinetic energy at impact. This should be equal to the potential energy at the drop height if there are minimal friction losses. So by knowing the drop height, the crossthead energy at the moment of impact can be estimated. The Dynatup 8200 tester does not use any sort of height measurement device, but other Dynatup testers do. It was felt that having this device would be a good feature to add.

Obviously, to measure the height one needs to use some sort of ruler. The distance between the two cross support bars was approximately 48" from the bottom of the lower one to the top of the upper one. A 48" long aluminum carpenter's straight edge was
purchased from a hardware store for use as a large ruler. It had graduations down to 1/16". Because of the way the rails were mounted, the crosshead never reaches the top of the lower cross support bar; so, it was obvious that the ruler could not simply be mounted by just drilling holes in the cross bars.

The first step was to mill 3/16" slots in the ruler so it could be adjusted up and down. This would allow the ruler to be adjusted for different specimen heights. Then, a 5" x 2" x 3/8" piece of steel was machined to act as a lower mount. This piece had a drilled and tapped 3/16" hole 1/2" from the top. It also had two 3/16" drilled holes which were 3/4" apart. These holes were 1/4" and 1" away from the bottom, respectively. The hole that is 1/2" from the top is where the bolt that secures the ruler was screwed into. The bottom two holes are used to attach the steel piece to the lower cross bar. This purpose of this steel piece was to raise the bottom (zero point) of the ruler to the height where the bottom edge of the lower crosshead is located when the tup impacts the specimen. Finally, a 3/16" hole was drilled and tapped in the top cross support bar to hold the top of the ruler.

The procedure for lining up the ruler during testing includes first installing the specimen in the specimen clamp. This will be described in Chapter 4. Then, the crosshead is gently lowered down, and the tup is rested on the specimen. Next, the two 3/16" bolts that hold the ruler are loosened, and the bottom of the ruler is lined up with the bottom of the lowest crosshead. This is the zero height. Finally, the two 3/16" bolts are tightened, thus securing the ruler in place. The falling weight is raised and locked to the electromagnet. The whole falling weight and quick release are then moved to the desired height. This height should be measured from the same edge that was used to set the zero height. Then, the quick release is tightened down and testing is ready to commence. The
complete layout of the velocity device and the ruler can best be seen by referring back to
the drawings in Figures 6 and 7.

3.10 Space Considerations and Final Assembly

As mentioned in Chapter 2, it was important to find a place to put this tester once it
was nearing completion. Since it could not be moved to the upstairs lab at CIMSS
because it weighed too much, the best solution was to obtain space in Randolph Hall.
This is the Mechanical Engineering Building at VPI. A letter was drafted to the
department head and space was obtained in a room in the basement. This space was fine
during the final stages of construction, but a computer was going to be used to fully
instrument this tester. The problem with this area was that it was in an open area where
many people had access. Because of security concerns, the computer would have had to
have been disconnected and moved to a locked room after every use. The floor also had a
rough grate which the computer cart would have to be rolled over. This would have
subjected the computer to potentially harmful vibrations. Because of these reasons,
another location was sought. Another memo was drafted, and this time space was
obtained in a locked cage. This is where the completed tester was put.

This completes the description of the mechanical portion of the tester. The falling
weight in the single crosshead configuration weighs 3.518 lb, while it weighs 7.348 lb
when it is in the frame configuration. Pictures of the tester up to this point can be seen in
Figures 22, 23, and 24. The first two figures show the front and side views of the tester,
while Figure 24 shows a close-up of the falling weight, tup, velocity device and ruler.
Figure 22: Front View of the Completed Impact Tester.
Figure 23: Side View of the Completed Impact Tester.
Figure 24: Close-up View of the Falling Weight, Tup, Velocity Device, and Ruler.
Chapter 4

Design and Fabrication of the Specimen Clamp

4.1 Design Considerations

Now that the main portion of the impact tester itself had been completed, a clamp to hold the test specimens had to be designed and built. Recall that the main purpose of building an impact tester from scratch was to be able to test multiple locations of specimens which have large boundary conditions. So, obviously, the specimen clamp had to be relatively large. At the same time, it had to be rigid in order to securely clamp the specimen. One further design consideration was the type of stresses that the plate should undergo. According to Driscoll (1985), shear effects become important if the unsupported span to thickness ratio of the plate is less than ten. If this ratio is greater than sixteen, the tensile stresses are dominant. Clearly with a large boundary condition and relatively thin plates, the tensile stresses are going to be the manner in which the impact load is absorbed.
Initially, the size of the boundary condition was supposed to be around 18" x 26". During the design and fabrication stages of the impact tester, this boundary size was changed. It was decided to use a boundary condition of 12" x 24". There are a couple of reasons for this change. First, the table only moved approximately 7 5/8" in the front to back direction; therefore, it did not make much sense to use a boundary condition much larger than this. The 12" unsupported length leaves slightly over 2" of the specimen remaining between the boundary in this direction and the impact location at the table's maximum adjustment. Since the plate response will be much different when it is impacted near the edge as opposed to near the center, the signals received by the sensors that were going to be used for impact detection would be very different as well. This signal distinction was the main reason for the change in the boundary condition size in the front to back direction. The other reason for changing this size was the dimensional limitations imposed by the construction of the tester itself. If the boundary condition was made any larger in this direction, the side mounted toggle clamps would hit the I-beams when the X-Y table was in the position closest to them. Finally, since the table has a mobility of 24" in the left to right direction, the size of the boundary condition was not a real concern in this direction. A length of 24" was chosen for this, because it would allow testing to occur along the full length of the boundary condition. It did not make much practical sense to make it any longer. Also, it made the length of the unsupported region in the left to right direction exactly twice that of the front to back direction.

The next design consideration was to decide on what type of boundary condition to use. The simplest and most repeatable type to make for impact testing some sort of a clamped boundary. This could be either clamped all the way around or it could be clamped only on two ends. Furthermore, a rubber layer can be added to make the
clamped-clamped into a simply supported boundary condition. It would not be a true simply supported case, but that is the best that can be achieved for impact. The reason that a clamped boundary is the best for impact testing is that the plate will develop high membrane forces and will produce a good vibration pattern. Also, most manufactured parts that are subject to impacts, such as army tanks, have rigidly attached components. Because of these reasons, a clamped boundary condition was used. It should be noted at this time that the specimens do not have to be plates. Beams can be tested in the same specimen clamp.

Now that the basic parameters had been decided upon, it was time to begin to think about the actual clamp layout. There were a few basic requirements. The first was that it obviously had to fit on the milling machine table. The X-Y table has two T-slots in it which are used to tighten objects down to the table. To hold the specimen clamp to the table, some T-shaped pieces which fit into the slots are used. These pieces have threaded holes in them where bolts can be screwed into. Then, bolts are simply put through holes in the specimen clamp, screwed into the T pieces, and tightened down. This pulls the T-shaped pieces up tightly against the top of the slots in the table which in turn secures the specimen clamp in place. The size of these bolts obviously dictated the size of the holes required in the specimen clamp.

The next design consideration was that the specimen clamp had to be located in the exact center of the X-Y table in the front to back direction. This is so an equal amount of the specimen can be tested on both sides of its centerline. This required some precise measurements and mathematical calculations when trying to locate the holes that would be used to hold down the table. The two slots in the table were not equidistance from the
two edges of the table. A slight error was made when the clamp was designed and these holes were located in the wrong place. The calculations were redone, and new holes were drilled in the proper location.

The final requirement that this clamp had to include was flexibility for future use. This was a design requirement set forth by the author. The reason the design had to be flexible was because this tester is to be used for research, where many different ideas and concept are tried. The boundary conditions will probably not always be 12" x 24". To make this specimen clamp versatile, sub-clamp assemblies were used. These sub-clamps sandwich the specimens. They are held tightly together with bolts along all edges. Then, these sub-clamps are secured to the main specimen clamp assembly with quick release toggle clamps. These toggle clamps consist two main parts. One is the handle and the other is the part that applies the pressure to whatever is being clamped. The two are connected by a lever mechanism which allows the clamping force input to the handle to be increased significantly at the pressure application point. The handle is pushed into a locked position which causes the pressure part to exert a large amount of force on the sub-clamp apparatus. This pressure holds the sub-clamp very securely and rigidly, while also allowing the easy removal of the sub-clamp by simply pulling on the handle out of the locked position. By allowing the actual sub-clamp assembly that holds the plates to be removable, multiple sub-clamps can be made with either identical or different boundary conditions.

There are many advantages associated with having removable sub-clamps. First, different size boundary conditions could be made without fabricating a whole new specimen clamp. All that would need to be done is to make a new sub-clamp assembly
with the desired boundary size and change the position of the sub-clamp bolt holes to ensure adequate clamping pressure. The other advantage of having removable sub-clamps is that each specimen can be left in its own individual clamp for the duration of its testing life without changing its boundary condition. This would be useful in case other specimens needed to be tested, but testing had not been completed on the sample in the sub-clamp. If this was ever the case, the sub-clamp assembly could simply be removed with the plate in it and a new sub-clamp assembly could be installed with a new specimen and possibly with a new boundary condition.

The last real design concern was how thick to make the steel framing. It was decided that the steel should be 3/8" thick. This was merely a judgment call. This decision was motivated by the fact that the specimen clamp should be fairly rigid. This is so not much energy is lost to the surrounding structure; yet, at the same time, the sub-clamps should not be so heavy that they cannot be easily lifted and removed. Again it was felt that 3/8" thick steel best matched these requirements. The design consists of the two steel sub-clamp pieces and the main clamp base, which has the toggle clamps attached to it. The mechanical drawings for these three piece can be seen in Figures 25, 26, and 27. Furthermore, a drawing that shows how these three pieces all fit together can be seen in Figure 28.

One important thing that should be noted from looking at Figures 25 and 26 is the two sets of holes around the perimeter. The sub-clamp was designed to allow for 1" of the specimen to be clamped all the way around the edge. Then, the outside bolts would be tightened to secure the specimens. Once this clamp was built, it was realized that the boundary condition was not acceptable as it was built. An aluminum plate was placed in
Figure 25: Mechanical Drawing of the Top Piece of the Sub-Clamp Assembly of the Specimen Clamp.
Figure 26: Mechanical Drawing of the Bottom Piece of the Sub-Clamp Assembly of the Specimen Clamp.
Figure 27: Mechanical Drawing of the Base Plate of the Specimen Clamp.
Figure 28: Schematic of the Assembled Specimen Clamp.

- Test Plate
- Guides
- Table Mobility 7 5/6"
the clamp for an initial set-up. It could be seen that the boundary condition was not clamping all the way around the edge. This observation was supported by the fact that the plate made an abnormal vibrating sound when it was impacted. It should not have done this if the boundary condition was uniform. Upon closer inspection, it was noticed that certain parts of the inner edge of the sub-clamp were not touching the plate. The reason for this was simple. The specimen plates have a certain thickness (in this case 1/16”). A certain space is occupied by this plate. So as the bolts were tightened, the outside edges of the sub-clamp got closer and closer to each other while the inside edges could not get any closer to each other than the thickness of the specimen. As the bolts were tightened more, the outside edges eventually touched each other. This in turn caused the opposite sides of the 2” wide sub-clamp edges to deflect upward and lose contact with the plate.

This problem could have been solved two ways. The first solution was to put spacers between the bolts around the whole outside of the sub-clamp. These spacers would have to be the same thickness as the specimen in order to still maintain good clamping force. This would keep the edge from deflecting, thus maintaining a good boundary condition. The second solution was to drill new holes in the sub-clamp which would allow bolts to clamp the specimen close to the inside edge. The one problem with this solution was that these holes would have to be drilled through the specimen plate itself. To locate these holes correctly in the specimen plate, the plate would first need to be placed in the sub-clamp. Then, one bolt at each corner of the outside row of holes would be lightly tightened to hold the plate in place. A 3/8” transfer punch (this is the same size as the holes) would then be used to indent the plate at the proper locations. Then, the plate would be removed and the holes would be drilled. This would be the best solution for aluminum (or any other metal) plates, but it is very difficult to drill good holes in
composites. Therefore, the solution was to drill the inner row of holes, while also leaving the outer row in place. This way, if metal plates are being tested, holes can be drilled in the plates and the inner row can be used. This is a simpler operation than cutting a number of little spacers with the same thickness as the specimen plate and trying to locate them correctly between the two pieces of the sub-clamp. It is also felt that the utilization of the inner row produces a slightly better boundary condition. On the other hand, when composite plates are being tested, spacers will probably need to be used in order to obtain an acceptable boundary condition. In actuality, drilling composites may not cause all that much damage, and the same procedure that was described for the metal plates could be used. This would only require the inner row of holes in future sub-clamp assemblies. Whether or not this is the case will not be known until drilling a full sized composite plate is actually attempted.

4.2 Fabrication of the Specimen Clamp

As stated, the plate was to made out of 3/8" thick steel. The first step was to order this steel. A standard 4' x 8' section of low carbon steel was ordered. This may seem like a lot, but this was the smallest size that could be purchased. Besides, the extra material can be used in the future to fabricate additional sub-clamp assemblies. The only other materials that had to be ordered were the toggle clamps. The order for these was placed with McMaster-Carr and consisted of six model 5111A43 vertical and horizontal mount toggle clamps and four model 5161A25 vertical handle weldable steel toggle clamps with straight bars. The maximum clamping force for the model 5111A43 clamps is 1000 lb, while the maximum for the model 516A25 can vary from 1000 to 450. The horizontal and vertical mount toggle clamps had to be used on the sides because of space considerations. If any other type of clamps had been used on the sides, they would have restricted the
mobility of the X-Y table by interfering with either the I-beams or the cantilever arms. Furthermore, a problem arose with the vertical handle toggle clamps. They turned out to interfere with the cantilever arms and had to be replaced. This will be discussed shortly.

At this point, the specimen clamp was ready to be fabricated. The steel was cut to size with a cutting torch and the edges were ground smooth. Then, a reference edge was marked on each of the two parts of the sub-clamp. This was to identify the correct orientation of the two pieces when they are assembled. Next, the two parts of the sub-clamp assembly were clamped on top of one another, and the holes were drilled in the proper locations. These first holes were drilled to the size required to tap the bottom part of the sub-clamp. By drilling through both pieces together, the holes would line up exactly on the two parts. Then, the holes on the bottom piece were tapped to produce the 3/8"-NC threads. Finally, the holes on the top part were drilled out to their designed diameter of 3/8".

The next step was to finish the clamp base plate. Notice in the design of the base plate in Figure 27 that there are six 2" x 3" x 3/8" steel plates welded around the edge. These plates were put there to attach the vertical and horizontal mount toggle clamps. Their size was dictated by the size of the bases of these toggle clamps. These clamps were to be welded on; so, instead of just welding them directly to side of the base plate, they were welded onto these steel mounting plates. The reason for this was that they could be more securely fastened this way. The base plate is only 3/8" thick and the toggle clamps' bases had to be mounted parallel to this vertical edge. If the toggle clamps had just been welded directly to the base plate, there would not have been as much weld area as if the steel plates had been used. So, the steel plates were welded on all sides to the clamp base plate.
Then, the toggle clamps were welded all around their bases to the flat vertical sides of these plates all around their base. This secured the toggle clamps very good.

Next, the holes that would be used to anchor the specimen clamp to the milling machine table were drilled. As mentioned earlier, these were drilled in the wrong location due to a calculation error by the author. This mistake was realized immediately once the base plate was first put on the X-Y table. The base plate was clearly not centered on the table. This mistake was corrected, and new holes were drilled in the correct locations.

The next step in the fabrication of the specimen clamp was to attach the remaining four toggle clamps. It was decided that these clamps would be attached with bolts. Holes were drilled and tapped in the correct locations and the vertical handle toggle clamps were attached. It was decided later that it would have been easier to just weld these clamps on. As it turned out, it was a good thing that they were not welded. When the specimen clamp was completed and put on the machine, it was immediately realized that the vertical handle toggle clamps would not work. The handles of these clamps stick straight up when they are in the clamping mode. The problem, as mentioned earlier, was that these handles stuck up too far, and they were interfering with the cantilever arms. This restricted the movement of the X-Y table, which was clearly unacceptable.

The solution to this problem was to remove these clamps and purchase and install new model 5128A13 horizontal handle toggle clamps. The handles on these clamps assume a low profile horizontal position when they are in the clamping mode, thus they would not get in the way of the cantilever arms. These clamps were just welded on, because it was easier than going back and drilling and tapping sixteen new holes.
The final step in making the specimen clamp was to weld two small pieces of steel along each long edge of the clamp base plate. These can be seen in Figure 28. The purpose of these 1" x 1/2" x 1/4" steel pieces was simply to guide the sub-clamp so it would sit in the same location on the base plate every time. One piece was located near each corner of the sub-clamp on the long side of the clamp base plate. They were welded along three sides in an upright position with the 1/2" length being perpendicular to the base plate. The only side that was not welded was the one nearest the sub-clamp. This was so the sub-clamp would not sit on top of a weld bead, which would cause it to be uneven. None of these steel locator pieces were needed on the two ends because the bases of the horizontal toggle clamps butted right up against the sub-clamp, serving as guides in this direction.

The specimen clamp as it was designed was complete; however, during initial impacts of a 1/16" thick aluminum plate with medium to heavy weight, it became apparent that having the bottom of the specimen plate only 3/8" above the steel base plate was not enough. The problem was that the plate would plastically deform to the point where the part of the plate where the tup hit the plate would deflect all the way to this base plate. This caused the forces to go way up because the base plate is very stiff. This problem was realized during the initial stages of the instrumentation process. The force readings at the peak of the impact became abnormally high during impacts with higher weights. Upon removing the aluminum specimen plate, flat spots from where the specimen plate hit the base plate could be seen. The solution was to weld a 2" wide lip onto the base plate directly under where the sub-clamp assembly sits. This lip was made out of 2" x 3/8" steel bar. This raised the bottom of the specimen plate to a total height of 3/4" to allow for higher deflections. This then completed the specimen clamp.
Chapter 5
Instrumentation and Data Acquisition Programming

5.1 Derivation of the Instrumentation Parameters

Now that the mechanical portion of the impact tester had been completed, the next step was to instrument the set-up. The many benefits of instrumented impact testing were described in the introduction. Basically, the critical parameters associated with the response of a specimen can be observed by utilizing instrumentation. Parameters such as the energy absorbed by the specimen throughout the impact can be calculated, as can the velocity of the tup and the deflection of the plate during the event. All this information can be calculated by simply acquiring two pieces of information. One is the velocity of the falling weight at the moment of impact. This can be calculated from the amount of time that the velocity card occludes the signal from the emitter/detector. The other important piece of information that must be obtained is the force applied to the specimen over the time of the impact. This data is gathered by the force transducer tup. The equations that are used to calculate the rest of the parameters will be presented in this section.
The two most important benefits that instrumented impact testing provides are the force versus time and the energy versus time plots. Much can be learned from looking at these plots. Things such as the yield point of the specimen can be determined. Also, the maximum load applied by a certain impact energy level can be found, as can the absorbed energy at this maximum load. These are just a couple of the added benefits that instrumenting an impact tester can provide. It basically allows the impact characteristics of the specimen to be observed.

There are numerous publications which go into great detail about the issues surrounding the instrumentation of impact testers. The basic idea of an instrumented impact test is that the tup has a certain amount of kinetic energy just prior to the impact. Some of this energy is then transferred to the specimen during the impact event. According to Ireland (1973), the falling weight's energy is then reduced by an amount \( \Delta E_0 \) and

\[
\Delta E_0 = E_I + E_{SD} + E_B + E_{MV} + E_{ME},
\]  

(2)

where

- \( E_I \) = increment of energy required to accelerate the specimen from rest to the velocity of the falling weight,
- \( E_{SD} \) = total energy consumed by bending the specimen,
- \( E_B \) = energy consumed by Brinell-type deformation at the specimen load points,
- \( E_{MV} \) = energy absorbed by the impact machine through vibrations after initial contact with the specimen, and
- \( E_{ME} \) = stored elastic energy absorbed by the machine as a result of the interactions at the specimen load points.
Obviously, the impact tester was built on a very rigid milling machine base and has a solid steel specimen clamp. Because of this, the energy lost to the impact machine, \( E_{MV} \), can be neglected. Furthermore, the hammer-striker assembly can be treated as a rigid body if \( E_{ME} \) is small enough. This energy is the strain energy of the tup cylinder. The replacement tup cylinder is 1" in diameter and is nearly solid. Recall that its length is 8.544". Assuming a maximum load of 1000 pounds and a beam of uniform cross-section, the strain energy absorbed by the tup cylinder can be estimated by

\[
U_N = \int \frac{N^2}{2EA} \, dz, \tag{3}
\]

where \( N \) is the axial load, \( E \) is Young’s Modulus (30 x 10^6 psi for steel), and \( A \) is the cross-sectional area of the cylinder with the hole drilled in it (0.723 in^2) (Boresi and Sidebottom, 1985). The strain energy at this maximum load works out to only 1.64 x 10^{-2} ft-lb. This is clearly insignificant when compared to the fact that the energy that will be absorbed at this force level is likely to be greater than 15 ft-lb. This means that this energy term can be neglected, and the falling weight can thus be considered a rigid body. All that is left, then, of the energy equation is

\[
\Delta E_0 = E_I + E_{SD} + E_B. \tag{4}
\]

These energies are all energies that are absorbed by the plate during the impact. In other words,

\[
\Delta E_0 = E_a, \tag{5}
\]

as calculated from the measured values of force.

The basic equation to obtain the absorbed energy is

\[
E_a(t) = \int_0^t PV \, dt, \tag{6}
\]
where time 0 is the instant of impact and time t is any instant of time after that (Ewing and Raymond, 1973). Recall that the force versus time data are being acquired with the data acquisition system. Also, the initial velocity is being calculated from data obtained from the emitter/detector. The equations to find this velocity will be covered in the next section.

The derivations of the equations that are required to fully instrument the impact tester were extracted from the Dynatup GRC 730-I Operator's Manual. This is the data acquisition and processing package that is used with the Dynatup model 8200 impact tester in the ESM department. They can also easily be derived from laws of physics. To begin, the total force acting on the tup hammer is the sum of the resistive force of the specimen, \( p(t) \), as measured by the piezoelectric force transducer, and the gravitational force in the direction of travel (32.174 ft/sec\(^2\)). Therefore, assuming that data is starting at \( t=0 \),

\[
f(t) = mg - p(t).
\]

Then, from Newton's law, the acceleration at time \( t \) can be found to be

\[
a(t) = \frac{f(t)}{m} = g - \frac{p(t)}{m},
\]

where \( m \) is the mass of the falling weight. Next, the velocity at time \( t \) can be calculated from the basic laws of motion. This is

\[
v(t) = v_{imp} + \int_0^t a(t) \, dt = v_{imp} + gt - \frac{1}{m} \int_0^t p(t) \, dt.
\]

In the 730-I operating manual, this equation did not include the impact velocity term; however, it was deduced that it needed to be included from looking at the equation. An initial condition is needed as a constant of integration for the equation to make sense.
the instant of impact (i.e., \( t=0 \)), the velocity of the tup is the velocity just prior to impact (\( v_{\text{imp}} \)). If this initial velocity term is not included, the equation would show that the velocity at \( t=0 \) was 0.

Continuing, the displacement of the specimen can be calculated at any impact time \( t \) by integrating the velocity data

\[
x(t) = \int_0^t v(t) \, dt = \frac{1}{2} gt^2 - \frac{1}{m} \int_0^t p(t) \, dt.
\]

(10)

Naturally, there is no initial displacement at \( t=0 \), so the constant of integration is 0.

Next, all that is required is to take this data and calculate the absorbed energy. Assuming a conservaion of total energy of the falling weight and the specimen throughout the test, the energy relationship can be written as

\[
E(t) = T(t) + V(t) + E_a(t) = \text{constant},
\]

(11)

where \( T(t) \) is the kinetic energy and \( V(t) \) is the potential energy of the falling weight at a given instant of time. If \( t=0 \) is the time of impact and \( x=0 \) is the displacement at impact, equation (11) can be written as

\[
E_a(t) = T(0) - T(t) - V(t).
\]

(12)

This is because both the potential energy and the absorbed energy will be 0 at \( t=0 \). By rewriting the kinetic energy terms as \( 1/2mv^2 \) and the potential energy in terms of \( mgx(t) \), the equation for absorbed energy can be written as

\[
E_a(t) = \left( \frac{m}{2} \right) (v_{\text{imp}}^2 - v^2(t)) + mgx(t).
\]

(13)
As stated previously and as can be seen from the above equations, the other parameter that is required is the impact velocity. The equations for this were also found in the GRC 730-I Operating Manual, but could have just as easily been derived from basic equations of motion for an object moving in a straight line under the accelerating force of gravity. The impact velocity can be computed from simply knowing the time that the emitter/detector signal is occluded and the width of the velocity flag. By using the equations of motion, an accurate velocity can be calculated. If the width of the flag was just divided by time the beam was occluded to obtain the impact velocity, the accelerating effects due to gravity during the measurement process would not be taken into account. Not only would the falling weight accelerate from the time the flag first interrupted the signal to the time the signal first reappears, but it would also accelerate from the time the signal reappears to the time the tup hits the plate. These velocity changes may be small, but they are nonetheless important.

For the derivations of the formulas, it is first necessary to define a few terms.

\[ t_0, x_0, v_0 = \text{the time, position, and velocity at an arbitrary initial time.} \]
\[ t_1, x_1, v_1 = \text{the time, position, and velocity when the emitter/detector signal is first occluded.} \]
\[ t_2, x_2, v_2 = \text{the time, position, and velocity when the beam first reappears.} \]
\[ w_{\text{flag}} = \text{the effective width of the flag.} \]
\[ t_{\text{ocel}} = \text{the time the signal was occluded (i.e., } t_2 - t_1). \]
\[ t_{\text{imp}} = \text{the amount of time after the signal was initially occluded until the impact occurred.} \]
\[ v_{\text{imp}} = \text{the velocity of the falling weight at the instant of impact (as defined earlier).} \]
To begin, the velocity and position at arbitrary points can be found from

\[ v = v_0 + g(t - t_0). \]  
(14)

This is then solved for \( v_0 \).

\[ v_0 = v - g(t - t_0) \]  
(15)

Next, \( v \) is set at \( v_1 \) and \( v_2 \) with \( t \) being set to \( t_1 \) and \( t_2 \), respectively.

\[ v_0 = v_1 - g(t_1 - t_0) \]  
(16)

\[ v_0 = v_2 - g(t_2 - t_0) \]  
(17)

These two equations are set equal to one another to give

\[ v_2 = v_1 - g(t_2 - t_1). \]  
(18)

To compute \( v_2 \) in terms of \( x_2, x_1, t_2, \) and \( t \), an expression for \( v_1 \) must be found. This is again done using the equation of motion for position

\[ x = x_0 + v_0(t - t_0) + \left( \frac{g}{2} \right)(t - t_0)^2. \]  
(19)

This equation is solved for \( x_0 \).

\[ x_0 = x - v_0(t - t_0) - \left( \frac{g}{2} \right)(t - t_0)^2 \]  
(20)

By setting the \( x \) equal to \( x_1 \) and \( x_2 \) and \( t \) equal to \( t_1 \) and \( t_2 \), the following equations can be found:

\[ x_0 = x_1 - v_0(t_1 - t_0) - \left( \frac{g}{2} \right)(t_1 - t_0)^2, \]  
(21)

and

\[ x_0 = x_2 - v_0(t_2 - t_0) - \left( \frac{g}{2} \right)(t_2 - t_0)^2. \]  
(22)

These two equations are equated and solved for \( v_0 \) as

\[ v_0 = \frac{x_2 - x_1 + \left( \frac{g}{2} \right) \left[ (t_1 - t_0)^2 - (t_2 - t_0)^2 \right]}{t_2 - t_1}. \]  
(23)
By letting the initial values (i.e., $t_0$, $v_0$, and $x_0$) be the point when the flag was first occluded (i.e., $t_1$, $v_1$, and $x_1$), an expression for $v_1$ can be obtained.

$$v_1 = \frac{x_2 - x_1}{t_2 - t_1} - \left(\frac{g}{2}\right)(t_2 - t_1) \quad (24)$$

Plugging this result into equation (18) gives

$$v_2 = \frac{x_2 - x_1}{t_2 - t_1} + \left(\frac{g}{2}\right)(t_2 - t_1). \quad (25)$$

This can be written another way with the simplified parameters defined earlier as

$$v_{\text{meas}} = \frac{w_{\text{flag}}}{t_{\text{accl}}} + \frac{8t_{\text{accl}}}{2}. \quad (26)$$

Finally, by using the same equations of motion, the additional velocity that the tup picks up from the time that the signal reappears to the time that the tup hits the plate can be included. This additional velocity is gained due to gravity. The equation for the impact velocity becomes

$$v_{\text{imp}} = v_{\text{meas}} + g(t_{\text{imp}} - t_2), \quad (27)$$

where $(t_{\text{imp}} - t_2)$ is the amount of time from when the signal first reappears until the tup hits the plate. The difference between $v_{\text{meas}}$ and $v_{\text{imp}}$ will be very small, but it still is important in order to obtain accurate velocity measurements. The difference between these two values would be greater if the velocity flag was adjusted too high during set-up.

### 5.2 Critical Concerns During Instrumented Impact Testing

There are many other important factors involved with instrumented impact testing which must be discussed. A very good paper by Cherish and McMichael (1985) covers many of the important issues associated with data interpretation. It is also a good reference for those just becoming acquainted with instrumented impact testing. All the
elements presented in this paper will not be covered in this section, but some of the most important topics as they apply to the impact tester that was built will be presented.

The parameters that need to be determined before conducting tests with this tester are the sampling duration (or test time) and the impactor weight. The impactor weight can be determined by adding together the weights of the individual components. The add-on weights have this value stamped on them. Furthermore, the falling weight and tup in the frame configuration weighs 7.348 lb, while the weight is only 3.518 lb when in the single crosshead configuration. These values include the weight of everything that is required for testing in the particular configuration.

With today's modern computerized data acquisition systems, the sampling rates can be set very high to increase resolution and the output plots can easily be re-scaled. For these reasons, the time range selected is not as critical as it used to be. The reference provides a four step procedure for choosing an acceptable time range. The list is as follows:

1. Consider the specimen to be tested and estimate the deflection that will be required to obtain complete fracture. A best guess made from past experience is all that is needed for this.

2. Convert deflection to time by the following equation

\[ t = \frac{d}{v} \]  \hspace{1cm} (28)

where \( t \) = time, \( d \) = the expected deflection to complete failure, and \( v \) = the impact velocity. Be sure to use a consistent set of units.

3. Increase the time by a factor of two for safety. An even larger safety margin (such as four or five) may be appropriate if the test is a "low-blow" test, in which the
impactor will bounce off the specimen. This is what will be encountered with most of the testing that is planned to be done at CIMSS.

4. Remember that the test time setting must be reevaluated each time the impact velocity or test specimen material is changed.

This time range is the total time during which the date acquisition system is taking data. To check the validity of this calculation for this tester, a 0.063" thick aluminum plate was secured in the specimen clamp. Then, after the data acquisition program that will be described in the next section had been completed, some initial data was gathered from one of the "low-blow" impacts. Without going into much detail at this time, the impact event lasted 11.79 msec. Also, the velocity was 6.772 ft/s and the maximum displacement was 0.3585". Since it is a rebounce test, the maximum displacement is used for the calculation. Using the above rules with a factor of safety of four, the recommended time range would be 17.6 msec. The actual time range used in this test was 18.8 msec. With this time range, there were good zero baselines before and after the impact event. So, it can be seen that these guidelines for the time range appear to be acceptable. They give enough time around the actual impact event to provide a good zero baseline. It should be noted that the data collection begins when the emitter/detector is first interrupted by the velocity flag. So if this flag is wider, it will take a longer time for the tup to first hit the specimen after the sensor is tripped. A 0.585" wide flag was used in this test, and the time range appears to be acceptable. If it were any wider or narrower, a slightly larger or smaller range should be used.

Another important factor in instrumented impact testing is the time resolution. This is simply the total time range chosen divided by the number of points gathered by the acquisition system during the test. In other words,
\[ r = \frac{t}{n} \]

(29)

where

\[ r \quad = \quad \text{the resolution with units of time per point}, \]
\[ t \quad = \quad \text{the time range selected, and} \]
\[ n \quad = \quad \text{the total number of points collected during the test}. \]

The important use of this resolution is in the calculation of the number of points defining the actual impact event, which is

\[ p = \frac{t_t}{r} \]

(30)

where

\[ p \quad = \quad \text{the number of points actually defining the impact event itself}, \]
\[ t_t \quad = \quad \text{the total time required for the test, and} \]
\[ r \quad = \quad \text{the resolution of the test}. \]

Cherish and McMichael indicate that, as a rule of thumb, at least 200 points are required to adequately define a test plot. This number depends largely on the type material behavior exhibited by the specimen. Materials that behave in a ductile manner with a relatively smooth, monotonic load curve require fewer points than those that have sudden failures, such as composites. Examining the resolution of the initial test with the aluminum plate, 1024 data points were gathered over 18.8 msec. The actual impact lasted 11.79 msec. This means that there were approximately 642 data points defining the impact event. This is much above the recommended minimum 200 data points.

One final issue mentioned in the reference that warrants discussion is filtering of the data. Many data collection systems incorporate analog filters to reduce "noise" caused by
vibrations. It was not believed that any analog filter was needed because of the high impedance force transducer that was used. This automatically reduces much of the noise caused by extraneous vibrations. The reference goes on to recommend that with computer acquisition systems, digital filtering as opposed to front-end analog filtering should be used. With the digital filters, the filtered data can be compared with the unfiltered data to see the filtering effects. This sounded like a good idea, and it was decided to employ digital filtering into the data acquisition program.

The final issue that will be discussed before moving on to the data acquisition programming is the different deformation stages of the impact event. Figure 29 shows an idealized load deflection plot (Knakal and Ireland, 1985). This curve is important in that it helps to describe the behavior of specimens at different stages. For this reason, it was decided to include a load versus displacement plot into the program. This plot, along with load versus time and energy versus time plots, provide the most important information about the material behavior during impact.

Stage A - This is where the inertial loads are encountered. These loads are caused by the specimen having to accelerate to the velocity of the tup during the initial stages of the impact. This inertial loads can also be seen on the force versus time plot. According to the reference, the load values in this stage are not representative of those required for the indicated deflection by static mechanics relationships.

Transition A - This is where the linear load-deflection deformation begins.

Stage B - This is the linear load-deflection deformation section. The specimen is basically behaving in an elastic manner, with the exception of some local plastic deformation around the impact head.
Figure 29: Idealized Deformation Stages and Transitions for Puncture Testing of Flat-Plate Specimens by a Hemispherical Probe (from Knakal and Ireland, 1985).
Transition B - This is the yield point, which is the beginning of the plastic deformation.

Stage C - This is where the first large scale permanent plastic deformation is occurring.

Transition C - Here occurs the maximum load applied during the impact event.

Stage D - In this stage, slow rate deformation of the specimen is occurring until failure occurs.

Transition D - This is where the material fails.

Stage E - In this section, rapid unloading occurs as the specimen fails and ceases to put up any more resistance.

In "low-blow" impacts, this curve should go up to the maximum load then round off and slope back toward zero. There will still be some plastic deformation in Stage D. The final deflection will be somewhere greater than zero. Here ends this section covering some of the important factors involved with instrumented impact testing. As indicated, there are many other issues concerned with these tests. The listed references in this section go into much more detail.

5.3 Investigation into Data Acquisition Software Packages

Now that the equations had all been derived and the critical issues had been learned, the next step was to program the equations into a data acquisition system. There were two main options when it came to data acquisition. One was to write an individually tailored computer code in a computer language such as C++, FORTRAN, or BASIC. This option required extensive knowledge of the chosen computer language and could have been very time consuming and frustrating. The advantage of this option is that the program is tailored to the individual operation, so it may run a little faster than a commercial data acquisition package. Purchasing a commercial software program was
another option. There are many commercial packages available, with some being more flexible and easier to use than others. As indicated, the main advantage of these packages are that most of the data acquisition procedure has already been set up. The problem with some of the less expensive programs is that they cannot be programmed. They can only acquire and display data, with limited processing capabilities. Some of these programs allow the user to write their own programs in C to process and handle the flow of the data. One final note is that many of these programs are sold by companies that also make data acquisition boards, and the software only works with that company's boards.

The other important part of the data acquisition system is the data acquisition board. This is the plug-in board in the computer that converts the continuous analog data to discrete digital data points. The main way that these boards are characterized by are by their sampling rate. Other options such as various analog filters can also be included on the boards. When the data acquisition part of this project was beginning, it was decided that a 50 kHz sampling rate DT2821 board from Data Translation would be used. This board had just been purchased by CIMSS for another purpose and would be available for use with this tester. After a while, it was realized that this board would not be acceptable. The impact event lasts a very short time. In order to have good resolution, a certain number of data points would need to be gathered during the impact. In order to acquire the sufficient number of points, it was estimated that the sampling rate for the force transducer channel would have to be in excess of 25 kHz. In fact, literature from two commercially produced testers indicate sampling rates of 21 kHz and 33 kHz. So, if it is estimated that 30 kHz would be needed, only 20 kHz would be left over to sample the emitter/detector channel plus any other sensor channels that may be used in the future for the impact location detection experiments that this tester was intended for. The sum of
the sampling rates on the individual channels cannot exceed the capabilities of the board. If the emitter/detector signal is not sampled fast enough, the initial velocity data could be faulty. This is because the exact instant the signal is blocked and reappears could be between data points. This problem would be the worst with a narrow velocity card and a low sampling rate. So, it was decided that a new board with a higher sampling rate was needed.

CIMSS owns many boards from Data Translation, so it was decided that a board would be purchased from them. For a restocking fee of 10% of the original cost of the board, the 50 kHz board could be returned. A new board would still be needed. It was decided that 150 kHz would be acceptable for this tester. The DT2821-F-8DI board was chosen. This board only has the capabilities for differential inputs. There are two types of inputs for data acquisition boards, single-ended and differential. Single ended inputs are all referenced to a common ground point, and are typically used when the input signal is greater than 1 V and the input wires are less than 15 feet long. The differential input channels each have their own ground reference. This type of input reduces noise from the wires or instruments (National Instruments, 1994). The force transducer wire used is 15 feet long, so differential inputs were chosen. The DT2821-F boards only have the capability of sampling one type or the other, but some of the other boards have the capability to choose either option on the same board. The disadvantage of differential inputs is that the number of available channels is cut in half, from 16 to 8. This does not cause any problems with this system because it is not anticipated that more than 8 channels will ever be needed.
There were a couple of additional pieces of hardware that had to be purchased to complete the data acquisition system. The first was a power unit for the force transducer. A model 480E06 battery powered power unit was purchased. This unit conditions the output voltage from the force transducer and passes on the conditioned signal to the A/D board.

The other piece of hardware that was required was a screw terminal panel. This is how the instruments interface with the computer. Basically, the various instrumentation wires are connected to pins and the signals are passed on to their respective places in the A/D board. A standard DT-707 screw terminal panel was purchased for use with the data acquisition board. This panel was factory configured for single-ended inputs, so it had to be converted for differential input once it was received. This was done by simply soldering 10 kΩ resistors in the positions indicated in the operator's manual. When using differential inputs, the positive incoming signal is connected to the correct channel input pin and the instrument's ground is connected to that channel number's return pin. The resistors simply provide a path for each channel's ground to be connected to the analog ground of the board. Finally, jumpers were connected across all unused channels to short them out and prevent any unwanted interference.

During the initial stages of testing with the data acquisition board, it was realized that the A/D board had a problem known as crosstalk. The way the tester is set up, the 5 V signal from the emitter/detector goes to one channel and the force transducer signal goes to another. The voltage across the force transducer channel should be zero unless a force is being applied. What was happening on long duration continuous data acquisition runs was that voltage from the emitter/detector channel was spilling over and actually causing
around 3 V across the force transducer channel. This is what is meant by crosstalk. It was believed at first that there might have been a problem with the A/D board, but this same problem was encountered with two other Data Translation boards at the CIMSS lab and one Data Translation board in a computer at the mechanical engineering Controls Lab. This seemed to indicate that it was a problem with all that company's boards when the system is set up in this manner. Additionally, the crosstalk would only occur when both the channels were sampled at the same time. Independently, the channels worked fine. Also, the voltage would not be immediately at the 3 V value. It would actually ramp up from 0 V. It was decided that this crosstalk, then, would not be a real problem because the impact duration is so short. In fact, the force data had a good zero baseline during all the initial tests. Just to be on the safe side, Data Translation was contacted for advice. Their suggestion was to connect a 10 kΩ or 100 kΩ resistor across the positive input and the ground of the force transducer. This just acts as a load to bleed off the voltage, and it will not affect the data gathered. The 100 kΩ resistor seemed to work the best, and it was permanently attached across the input and ground pins.

This screw terminal panel was then mounted inside a 6" x 10 1/2" x 3 1/2" aluminum enclosure box. A 5" x 1/2" slit was cut out of one end to allow for all the wires to enter the box. The computer interface cable was connected to the A/D board rear panel connector and to the screw terminal panel. This connection should always be performed with the computer turned off. Other connections to the screw terminal panel include the V_e and ground wires from the emitter/detector, and the positive and ground wires from the force transducer power unit. The wires from the emitter/detector were run into channel 0 and the force transducer wires were connected to channel 2.
As mentioned earlier, the data acquisition system must be triggered to start taking data. The screw terminal panel has a trigger input pin. For this board, the trigger channel must have greater than 2.0 V on it at all times. Then, the system will trigger when the signal trips below 0.8 V. It will trip the instant the trigger goes low. This type of trigger worked out perfectly for this tester. The signal from the emitter/detector was also connected to this trigger pin and its ground. Then, when the flag occludes the signal, the voltage will drop to near zero and data acquisition system will immediately trigger.

During initial testing, a problem was encountered with this triggering of the data acquisition system. The flag was passing through the emitter/detector and properly triggering the system, but it would also pass back up through the sensor again if it rebounded from the specimen. This was causing an acquisition error, because the system was retriggering again before the data from the first trigger had been completely gathered. This problem was addressed to the technical support staff at Data Translation, and the recommended solution was to use a timer chip to hold the trigger signal low for a certain time duration after the initial trigger. This way, the system would not retrigger. The CIMSS lab assistant, Manish Sabu, was knowledgeable on these chips, and a trip was made to the Electrical Engineering building to pick one up.

A Texas Instruments model SN74LS123N mono-stable multivibrator was obtained. The connection scheme was obtained from a Texas Instruments catalog, and the chip was wired. By connecting different capacitors and resistors in the wiring network, different time durations can be achieved. There is an equation relating the values of these two parameters to the switching time duration. A simple circuit was made with the chip on a wiring board according to the wiring diagram shown in Figure 30. In addition, the labeled
chip pins and the appropriate input and output values can be seen in Figure 31. The chip has a recommended $V_{ee}$ of 5 V. The voltage on the wire that powered the emitter/detector was around 8 V DC, even though the converter was supposed to provide only 6 V. The wires coming out of the converter were first connected to the circuit board. Then, the voltage was stepped down to 5 V using a 150 $\Omega$ resistor in series with the positive wire. Then, this 5 V was sent to both the emitter/detector like before and to the $V_{ee}$ pin of the timer chip. The other input to the chip consisted of the return signal from the emitter/detector. The output wires were connected to the appropriate pins so that the signal would go low and hold low for approximately 4 seconds after the trigger was received. To achieve this time, a 100 $\mu$F capacitor and a 100 k$\Omega$ were used. The outputs from the chip were connected to the trigger pin and the ground. The circuit board was then attached inside the same enclosure box as the screw terminal panel. This solution worked perfectly and the system now functions correctly.

Now that all the hardware had been taken care of, the other main concern was which software option to pursue. The author was not proficient in any computer languages other than FORTRAN, so it was decided to pursue commercial software packages. There were two software packages that were chosen. Both options were compatible with Data Translation boards. One option was a program called Labtech Notebook, while the other program was a brand new software package called DTVEE. This package was developed from a similar program developed by Hewlett-Packard called HPVEE. DTVEE is what is known as a graphical programming language. Many different programming commands are combined into one graphical icon, which performs a certain function. Each individual graphical block is then connected by a "wire" on the screen. Another similar graphical programming software package that is widely used is LabView by National Instruments.
Figure 30: Wiring Diagram for the Timer Chip.

Figure 31: Timer Chip Pins and Input and Output Values Used for this Application.
The advantage of these packages is that the development time of projects can be reduced significantly. They also provide a graphical user interface. Because of this ease of use and short development time, it was decided to purchase the DTVEE software package. The software was still in its final stages of development and was not available until December 1, 1993. At this time, the software was received and the programming began.

5.4 Programming the DTVEE Data Acquisition Software Package

The first step once this software package was received was to install it on the computer with the new DT2821-F-8DI board and go through the Getting Started manual and programs. Then, the two other manuals that came with the program, Using DTVEE and Advanced Programming Techniques, were read. It became immediately apparent that the right choice had been made in choosing this program, as it was extremely easy to learn to use. This program includes example programs that describe nearly every operation that the program offers, along with a very thorough on-line help directory. These were both very helpful when learning how to use the program and during the actual programming operations. Once the familiarization was complete, the programming for the instrumentation of the impact tester was started.

Complete printouts of the two programs that were written can be found in Appendix A and Appendix B. Each graphical icon has a number next to it. These numbers will be referred to throughout the description of the programs. It should also be pointed out the functions of the individual pins on the icons. The pins on the left side of the objects are the input pins, while the pins on the right side are the output pins. The top and bottom pins are the sequence pins. These are used to control the flow of the program, as far as which objects execute before others. The top pin is the sequence in pin and the bottom
pin is the sequence out pin. This completes the brief introduction to the program. The best way for a new operator to really become familiar with this program is just to sit down at a computer and work through the programming exercises that are described in the manuals.

As mentioned already, two programs were written for use with this impact tester. The first program is called IMPACT.VEE and is the one that is used for the actual testing and data acquisition. The other program is called OLDFILE.VEE and is used to call-up old data files that had been saved in the impact program. The two programs will each be described in a step-by-step manner. It should also be noted that English units were used throughout this program. This was done because the output from the Dynatup tester also is in English units. Additionally, many references list English units. So, this was done for ease of comparison during calibration. DTVEE has a conversions library which can be used in the future to convert to metric units if it is so desired.

The first program that was written is shown in Appendix A and is called IMPACT.VEE. This program is the one that first acquires the data, then processes it, saves it in a file, and plots the results. The user then has the option to go on and digitally filter the raw data if significant noise is encountered. In this project, much refinement was done throughout the development of these programs; but, if in the future new enhancements are desired, program improvements can easily be made.

The obvious place to start a program is with the user defined inputs. This program has two views. One is a detail view where the whole program can be seen, and the other is a panel view where a simplified user interface only shows the pertinent objects. This
software package also allows for sub-programs to be incorporated into the program through the use of UserObjects and UserFunctions. The only difference between the two is that UserFunctions can be stored in a library so that other programs can use them. This first block (321), is the initial inputs UserObject. The nice feature of UserObjects is that they can have their own panel view, which pops up when the UserObject executes. This initial inputs object consists of the user defined time range (321.0), the total weight of the falling head in pounds (321.1), a filename input which should also include the appropriate directory in which to store the data (321.2), and an input for a specimen ID (321.4). Recall that the falling weight weighs 3.518 lb in the single crosshead configuration and 7.348 lb when it is in the frame configuration. Plus, any add-on weight must be included in the total weight. This object also includes a notepad box which gives instructions to the operator (321.7). These inputs are all set so that they will not be cleared at the start of the program. This way, the operator does not have to keep entering the same set-up over and over again if multiple tests are being performed with the same experimental set-up. When the user has completed all the entries, the continue button is pressed and the UserObject is exited. The pop-up panel view can be seen in Figure 32.

The values that were set in the initial inputs object are then passed on. The weight, filename, and specimen ID are saved as global variables. Once a global variable has been defined, it can be called from anywhere in the program, including from formulas. Also at this time, the exact time of the program's execution is saved as a global though the use of the now() command (355). This gives the number of seconds that have elapsed since 0 AD. The To String (356) command then formats this value into the day of the week, the day of the month, the month, the year, and the time in H:MM:SS format. For example, a typical value would look like Sat 29/Jan/1994 1:03:44 PM. This time and date is saved as
Figure 32: Pop-up Panel View of the Initial Inputs UserObject in IMPACT.VEE.
a global (350) and is used later as the title of the data plot so the operator can know what
time the test took place. The final output from the initial inputs object goes into a formula
box (333). The purpose of this box is to compute the sampling rate for the force
transducer channel. The Dynatup GRC 730-I data acquisition gathers 1024 data points
over the duration of the test. This program was set up similarly. So, the user must first
define the test duration, then the formula box divides 1024 by the time duration to obtain
the samples per second (or sampling rate) for the force transducer channel.

This determined sampling rate is input into a Concatenator (121), which makes a 1-D
array of the sampling rate for the velocity channel (120) and that determined for the force
transducer channel. The velocity channel was chosen to be sampled at 30 kHz. This gave
greater than 99% accuracy during the measurement. The output from the Concatenator
goes into the Set Sampling box (101). Here, the initial sampling rates set in the A/D
Config box (5) are changed. The appropriate channels that are to be sampled are set in
this configuration box. Other things set in this box are the trigger configuration (external),
channel gains, channel types, and burst mode enable. Basically, this box configures the
software to the type of board that is in the computer. The output from this configuration
box then goes into a Get Sampling box (339) and a Get Data Mode box (336). These two
objects just get the configuration from the configuration box. The outputs from these go
into the same Set Sampling box (101) as before and a Set Data Mode (337) box. By using
the Get objects, only the one configuration parameter that is to be changed needs to input.
The rest of the system configuration can just be passed on. The one configuration item
that is being changed is the force transducer sampling rate. The Data Mode boxed had to
be included because the A/D configuration gets set back to the system defaults when the
Set Sampling icon is used. Since this system is only acquiring 1024 data points after the
trigger, the burst mode must be reset. This was required because the system default is continuous mode. The burst mode is set by the Set 1024 Points box (338).

The output from the Set Sampling box goes to another Get Sampling object (153). This is used to extract the exact sampling rates that were used to acquire the data. These rates were required in order to calculate the time increments of data points for future calculations. The data acquisition system will always sample channels at multiples of others. For example, if the sampling rates are input to the Set Sampling box as 30 kHz and 50 kHz, the software will automatically round these to some multiple of each other such as 27.33 kHz and 54.66 kHz. This is done because the A/D conversion is performed by sweeping all the active channels. When the sampling rate of one channel is say half that of another, the slower channel will get sampled once every two scans through the channel list. The Get Values boxes (155 and 196) simply extract the appropriate value from its location in the sampling rate array.

The output from the Set Data Mode object then passes to the Get Data Panel boxes (4 and 6). These objects are where the data are actually acquired. The program waits here until the external trigger is received. Once it occurs, box 4 takes 1024 points on channel 0 and box 6 takes 1024 points on channel 2. These could be any channel, it just so happens that channels 0 and 2 were chosen as input channels. The data that are acquired are saved as a Velocity Global (116) and Voltage Global (347). The values from the force transducer are then converted into force values by dividing (341) them by the calibration coefficient of the force transducer (14), which is 5.3 mV/lb. These force values are then stored as an array in the Pressure Global box (115). The ability to multiply the whole array by a number in a single step is another nice feature of the program.
At this point, the data has all been acquired. The next step is to process it. The first calculations that are required find certain threshold values. The velocity threshold is found in the Velocity Threshold Object (141). This UserObject checks the voltages in the velocity array to determine when the signal first reappears after the flag stops occluding the signal. When the flag is blocking the signal from the emitter/detector, the voltage output is very near zero. Then, when it reappears, the voltage will jump back up to near 5 V. The point in the voltage array where it makes this jump needs to be determined to find the time that the signal was occluded. To do this, array is checked from 0 to 1023 (the arrays in DTVEE are all zero based) using the For Range object (141.16). The counter (141.14) counts the number of times that the loop executes. Then 1 is subtracted from this in box 141.13 to get the array index. When the voltage exceeds the threshold value of 50 mV, the If/Then/Else box (141.3) breaks the for range loop and the index in the array where this occurs is the final number stored as the Vthresh global(141.7) and the UserObject is exited.

The program then moves on to the Pressure Threshold Object (340), where the exact same operation is performed on the voltage array from the force transducer as was done to the velocity data. Again, a threshold value of 50 mV was chosen to exclude any spikes that might occur due to noise. This value was chosen after running several experiments and looking at the voltage data file. This is another nice feature of DTVEE. After the program is run, the arrays can be viewed by simply double clicking the mouse on the appropriate input or output pin of the objects. The data array or scalar value will appear. The array can then be scrolled through to observe the values.
The next step is to calculate the impact velocity. This is done in the Impact Velocity Object (197). In this UserObject, gravity is first set and stored as a global (197.5 and 197.4). Then the width of the velocity card is placed in box 197.3. This value needs to be changed if a new velocity card is ever made. The one used for these tests was measured to be 0.5850". Boxes 197.2 and 197.1 just contain the equations derived earlier to obtain the impact velocity. Included in both of these equations is the time step for each sample. This is just the sampling rate of the velocity data divided by the total number of points gathered, which was 1024. The impact velocity is then saved as a global variable in box 197.0.

The program next goes to the block that imports the UserFunctions that will be used from the UserFunctions Library where they are stored (367). These UserFunctions will be discussed individually when they are called.

The program then progresses to the first of its UserFunctions (289). This one is called Pzero and it finds the point where the pressure returns back to zero pounds of force. The IMPCTFXN.VEE UserFunctions program is shown in Appendix C. The Pzero function again uses a For Range command (F0.0) to check the values in the pressure array. Since the pressure will obviously return to zero only after its maximum value, the array was checked from this index on using a control input to the For Range. The index of the maximum pressure was obtained using the maxIndex(x) object (F0.9) with the pressure array as the input (F0.13). This index was also saved as a global variable in box F0.11. Then, each individual value in the pressure array is checked from the maximum index on until the pressure becomes lower than 3 lb. This again is a number gained from initial
testing. Once the pressure goes below this point, the If/Then/Else (F0.2) breaks the For Range loop and the index where this occurs is saved as a global (F0.7).

The program then progresses to the Energy Absorbed UserFunction (301). The first UserObject in this UserFunction calculates the time array for the impact event (F1.0). Time $t=0$ is when the impact first occurs. The index of the pressure array where this occurred is the threshold value. Basically, a 1024 point array of all zeros is allocated (F1.0.1), and then is filled with time values in the Set Values box (F1.0.2). The time for each sample is determined by dividing the pressure sampling rate by 1024 (the total number of samples) (F1.0.14). The complete time array is stored as a global (F1.0.4) and consists of both positive and negative values, with zero being at the impact point. Also in this UserObject, the time-per-sample saved as a global (F1.0.15) for use in future calculations.

The UserFunction then continues on by first finding the mass of the falling weight by dividing the user defined weight by gravity (F1.15). Then, the pressure array is resized by chopping off all the data points before the threshold index. An array of this new size is formed with the same pressure values that were in these same positions past the threshold index, but with different indices (F1.21). The new size of the array is stored as a global (F1.21.0), as is the index of the last entry in the new array (F1.21.13). This index is just one less than the total size. This newly sized pressure array is then saved as a new global (F1.4) and will be used in all the energy calculations.

The UserFunction then progresses to the calculations whose equations were given in Section 5.1. The first equation calculates the force array, $f(t)$ (F1.5). This array is saved
as a global (F1.6). Then, this array is divided by the mass to get an acceleration array (F1.7), which is also saved as a global (F1.8).

The program flow then goes to the Velocity Object (F1.20). This program includes an integration algorithm (F1.20.6) which uses Simpson’s 1/3 rule to integrate from the first element in the array, which has an index of 0 (F1.20.35), to each index of the new pressure array (F1.20.0 and F1.20.1). The result must then be multiplied by the time step (dt) (F1.20.33), which was found earlier, to obtain values for the velocity array (F1.20.7). The initial velocity is then added to every value in this array as per the equations (F1.20.8 and F1.20.23). The result is stored as a global (F1.20.10) and is an array the exact same size as the new pressure array. This algorithm integrates every point of the array to assure the highest accuracy. The program then goes on to calculate a displacement array (F1.19) in the exact same manner. This calculated displacement is converted from feet to inches in box F1.19.26.

The final UserObject in this UserFunction calculates the absorbed energy array (F1.11). Each value in the absorbed energy array is calculated using Equation 13 given in section 5.1. This is done in Formula box F1.11.2. The absorbed energy array is then saved as a global (F1.11.5), and the EnergyCalc UserFunction is complete.

The program flow then continues onto another UserObject (342), which resizes the absorbed energy and displacement arrays to their original size of 1024 points. It does this by setting the first indices that were not used in the calculations to zero. Two arrays of all zeros are first allocated (342.2 and 342.11). Then, the values that were calculated for these two arrays overwrite the zero arrays beginning at the pressure threshold index
(342.3 and 342.10). The resulting arrays are then stored as global variables (342.12 and 342.13). The zero baseline was used because, naturally, the absorbed energy and displacement of the specimen will be zero before the impact occurs. These arrays were resized so they could be plotted with the force versus time arrays, both of which contain 1024 points.

The program then goes on to calculates the impact energy in box 332 and stores this as a global variable (331). The impact energy is just the kinetic energy at impact, or \(1/2mv^2\). The program then puts all the pertinent information into a record (253), and saves this record under the filename that the user defined in the initial inputs (297 and 351).

The next step is to call the Critical Values UserFunction (375). This function just obtains any critical values that may be useful in the analysis of the data. The index of the maximum pressure (F2.37 and 2.38) is used to calculate the absorbed energy at the maximum load (F2.3 and F2.13), the displacement at the maximum load (F2.27 and F2.28), and the time to achieve the maximum load (F2.1 and F2.15). Other parameters obtained in this function include the maximum displacement (F2.19 and F2.20), the actual value of the maximum load (F2.5 and F2.11), the total energy absorbed at the point where the pressure first returns to zero (F2.2 and F2.14) as calculated earlier, and the energy that was absorbed after the maximum load (F2.32 and F2.33). Also, the total time of the impact was found (F2.0 and F2.16). The total time is the time value where the pressure returns to zero, since time zero was the moment of impact. Finally, this UserFunction finds the final displacement of the specimen (F2.30 and F2.29).
The program progresses to the first of the two Graphical Output UserObjects (393). This graphical output is a pop-up UserObject, and the title on this box has been removed to allow for a bigger pop-up screen. It had to be left in this open view because the UserObject cannot be reduced to the icon view without the title block. This is one of the few little quirks of this software package. The panel view of this object pops-up once either one of its two input pins are activated. These pins are called XEQ inputs, which cause the UserObject to execute. The panel view of can be seen in the plots shown at the end of Chapter 6. It includes a X-Y plot of the force and energy versus time. By deleting the title of the UserObject, this panel view and plot could be bigger. The detail view just shows the X-Y plot icon (393.1) and getting the pertinent global variables. The plot has two separate Y scales, one for the energy values (393.3) and one for the pressure values (393.4). The X-axis is the time array (393.5). Additionally, the date (393.30) is a control input that sets the title of the X-Y plot. Two other push buttons can also be seen. One is a Print button (393.34) and the other is a Continue (393.35). When the Print button is pressed, the graph is sent to the printer. If a print-out of the whole screen is desired, the Print Screen command under the File pull down menu should be used. When the Continue button is depressed, the program exits the UserObject.

After exiting, the program progresses to another pop-up graphing UserObject (392). In this UserObject, the force versus displacement are plotted (392.1). The first step in this object is to resize the pressure and energy arrays so they only contain values during the impact event (392.40). This is done by truncating the arrays to only include values between the pressure threshold value and the point where the pressure returns to zero. These arrays are saved as new globals (392.40.40 and 392.40.19). These new arrays are then called up (392.4 and 392.5) and the displacement is plotted on the X axis while the
pressure data is plotted on the Y axis. Again, the date is used as the title of the graph (392.30). Finally, the object contains push-buttons to print the graph (392.34) and to continue (392.35).

The program then progresses to another pop-up user object which outputs the critical values associated with the impact (394). Each critical value found in the Critical Values UserFunction is output, as are the specimen ID (394.0), the impact velocity (394.1), and the impact energy (394.2). A printout of this pop-up menu can be obtained by pressing the print button (394.29). The object is exited by pressing the continue button (394.28). The panel view of this object can be seen along with the graphs at the end of Chapter 6.

After exiting this pop-up UserObject, the program progresses to another pop-up UserObject (374). This one gives the operator the option of filtering the pressure data and recalculating all the other parameters. If the No button (374.1) is pressed, the program will end its run (374.7). If the Yes button (374.3) is pressed, the flow goes to a counter (374.11) then to an accumulator (374.10). On each execution, the counter will read 1. The accumulator clears only at the beginning of the program, so it will count the total number of times the yes button has been pressed. The first time it is pushed, the accumulator will read 1 and the If/Then/Else check (374.9) will pass the flow to the Exit UserObject command (374.5). The data then goes on to be filtered and replotted in the same graphing objects. This pop-up object will then appear again. Should the Yes button be depressed again, the accumulator will read 2 and the If/Then/Else loop will cause an error to be raised (374.8) which says that the data has already been filtered. The program then stops.
After the first push of the Yes button, the program proceeds to filter the original pressure values. This program uses a Polysmooth command (327). With this command, the software fits a 4th order polynomial to the data and new data points are calculated based on this curve fit. If in the future the data contains really bad noise, a MeanSmooth filter may be needed. This filter takes the average of a certain number of points around each data point and can suppress more noise.

The program then goes on to recalculate everything (acceleration, velocity, etc.) from this new filtered pressure array, and the new values are plotted in the graphing objects. These steps overwrite globals that had been set in the previous sections, but the original data has already been saved to a data file. This is the end of the IMPACT.VEE program.

Once the IMPACT.VEE program has been run, the original data is all saved as a record in a data file. The program OLDFILE.VEE enables the user to call up this data and replot it. It also allows the operator to filter the data again, as was done in the original program. This program can be seen in Appendix B.

This program begins with the panel view of a pop-up UserObject (74). This panel view can be seen in Figure 33. This is just where the user inputs the name of the data file that is to be examined along with the directory that it is in.. The program then proceeds to read the file with the given name (66) and unbuild the record (111). The parameters are all saved using the same names as the initial program. Then, the Critical Values UserFunction is called to determine the important values (117 and 82). These values were not saved in the record. Next, the same two graphing objects appear (130 and 131). The
Figure 33: Pop-up Panel View of the Initial Inputs UserObject in OLDFILE.VEE
rest of the program executes exactly as it did in IMPACT.VEE. The only difference is that gravity has to be redefined (77 and 76) because it is not saved in the record.

This ends the discussion of the DTVEE programs that were written to perform the data acquisition and energy calculations. Basically all that was done was to input the equations derived in the first part of the chapter into a format that the computer could understand. The program takes approximately two minutes to calculate the energy values and develop the plots. Also, the data files obtained from testing are all around 50 kB. All the calculations were performed with ASCII code, so this processing time and file size could possibly be reduced if another type of data type was used. Another nice feature of this software is that the program can be secured once it is completed. When this is done, all that will show up are the panel views, and the program cannot be changed. If this is done, an original copy of the program should be kept unsecured in case changes ever need to be made, because the programs cannot be unsecured. Finally, as stated earlier, these original programs can be easily refined as the need arises.
Chapter 6
Calibration Check of the Instrumented Impact Tester

6.1 Comparison with a Commercial Tester

Now that the instrumented impact tester had been completed, the final step of this project was to make sure the data produced by the instrumentation was accurate. To check this, four identical tests were performed at the same drop heights on both the Dynatup 8200 commercial instrumented impact tester and the tester that was built here. The four tests for each tester were then averaged and the two averaged test results were plotted on the same graph. All the equations used for the CIMSS tester were exactly the same as the commercial tester’s; therefore, the results should be somewhat similar.

There are two main differences between these two testers. The 8200 has a 5/8” diameter hemispherical cap, while the CIMSS has a 1/2” diameter hemispherical impact head. This will change the stress force distribution in the impact area slightly. This will also affect the force applied to the specimen a small amount. The reason for the changes is that a smaller projectile will penetrate further.
The other main difference between the two testers is the type of force transducer used. The Dyratup uses strain gages attached to a shaft to determine the force, while the tester that was built uses a more modern high impedance piezoelectric crystal force transducer. So some differences in the plots could be attributed to this. There would also be some variations in the tests due to the testing process itself. The specimens were impacted as close to their center as possible, but all the impacts will not be in the exact same location. This problem should be taken care of by averaging the four tests with each tester. As an overall goal, it was decided that most of the values should be within ten percent of each other. This would account for variations due to instrumentation and impact head size.

Since the large boundary condition specimen clamp that was built for this tester would not fit in the commercial tester, another smaller specimen clamp was used. This clamp was designed to hold 2" x 6" specimens. The last 1 1/2" of the ends were clamped, leaving an unsupported length of 3" in the center. Also, only the two ends were clamped. The bolts that secure the specimen in the specimen clamp were all torqued equally to 15 ft-lb. This was to ensure a repeatable boundary condition. The concern was that the specimens might slip in the clamp during the impact, but no slippage was observed during the tests.

For these initial tests, 0.63" thick 6061-T651 aluminum alloy specimens were tested with a 12" drop height. This aluminum alloy has a relatively high strength. Finally, to make the two testers as similar as possible, the CIMSS tester was used in the frame configuration and enough weight was added to the falling head to make it the identical weight as the commercial tester. The falling heads, then, both weighed 10.5 lb. The results from these tests are discussed in the next section.
6.2 Results and Discussion

6.2.1 Experimental Results

The averaged results from the four tests on each tester with the 6061-T651 aluminum strips can be seen in Figures 34 and 35. Figure 34 shows a comparison of the force versus time curves, while Figure 35 provides a comparison of the energy versus time results. The data from the tests were all entered into a spreadsheet. These data files were then truncated so they all began at the moment of impact. This way, the same time scale could be used for the X axis beginning with $t=0$ thus allowing comparison of the two plots with the same scale. The critical values obtained from the four tests with the Dynatup 8200 tester along with the averages can be found in Table 1, while the critical values and averages from the tests with the CIMSS tester can be found in Table 2.

6.2.1 Discussion

Examining the results from averages of the force data, it can be seen that the two plots are nearly identical. In fact, the total time of the impact appears to be exactly the same. Also, during the first portion of the impact, even the small oscillations due to the inertial loading line up and have nearly the same magnitude. The only difference between the two plots is the force level. The Dynatup tester indicates a slightly greater maximum load. This can be attributed to the larger impact head. Because the CIMSS tester has a slightly smaller head, the tup will penetrate the specimen a little more. This causes a more gradual loading of the specimen with the maximum load being reached slightly later. The value of this maximum load will also be a little lower because the specimen will undergo greater deflections; hence, the accelerations will be lower. This translates directly to lower forces. The difference is small, though, because there is only a 1/8" difference in diameter between the two impact heads.
Figure 34: Comparison Plot of the Averaged Force Versus Time Results from the Dynatup 8200 Tester and the CIMSS Tester.
Figure 35: Comparison Plot of the Averaged Energy Versus Time Results from the Dynatup 8200 Tester and the CIMSS Tester.
<table>
<thead>
<tr>
<th>Impact Velocity (ft/sec)</th>
<th>Impact Energy (ft-lb)</th>
<th>Max Load (lb)</th>
<th>Time to Maxld (msec)</th>
<th>Deflect. @ Maxld (in)</th>
<th>Energy @ Maxld (ft-lb)</th>
<th>Total Time (msec)</th>
<th>Total Energy (ft-lb)</th>
<th>Final Deflect. (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.91</td>
<td>10.20</td>
<td>1020.5</td>
<td>4.19</td>
<td>0.28</td>
<td>10.34</td>
<td>6.98</td>
<td>7.80</td>
<td>0.20</td>
</tr>
<tr>
<td>7.96</td>
<td>10.35</td>
<td>1027.6</td>
<td>4.20</td>
<td>0.28</td>
<td>10.49</td>
<td>6.98</td>
<td>7.90</td>
<td>0.20</td>
</tr>
<tr>
<td>7.95</td>
<td>10.32</td>
<td>1024.0</td>
<td>4.19</td>
<td>0.28</td>
<td>10.44</td>
<td>7.00</td>
<td>7.92</td>
<td>0.20</td>
</tr>
<tr>
<td>7.92</td>
<td>10.22</td>
<td>1014.0</td>
<td>4.28</td>
<td>0.28</td>
<td>10.40</td>
<td>7.02</td>
<td>7.87</td>
<td>0.20</td>
</tr>
</tbody>
</table>

**Averages**

| 7.94 | 10.27 | 1021.5 | 4.22 | 0.28 | 10.42 | 7.00 | 7.87 | 0.20 |

Table 1: Critical Values from the Dynatup 8200 Tester with 6061-T651 Aluminum.

<table>
<thead>
<tr>
<th>Impact Velocity (ft/sec)</th>
<th>Impact Energy (ft-lb)</th>
<th>Max Load (lb)</th>
<th>Time to Maxld (msec)</th>
<th>Deflect. @ Maxld (in)</th>
<th>Energy @ Maxld (ft-lb)</th>
<th>Total Time (msec)</th>
<th>Total Energy (ft-lb)</th>
<th>Final Deflect. (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.87</td>
<td>10.10</td>
<td>983.9</td>
<td>4.25</td>
<td>0.284</td>
<td>10.22</td>
<td>6.97</td>
<td>8.01</td>
<td>0.211</td>
</tr>
<tr>
<td>7.89</td>
<td>10.15</td>
<td>990.4</td>
<td>4.27</td>
<td>0.285</td>
<td>10.29</td>
<td>6.97</td>
<td>8.03</td>
<td>0.212</td>
</tr>
<tr>
<td>7.87</td>
<td>10.10</td>
<td>989.5</td>
<td>4.28</td>
<td>0.285</td>
<td>10.25</td>
<td>6.98</td>
<td>7.96</td>
<td>0.211</td>
</tr>
<tr>
<td>7.87</td>
<td>10.10</td>
<td>985.8</td>
<td>4.27</td>
<td>0.284</td>
<td>10.23</td>
<td>6.97</td>
<td>8.00</td>
<td>0.212</td>
</tr>
</tbody>
</table>

**Averages**

| 7.87 | 10.11 | 987.4  | 4.27 | 0.285 | 10.24 | 6.97 | 8.00 | 0.212 |

Table 2: Critical Values from the CIMSS Tester with 6061-T651 Aluminum.
Since the energy equation is derived from the force data, there will also be a slight difference in the two energy curves. It can indeed be seen that the energy absorbed up to the maximum load is higher for the Dynatup 8200. The final absorbed energy for the CIMSS tester then ends up being slightly greater. This corresponds accordingly to the amount of rebound that each tester's falling weight exhibited, which was measured. The Dynatup 8200 tup bounced off the specimen approximately 1/8 to 1/4" more than did the CIMSS tester's. This corresponds to 0.11 to 0.22 ft-lb of energy, respectively. This is close to the measured difference between testers.

Continuing on with an analysis of the critical values, it can be seen that the velocities are very close to each other as would be expected with similar drop heights. This height might not have been exactly the same on both testers. Also, different amounts of velocity could be lost to friction during the fall with the two testers. The CIMSS tester uses ball bushings to help the falling head slide down the rails, while the Dynatup tester uses brass bushings. Moving on, the impact energy uses the impact velocity in its calculation, so these values will vary accordingly.

The next parameters in the table are the maximum load, the time to maximum load, the deflection at maximum load, and the energy at maximum load, which have all already been discussed. They are all within the preset goal of ten percent agreement. In fact, all the averaged values in table are well within this goal.

The final important parameters in these tables are the total time of the impact, the total absorbed energy during the impact, and the final deflection of the specimen. As seen earlier from the comparison plots, the total times are almost identical. The total absorbed
energy and final deflection have already been discussed. The final deflections of the specimens were also measured by hand with a set of calipers to make sure they were in the right neighborhood. The hand measured deflections for the Dynatup tests averaged 0.193" while the average was 0.201" for the CIMSS tests. Considering the inconsistencies associated with hand measurements, the computer generated values appear acceptable.

As a conclusion, the results from the two tests unequivocally agree with each other when the different tup diameters are taken into account. Furthermore, the critical values are all well within the preset ten percent limit for agreement. This tester can now confidently be used for all low velocity instrumented drop weight impact testing.

One final concern associated with this tester is that the forces tend to get a little large when many add-on weights are used and the impactor is dropped from the higher heights. The force transducer has a range of 1000 pounds, but it should still produce voltages above that. According to the manufacturer, the transducer will lose some of its linearity above 5 V of output, but it should still be relatively accurate up to the 10 V sampling capabilities of the data acquisition system. The forces should really never be beyond 2000 pounds anyway. If this becomes a concern in the future, a separate force transducer with a sensitivity of 1 mV/lb (or 5000 lb for 5 V output) can be purchased.

As a final test, a 0.63" thick 6061-T651 aluminum plate was secured into the large boundary condition specimen clamp fabricated for this tester. Using all the possible add-on weights (15.408 lb drop weight), the falling head was dropped from the same 12". The results and output format of this program can be seen in Figures 36, 37, and 38. From the

135
plots, note that the time duration of the impact increases with the larger boundary condition. Also, since the deflections are greater, there is a lower maximum force that is applied than during the calibration tests. It should be mentioned that this test is meant to just be an illustration of the program output and the force level with the large plates. The aluminum plate that this test was performed on had already been impacted many times, and had many regions of localized plastic deformation. A new plate might behave differently. This completes this chapter on the calibration of the instrumented impact tester.
Figure 36: Force and Energy Versus Time Results from the CIMSS Tester with a Large Boundary Condition, a 6061-T651 Aluminum Plate, and a 12" Drop Height.
Figure 37: Force Versus Displacement Results from the CIMSS Tester with a Large Boundary Condition, a 6061-T651 Aluminum Plate, and a 12" Drop Height.
<table>
<thead>
<tr>
<th>Specimen ID</th>
<th>Impact Velocity (ft/sec)</th>
<th>Impact Energy (ft-lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>12&quot; Drop Big Plate, All Weight</td>
<td>8.128</td>
<td>15.82</td>
</tr>
<tr>
<td>Maximum Load (lb)</td>
<td>Time to Maximum Load (sec)</td>
<td>Absorbed Energy at Max Load (ft-lb)</td>
</tr>
<tr>
<td>888.1</td>
<td>8.474m</td>
<td>16.56</td>
</tr>
<tr>
<td>Total Time of Impact (sec)</td>
<td>Total Absorbed Energy (ft-lb)</td>
<td>Final Deflection (in.)</td>
</tr>
<tr>
<td>14.68m</td>
<td>9.641</td>
<td>0.3026</td>
</tr>
<tr>
<td>Deflection at Maximum Load (in.)</td>
<td>0.574</td>
<td>0.574</td>
</tr>
<tr>
<td>Energy Absorbed After Maximum Load (ft-lb)</td>
<td>-6.916</td>
<td>Continue</td>
</tr>
</tbody>
</table>

Figure 38: Critical Values Output from the CIMSS Tester with a Large Boundary Condition, a 6061-T651 Aluminum Plate, and a 12" Drop Height.
Chapter 7
Conclusions and Recommendations

In this thesis, the complete design for an instrumented drop weight impact tester was presented. This impact tester was built using an old milling machine for a base. This was done in order to obtain a relatively inexpensive yet precise X-Y table which can move a large boundary condition plate around. This base was also very rigid and stable which minimizes the amount of energy that is lost to the surrounding structure. The specimen plates needed to move so that multiple locations could be tested without changing the boundary condition after each test. This tester was built for a specialized research application involving impact detection involving intelligent material systems and artificial neural networks; however, the basic design can be easily adapted for individual research applications.

It was shown in this thesis that the amount of energy lost by the impactor during the impact can be assumed to be completely absorbed by the specimen that is being tested. The entire design process involved with building this tester was discussed. Included in
these discussions were not only the design drawings and fabrication procedures, but also the mistakes that were made. It is hoped that by including these mistakes in this thesis, they can be avoided by individuals attempting a similar process in the future.

Also described in this thesis was the specially designed specimen clamp. Both this specimen clamp and the falling crosshead were designed with testing flexibility in mind. The boundary condition size can be changed relatively easily, while the weight of the falling impactor can be varied from 3.518 lb all the way up to 15.408 lb. With this highest weight, the impact energy will be a significant 45.6 ft-lb. At the same time, with very light weights and short drop heights, the impact energy can be as low as 0.6 ft-lb. This is clearly a large range which adds greatly to the usefulness of the tester.

Further discussion included all the equations and hardware required to fully instrument the impact tester. This instrumentation process included a description of the programming done using a graphical programming data acquisition and processing software package called DTVEE.

Finally, this thesis included some initial tests performed with identical test set-ups on both a commercially available impact tester and the tester that was built during this project. The results of these comparison tests indicated that the tester that was built is more than acceptable for use as an instrumented impact tester.

The two main reasons that building an impact tester was pursued were to be able to tailor it for a specific research application and to obtain instrumented impact testing capabilities at a lower cost than if a tester had been purchased. It was estimated that a
commercially produced impact tester would have cost around $28,500 to purchase and modify. The actual cost for this tester worked out to be as follows:

<table>
<thead>
<tr>
<th>Description</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hardware Costs</td>
<td>$1,570</td>
</tr>
<tr>
<td>Electronics, Software, and A/D Board</td>
<td>$3,272</td>
</tr>
<tr>
<td>Labor Estimates</td>
<td></td>
</tr>
<tr>
<td>Shop time 20 hours @ $34/hr</td>
<td>$680</td>
</tr>
<tr>
<td>Grand Total:</td>
<td>$5,522</td>
</tr>
</tbody>
</table>

As can be seen, it was significantly less expensive to build a fully functional instrumented impact tester from scratch than to purchase and modify a commercial one. Plus, as an added benefit, much information about impact testing, design work, and data acquisition was learned during the design, fabrication, and calibration processes of the test apparatus.

There are a few additional things that could be done to this impact tester in the future to possibly improve its performance. Obviously, the impact tester computer code could be revised and improved as the need arises. This should not be too difficult, because the DTVEE program is so easy to use and learn. Another change that could be made to this tester is to add some sort of mechanical rebound catch. This catch would catch the impactor if it rebounds back off of the specimen. Right now, a small piece of rope is tied to the falling weight. This cord needs to have enough slack in it before testing begins, so it will not restrict the free-fall of the weight. After the impactor bounces off of the specimen, the chord is simply pulled taught to catch the weight. This cord seems to work very well, but something more complex may still might want to be added in the future. One additional recommendation might be to purchase another force transducer tup. This one should have a higher load range and could be used for higher force impacts.
The final improvement that could be made to this tester in the future would be to incorporate something that would accelerate the falling weight at the moment that it is released from the quick release. By accelerating the weight by some outside influence, the impact velocities can be increased. So, instead of just relying on gravity to accelerate the falling weight at a constant rate from rest, an external force would be applied. This would greatly increase the initial acceleration, which would in turn give the falling weight higher velocities. Either a pneumatic assist or a mechanical spring could be used. If the spring is used, the size and attractive pull of the electromagnet would have to be increased. If compressed air is used, a burst could be shot at the falling impact head at the moment that the quick release button is pressed. This would thus apply a force to the impact head. One impact tester offered by Dynatup, the model 8250, offers such a pneumatic assist. This increases the maximum impact velocity from 12 ft/sec all the way up to 44 ft/sec. Of course, higher forces would be encountered by the tup if this was done, and a higher range force transducer would need to be purchased. These are all the recommendations that can be made at this time. There are sure to be new modifications that will come up, but these cannot be anticipated at this time.
References


National Instruments Company, "Data Acquisition Tutorial", IEEE 488 and VXIbus Control, Data Acquisition, and Analysis, 1994 Catalog, P. 3-10.


Rogers, C. A., 1993, Research Focus Brochure for the Center for Intelligent Material Systems and Structures, Virginia Polytechnic Institute & State University, Blacksburg, VA 24061-0261.


Appendix A
Listing of Computer Program IMPACT.VEE
"Instrumented Impact Testing" : Initial Inputs Object  <Network>
"Instrumented Impact Testing" : Velocity Threshold Object  <Network>
"Instrumented Impact Testing" : Pressure Threshold Object  <Network>
"Instrumented Impact Testing" : Impact Velocity Object <Network>
"Instrumented Impact Testing" : Resizes Energy and Disp. Arrays  <Network>
"Instrumented Impact Testing" : Displ Graphing Object  <Network>
"Instrumented Impact Testing": Displ Graphing Object <Network>
Object: Resizes Displ and Press Arrays to Impact Event
"Instrumented Impact Testing": Outputs Crit. Values Object  <Network>
Appendix B

Listing of Computer Program OLDFILE.VEE
"Retrieves Old Files": Initial Inputs Object

Note Pse
Enter the file name to retrieve with the director then press continue.

Continue

Enter Filename
/craig/ut/vee/data/bigplate

Exit UserObject
Appendix C

Listing of UserFunctions Program IMPCTFXN.VEE
"Impact UserFunctions"  <Network>
"Impact UserFunctions"  <Network>
"Impact UserFunctions": PZero
Network
"Impact UserFunctions" : EnergyCalc  <Network>
"Impact UserFunctions" : EnergyCalc: Time Array Object  <Network>
"Impact UserFunctions": EnergyCalc: New Pressure Array Object <Network>
"Impact UserFunctions" : EnergyCalc: Velocity Object  <Network>
"Impact UserFunctions" : EnergyCalc: Displacement Object  <Network>
"Impact UserFunctions" : EnergyCalc: Absorbed Energy Object  <Network>
"Impact UserFunctions" : CritValues  <Network>

The index of the maximum pressure value was found in the pressure returns to zero user functions and was called Maxpt.

F2.24

F2.27
Get displ Global max(x) Set maxdisplGlobal

F2.37
maxIndex(x) Set Maxpt Global

F2.5
Get Pressure Array max(x) Max Load Global

F2.26
Get Displ Global

F2.3
Get Eabs Global

F2.0
Get Time Array

F2.1
Get Values

F2.16
Get Values

F2.14
Etotal Global

F2.33
Finds E after Max load Eafter Global

F2.10
Get Time Array

F2.16
Total Global

F2.18
Tmaxid Global
Vita

Craig Thomas Dempsey was born in Champaign, Illinois on May 26, 1970. He grew up in the country about fifteen miles west of Champaign outside of White Heath, Illinois. He graduated from Monticello High School in Monticello, Illinois in 1988, and he went on to attend the University of Illinois majoring in Mechanical Engineering. In May of 1992, he graduated with honors with a Bachelor of Science degree. He began graduate work at Virginia Polytechnic Institute and State University in August of 1992, and completed his Master of Science in Mechanical Engineering in February of 1994. He is going to be employed by Merck & Co., Inc. in West Point, Pennsylvania. He will be a process engineer in the Packaging Technology division of Merck Manufacturing.