# ABSTRACT

BOAZ, ROBERT DALE. Design of a Pneumatic Baling System for Burley and Flue-cured Tobacco. (Under the direction of Michael D. Boyette.)

Current tobacco baling technology utilizes hydraulic power to press tobacco into bales.  The high system pressures at which hydraulic systems operate pose a risk to workers.  Hydraulic systems are costly and hydraulic oil leaks contaminate baled tobacco.  A pneumatically driven, vertically oriented, multi-stroke baler was designed as an affordable alternative to current hydraulic balers.  Pneumatics was chosen due to the lower system operating pressure and absent risk of tobacco bale contamination.  The transmission of power was achieved through a reversible pneumatic gearmotor turning left and right hand acme threaded rods coupled together to form a powerscrew.  The plunger was driven by a scissor-jack design and was used to take advantage of the non-linear force response of tobacco.  The scissor-jack was driven by acme nuts traveling along the acme rod of the powerscrew.  The baler was tested with burley tobacco grown during the 2007 season at the Central Crops Research Station in Clayton, NC.  The compressive force and plunger displacement was measured for each bale produced.  These readings were used to determine the compressive force as a function of plunger travel and the compressive force as a function of bale density.  The baler required 3-4 presses to produce burley bales roughly 42 inches cubed and weighing approximately 500-600 pounds.

Design of a Pneumatic Baling System for
Burley and Flue-cured Tobacco


by
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# DEDICATION

I dedicate this thesis to my parents, Loraine Boaz Pruitt and my late father William Dewey Boaz Jr. for instilling in me my love for agriculture.

# BIOGRAPHY

Robert Dale Boaz was born October 17<sup>th</sup>, 1981 in Reidsville, NC. He grew up on a small family farm in Caswell County, NC involved in the production of flue-cured tobacco, small grains, and beef cattle. He graduated from Bartlett Yancey Senior High School in 2000 where he was heavily involved in the Bartlett Yancey FFA Chapter serving as Chapter President his senior year.

After high school he attended Rockingham Community College in Wentworth, NC participating in the college transfer program. He graduated in May 2002 with the Degree of Associate of Science. He then transferred to North Carolina State University in Raleigh, NC pursuing the degree of Biological Engineering. He graduated in May 2005 with a double concentration in Agricultural Engineering and Environmental Engineering. He immediately entered the Biological and Agricultural Engineering graduate program working under Michael D. Boyette researching burley tobacco mechanization. During his time at NC State University, he was involved in the ASABE student club and the Pack Pullers.

Bobby recently accepted a job opportunity with John Deere and will begin his career in Des Moines, Iowa July 2008.

# ACKNOWLEDGMENTS

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Last but certainly not least, to my high school agricultural teacher Cy Vernon for teaching the me importance of the 6 P's: Proper Prior Preparation Prevents Poor Performance.

# TABLE OF CONTENTS

# LIST OF TABLES

# LIST OF FIGURES

# Chapter 1

## Introduction

Tobacco growers in the United States produce predominately two main types of tobacco: burley and flue-cured.  Tobacco production in America can be traced back to the early 17th Century when John Rolfe began experimenting with the plant in Colonial Virginia (Herndon, 1957).  The increasing demand for tobacco in England ensured the stability of the American colonies.  Exporting tobacco was a major source of income for the settlers.  As the demand increased, so did production.  The colonies produced 20,000 pounds of the crop for export in 1618 and 500,000 pounds in 1627.  By 1631, cultivation of the crop had spread to Maryland.  Virginia and Maryland combined to export 1.5 million pounds in 1639 (Tso, 1990).  Tobacco helped establish the colonists' success in the New World.

Since then, tobacco production has advanced tremendously.  In the 2007 growing season, growers in the US produced 192 million pounds of burley (North Carolina Cooperative Extension Service, 2008) and 455 million pounds of flue-cured (North Carolina Cooperative Extension Service, 2007).  Much of the success of increased tobacco production, particularly with flue-cured, can be attributed to the advancement of tobacco technology including advances in genetics, chemical application, greenhouse transplants, mechanical harvesting, bulk curing, and baling.

**1.1 Market Preparation**

Market preparation describes the act of bulking cured tobacco leaves into a package for transport to the marketing facility. Producers have used various methods over the years. The earliest method involved packing leaves into large barrels called hogsheads. These round barrels made it easy for farmers to roll their tobacco to the marketing facilities (Herndon, 1957). When production made packing leaves into wooden barrels impractical, producers then started marketing their leaf in a tied form. Handfuls of oriented leaves were tied together into "hands" and were placed on sticks and taken to market. Once the tobacco arrived at the facility, the hands were taken off the sticks and placed into large baskets (Van Willigen and Eastwood, 1998). In due course, packaging methods have changed according to the preferred standard accepted by leaf buyers. Burley and flue-cured have each followed different paths for market preparation. The differing production practices for harvesting and curing dictated which market preparation tactic was most feasible for each tobacco type.

*1.1.1 Flue-cured*

Flue-cured tobacco involves a multiple-stage harvest. As the plant ripens from bottom to top, leaves are harvested individually and placed in a curing structure where heat is applied to cure the leaf. Typically, four harvests occur to the plant during a single season. After the leaves are cured, they are removed from the curing structure and prepared for market. The flue-cured industry first ventured to an untied form of market preparation in the 1960's. Approximately 200 pounds of

leaves were placed in burlap sheets measuring 8 feet square. The four corners of the burlap sheet were tied together forming a circular pile approximately 42 inches in diameter (Ellington, 2000). These piles were referred to as market lots. Moving to a non-oriented, loose-leaf packaging method proved to be very successful on the farm. Labor efficiency increased dramatically. However, handling and storing market lots proved to be difficult and inefficient. A system that benefited everyone involved in the marketing process needed to be developed.

The idea of using another method was introduced in 1978 when baling flue-cured tobacco was investigated. Baling involves placing loose leaves in a rigid chamber and compressing the leaves to form a cube. The bale is secured by wrapping wires around the bale. However, this method was not accepted at the time. Baling was reintroduced in 1996 and became a success because it increased the efficiency of the marketing, transportation and processing of flue-cured tobacco for the grower and industry. These bales were approximately 42" x 42" x 40" with a target weight of 750 pounds (Ellington, 2000). This produced an optimum bale density for storage and transportation.

### 1.1.2 Burley

Burley tobacco differs from flue-cured in that the harvest is single-stage. The entire plant is harvested and hung in a structure to air cure. The market preparation of burley includes bulking the tobacco down after the air curing is complete, stripping the leaves from the stalk, separating the leaves into different grades based on stalk position, and then preparing the leaves for market. The burley industry remained

3

with the practice of marketing leaf in hands until the 1970's when experimentation with baling began.  Morrison and Yoder realized the benefits of baling and suggested that it be considered a packaging method for burley because it offered a reduction in labor for market preparation and utilized modern technology (1973). The market practice was adopted for the 1981-1982 marketing season (Duncan and Smiley, 1987).  The standard bale size was 12" x 24" x 36" and weighed 80-90 pounds (Duncan et al., 2008).  This system remained unchanged for nearly 25 years.

Burley leaves that were baled were required to be oriented in the same direction with stems to the ends and leaf tips towards the middle.  It required more time to follow this practice of orienting the leaves than if the leaves were oriented randomly.  Because of this, the burley industry began considering the concept of marketing burley in "big" bales similar to what was used in the flue-cured industry. Big balers could receive non-oriented leaves and thus were seen as more efficient than the practice of baling in "small" bales.

A study at the University of Kentucky Woodford County Research Farm supported this idea by showing an increase in labor efficiency when big bales were used compared to small bales (Duncan et al., 2008).  Leaf buyers accepted burley in 500-600 pound bales first during the 2005-2006 marketing season.  This weight range produced big bales having the same density as that of small bales based on the standard chamber dimensions of big balers.  Because the bales are produced in the same machine as flue-cured bales, producers that grow both types of tobacco

4

now have the luxury of owning one machine to handle market preparation for both types.

## 1.2 Baler Designs

There are numerous variations on the design of baling machines.  The function, however, is basically the same for each.  The machine utilizes a plunger to compact tobacco leaves into essentially a 42 inch cube.  For burley, the target bale weight is 550 pounds and 750 pounds for flue-cured.  A corrugated fiberboard slipsheet is used to protect the tobacco on the bottom of the bale and partially cover the sides.  The bale is wrapped horizontally with 4-5 wires ranging in length from 144-150 inches.  These wires have preformed loops on the end, which simplify the process of connecting the wire ends together.

Balers differ in their speed of operation and degree of mechanization.  Both of these factors affect the cost of the machine.  The latest reported prices range from $5,000 to $35,000 (Ellington and Gooden, 2007).  Some growers have fabricated homemade devices, with very little invested into the machine, which accomplishes the goal of producing a bale that meets market specifications.

The primary difference in baler designs is in the mode of operation.  Balers can either be horizontally or vertically oriented machines that press the bale in one single stroke or require multiple strokes to form a bale.  Figure 1.1 shows a single stroke, horizontal baler and figure 1.2 shows a multiple stroke, vertical baler.  The orientation describes the direction the plunger travels to compress the leaves.

5

Single stroke balers can hold the target weight of loose leaves and form the bale in one single stroke. In order to accomplish this, these balers have a large baling chamber to accommodate this volume of loose leaves and a long plunger stroke. Multiple stroke balers are of a much smaller design than single stroke balers. The bale chambers of these machines are filled with loose leaves before the plunger makes a compression stroke. The plunger retracts and the baling chamber is filled again with loose leaves. This process of filling and compressing continues until the target weight is reached. Usually 3 or 4 strokes are needed to produce the bale. Due to cost and size, the most common type of baler found on tobacco farms now are vertically oriented, multiple stroke machines.



Figure 1.1: A single stroke, horizontal baler.

Figure 1.2: A multiple stroke, vertical baler.

### 1.2.1 Forces

The baling machine relies extensively on hydraulic systems whether it is oriented horizontally or vertically, or it requires one or multiple strokes to form the bale. Commercially produced tobacco balers typically utilize one or two hydraulic cylinders to actuate the plunger. These balers either have their own dedicated hydraulic pump and reservoir, or they take advantage of the hydraulic system on a tractor by connecting to its remote hydraulic ports. These hydraulic systems are capable of producing the large amount of force needed to compress tobacco and operate at system pressures of approximately 2,000 psi. Ellington demonstrated that a single stroke baler might have to produce as much as 46,000 pounds of force

distributed over a 42" x 42" surface to form a bale of flue-cured (26.1 lb/in$^2$) (Ellington, 2000). The more commonly used multiple stroke balers require much less force to form a bale. Burley small bales were typically formed using air cylinders or bumper-type car jacks that produced up to 1,200 pounds of force distributed over a 12" x 36" surface (2.78 lb/in$^2$) (Duncan and Smiley, 1987). The rate of compression and the moisture content of the leaves affect the amount of force needed to form the bale.

### 1.2.2 Safety

Safety is always an important topic when using machinery. Tobacco balers are no exception. Reed states that several injuries have been reported from workers operating a tobacco baler (2006). The hydraulic systems on tobacco balers pose a serious risk to the operator as well as contamination of the tobacco.

At system pressures of 2,000 psi and the high temperature of working hydraulic oil, a burst hose can be very harmful to those operating and working near the machine. Contamination of the baled tobacco with hydraulic fluid offers another avenue of marketing risk. Studies have shown animals that swallowed or inhaled certain types of hydraulic fluids experienced nerve damage, tremors, diarrhea, sweating, breathing difficulty, and sometimes weakness of the limbs or paralysis (Gannon, 2005). Exposing tobacco to hydraulic fluid could promote these symptoms in users of tobacco products, thus contaminated tobacco is not marketable and must be destroyed. Currently, there are no federal government recommendations to protect humans from the health effects caused by hydraulic fluids.

8

## 1.3 Objectives

The abundance of hydraulically operated balers and the potential hazards associated with their use begs the question of whether tobacco can be produced in a big baler through some other feasible means.  The first objective of the project was to design an affordable alternative to current hydraulic balers that is comparable in operation.  The design was based on the popular multi-stroke vertical baler.  During the design process, published information was not found that reported any expected force values produced by a multi-stroke big baler.  The second objective was to record the compression forces encountered during bale formation from load cells incorporated into the machine design.  This information was used to evaluate the machines performance and to generate force relationships as a function of average bale density and time for a multi-stroke big baler.

9

# Chapter 2

## Literature Review

Post-harvest packaging of agricultural commodities is used to preserve quality and to increase efficiency in transportation, marketing, and post-market processing. Horticultural crops are bulked down into boxes. Compaction of agricultural materials reduces volume and creates a defined shape for more efficient handling. Forage crops are often compacted into square or round bales. Seed cotton is harvested and compacted into a module that can weight as much as 25,000 pounds. The seed cotton is then ginned and the lint is baled into 500 pound bales. One 21" x 33" x 55" bale of cotton is equivalent to over 1.5 acres of field cotton (National Cotton Council of America, 2008). Crops harvested in a fibrous or leafy form yield good results when compacted into a defined shape.

### 2.1 Compressing Tobacco

Tobacco is baled by loading loose-leaf into a baling chamber. The leaves are compacted by a plunger into a 42 inch cube. The actuation of the plunger is most often performed by hydraulic cylinders on commercially-produced balers. Hydraulic cylinders are used because they can provide large amounts of force and can have a long stroke. Longer strokes allow more leaves to be loaded into the baling chamber before the need to actuate the plunger. The plunger does not have to be actuated

by hydraulics. Ellington and Gooden state that any type of baling equipment will suffice as long as the goal of meeting the desired bale specifications is met (2007). While hydraulics tend to be very popular, any means of actuating the plunger can be used as long as the end result is a 42 inch cube of tobacco of the target density.

Biological materials exhibit a different response when subjected to a force, compared to most other materials. The physical properties of steel and other common materials exhibit an initial linear region in their stress-strain curve. Biological materials, like tobacco, exhibit a non-linear force response initially. The required force to press tobacco increases as the deformation rate increases. This phenomenon is represented in figure 2.1. It is suggested this increase is due to the hydrostatic pressure build up between the cellulose fibers. As the volume between the fibers decreases, the pressure increases (Ellington, 2000). As this pressure increases, more force is required to press the leaves. Overtime, this pressure releases and the compressed tobacco experiences a stress relaxation.

Mechanical properties of biological materials are also affected by other physical characteristics. The moisture content of tobacco is considered the most important factor affecting its compressibility. Baling tobacco at lower moisture contents requires more force as demonstrated by Ellington (2000). Understanding how a material behaves under load can aid in the design of machines that handle the product.
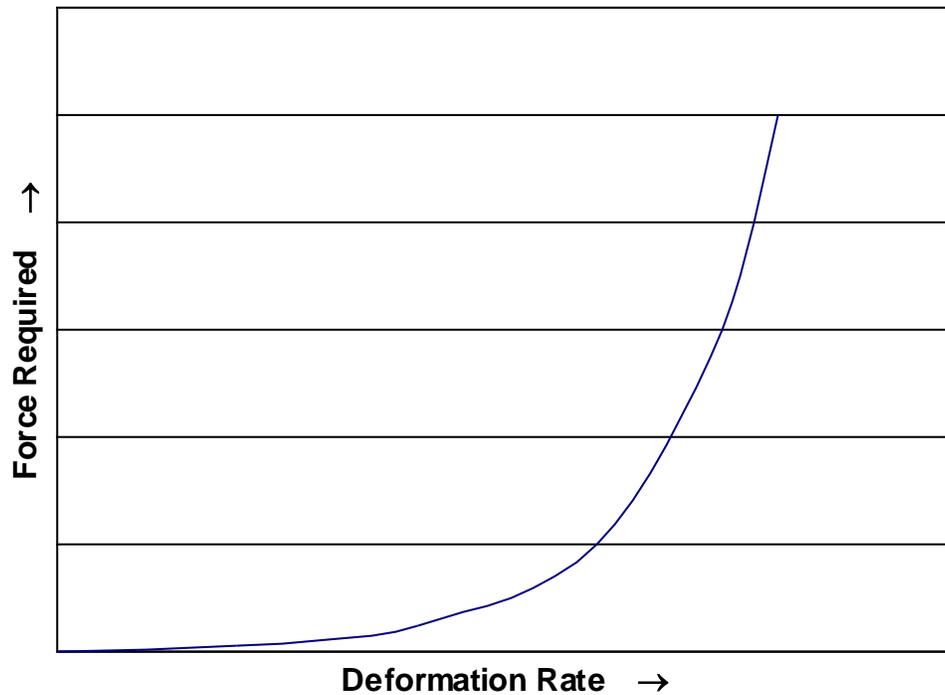
Figure 2.1:  Typical non-linear force response for baling tobacco.

## 2.2 Source of Power

Actuation of the plunger can originate from fluid power sources, like

hydraulics and pneumatics, or from an electromechanical device.  Both types can

utilize linear actuators (cylinders) or rotary (motor) devices.  Each power source has

positives and negatives associated with its usage.  Cost, reliability and safety are

three areas considered by the author to be most significant for implementation on a

tobacco baler.

Fluid power actuators are relatively small, even for applications requiring

great forces.  Electric motors have to be sized for the maximum load, whereas fluid

power sources only have to be sized for the average load (Nachtwey, 2008b).  Fluid

power sources are smaller and quieter and offer the ability to build machines at

considerable savings compared to machines involving purely electrical or

mechanical motion (Nachtwey, 2008a).  Using fluid power sources would result in a

cost savings for the overall machine.

Nachtwey discourages the use of an electric actuator for any press type

application (2008b).  Pressing requires a constant torque and could cause the motor

to overheat, because holding torque requires continuous power.  An electric motor

draws a large amount of current to maintain torque even while stopped and most

motors will overheat and fail under these conditions (Nachtwey, 2008a).  In press

applications where a constant holding torque must be applied, fluid power has the

advantage.  When fluid power systems are not moving, no energy is used.

One of the potential hazards associated with current hydraulic balers is the

risk of contaminating the bale with hydraulic fluid.  Hydraulic cylinders and hoses

placed over or near the baling chamber offer a chance for leaks or burst hoses to

lead to tobacco contamination.  The quality of the product in the machine and the

safety of people operating the machine are of the utmost importance.

Turner and Post state that air motors are a safe and reliable way to generate

torque and rotational motion (2007).  Pneumatic power has an advantage in

applications where the presence of hydraulic oil could cause problems, like food

processing machines (Nachtwey, 2008a).  With pneumatic devices, there is no worry

about leaks and oil contamination as there is with hydraulic components.  Pneumatic

systems are also typically less expensive to build than hydraulic systems.

Compressed air products also provide several benefits over electrical

devices.  First of all, air motors have a higher power density, which means more

power is produced in an air motor compared to an electric motor of equal size.

Electric motors must be oversized to dissipate heat whereas air motors generate

little to no heat.  Secondly, when subjected to a torque overload, air motors will use

no power and simply stall.  Under the same conditions, an electric motor will be

prone to burning out.  Finally, maintenance of electric motors is typically more costly

since they are usually sent to shops that specialize in electric motor repair.  Air

motors can be repaired by any technician experienced in repairing tools, pumps,

compressors, gear boxes, or other rotating equipment (Turner and Post, 2007).

Even though the initial cost of an electric motor may be lower, the overall cost of air

motors is typically less.  Downtime and turnaround time is lower for an air motor and

these factors affect the overall cost of operation.


**2.3 Scissor Press**

The non-linear force response of tobacco presents a unique design

opportunity for a tobacco press application.  As discussed in section 2.1, more force

is needed to press tobacco as the deformation rate increases.  Also, more force is

required as the density of the bale increases during a press.  In other words, the

system must produce more force when the plunger is at the end of the stroke than

when it is beginning the stroke.  A plunger press based on a scissor-jack design was

presented as a way to take advantage of the natural tendencies of tobacco

undergoing compression.

A scissor-jack, or scissor lift, is a type of platform which usually only does

work in one plane.  The scissor lift shown in figure 2.2 lifts against the basket weight

in the vertical direction.  If oriented vertically, the scissor lift would move up and

down in the vertical plane.  Consequently, if oriented horizontally, the lift would move

side to side in the horizontal plane. The mechanism to achieve this is the use of

linked, folding supports in a criss-cross 'X' pattern that pivot where the legs cross.

Most scissor lifts are oriented vertically and are used to create an aerial work

platform, which raises workers high into the air.  A compound scissor lift involves

multiple stacked 'X' patterns which greatly increases the stroke of the lift in a very

small footprint.
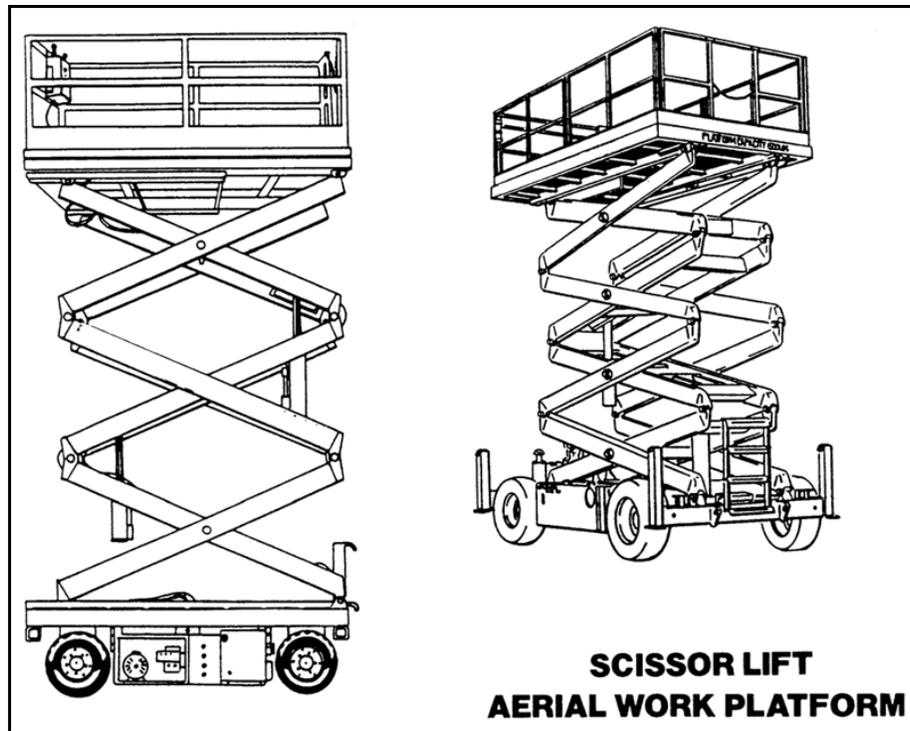
Figure 2.2: Typical compound scissor lift

(www.aerialspecialists.com).

A scissor-jack has two main advantages for use in a tobacco baler.  The first

is that as the actuator moves at a constant speed, the platform velocity decreases

(Figure 2.3).  This is because the scissor arm ends located at the platform

attachment make an arc as they travel the full stroke of the scissor lift.  At the

beginning of the stroke, the arm ends are traveling in the part of the arc that has an

infinite slope.  At the end of the stroke, the arm ends are in the part of the arc with a

zero slope.  If used to press tobacco, this decrease in speed would give the tobacco

more time to exhibit its natural stress relaxation during compression.  The second

16

benefit is the mechanical advantage of the scissor-jack increases as the end of the

stroke is reached (Figure 2.4).  This would provide more force to press tobacco to

the target density when more force is required.
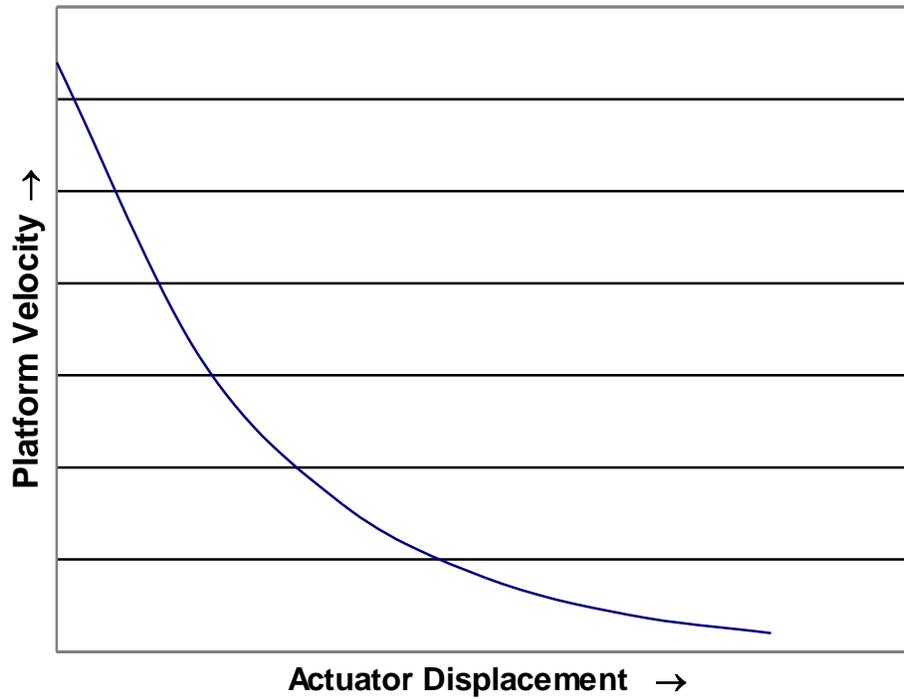


Figure 2.3:  Typical plunger velocity response as actuator displaces at constant
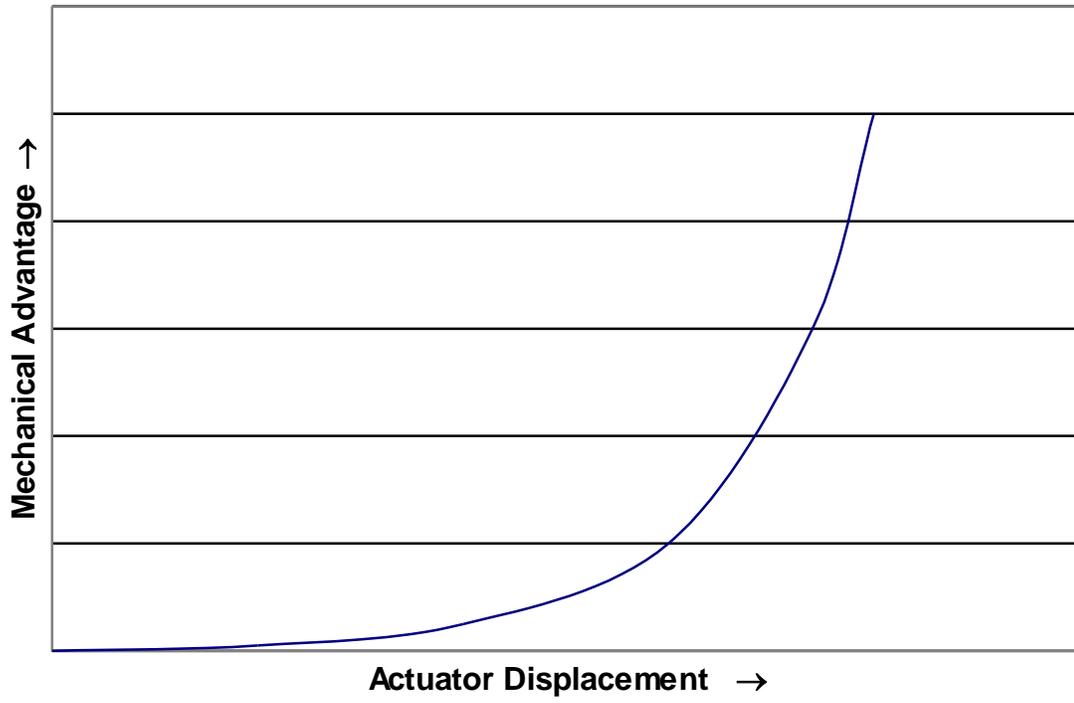
velocity.

Figure 2.4: Typical mechanical advantage response as actuator displaces at

constant velocity.

# Chapter 3

## Baler Design and Construction

A tobacco baler was developed to satisfy the project objective of producing an affordable alternative to current hydraulic balers capable of baling burley and flue-cured leaves. Pneumatics was chosen as the means to actuate the plunger via a scissor press based on the absent risk of tobacco contamination, safety, reliability and cost. The machine was of the vertically oriented, multiple stroke baler type. An important aspect of the project was to keep the machine behavior and operation similar to other balers of this type due to their popularity. This would allow seamless implementation of the machine into production operations already using multiple stroke, vertical balers. The main design criteria for this project focused on retaining the positive aspects of current multiple stroke, vertical balers and offering improvements.

### 3.1 Design Criteria

In order to keep the operation similar to current machines, the baler would have to perform a press cycle in a certain amount of time. A press cycle is described as the plunger starting in the home position, extending the length of the stroke for compression, and then returning to the home position to allow for loading of the baling chamber with leaves. A press cycle of one minute was deemed

desirable.  The amount of time required to complete a press depends on the amount of force the machine is capable of providing.  Due to the nature of compressing tobacco, a small amount of force would take a longer press time.  A large amount of force could press a bale rather quickly.  Due to leaf damage that occurs during rapid compaction, an intermediate amount of force and press time is desired (Duncan and Smiley, 1987).  The type of tobacco being baled also affects the amount of force required.  Flue-cured is baled at a lower moisture content and to a higher density.  Both of these factors result in more force being required to bale flue-cured as opposed to burley.  The machine design was based on baling an average bale of flue-cured.  The thought was that if it could bale flue-cured, it could also bale burley.  Based on previous baling work and discussions with advisors, it was determined that a force production of 10,000 pounds (5.67 lb/in$^2$) would be the target.

Another important criterion was to keep the cost low.  The machine was to be constructed at a price below $10,000.  Many commercially produced machines can range as high as $55,000 depending on the options.  Meeting this challenge meant taking manufacturability and assembly into account during the design phase.  Several guidelines were followed during the design process to increase efficiency in these areas.  The design was simple in order to keep the overall number of parts at a minimum.  The bill of materials used common parts and standard steel shapes of as few different materials as possible.  Parts were designed for easy fabrication to avoid complex tooling.  The design also called for modular products with several subassemblies being fabricated and fitted together during final assembly.  Assembly

proceeded vertically with parts, components and subassemblies added on top of the existing structure allowing gravity to be helpful, not a hindrance.  Following these guidelines helped to ensure that costs were kept low and that fabrication and assembly proceeded smoothly.  The required maintenance and serviceability of the machine was also of great importance.  The machine should be easily serviceable and simple to maintain and operate.

## 3.2 Component Design, Fabrication and Assembly

The design process has often been referred to as an iterative process.  That statement held true for this project as well.  Every phase of the machine development: design, fabrication, and assembly, was connected to each other through an interwoven web of detail.  The dimensions and specifications of one component often dictated another.  Many times an earlier design or drawing was modified in order to be compliant with a component sized later. For many parts, there were several revisions of a drawing before a design was finalized.  The design process started with the specifications for bale size and the bio-rhealogical characteristics of cured tobacco.  In order to understand the concept and component creations for the machine, it is described according to the various subassemblies that make up the entire system.

### 3.2.1 Chamber

The baling chamber was designed with leaf loading in mind.  The height of the chamber gave it a nearly 2:1 uncompressed to compressed volume ratio.  This

allowed much more tobacco to fill the chamber before requiring a press.  The most

noticeable difference between this baler and most multi-stroke machines is the

overall height of this system.  Most multi-stroke, vertical balers are shorter which

allows them to be top loaded.  Due to the plunger operation of this machine, it

cannot be top loaded.  To overcome this, a dual door system was made for flexibility

in front loading leaves.  Once the lower half of the chamber was full, the bottom door

was closed while the top door remained opened to allow leaves to be loaded

manually more easily.  Also, the rear wall height stops short of the sidewalls

providing an opening between the retracted plunger and the top of the rear wall.

This feature is visible in figure 3.1 and was designed into the chamber to

accommodate rear loading through a conveyor system if deemed desirable.
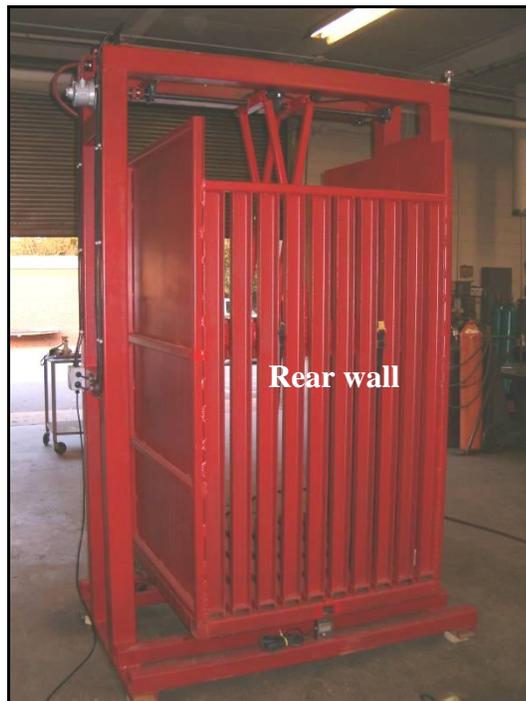


Figure 3.1: Rear baler view showing conveyor leaf loading feature.

The floor and rear wall of the chamber was comprised of c-channel members offset by a 1-1/4 inch gap to provide slots for the bale wires. The orientation was such that the web of the channel would contact the bale. Figure 3.2 illustrates these chamber features. The floor channel was welded to 3 inch tubing that formed a square base for the chamber.



Figure 3.2: Baling chamber floor and rear wall C-channel.

The channel was standard C3x5 structural steel. It had a 3 inch web, 1-1/2 inch flanges, 1/4 inch thick and weighed 5 pounds per linear foot. Ten channel members were spaced to make the chamber width 41-1/4 inches and cut to make the chamber length 41-1/4 inches also. These dimensions were set to meet bale

specifications of an approximate 42 inch cube.  Slight lateral expansion occurs upon

the bale exiting the chamber.  It expands much more in the direction of compression.

Since this is the case, the chamber height was sized so that the compressed bale

was 38 inches tall.  This also facilitated the hooking of the bale wires.  When fully

retracted, the distance between the plunger and chamber floor was 68 inches.  The

doors and sidewalls were made of 12 gauge sheet steel and 1-1/2 inch tubing giving

the chamber an overall height of 78 inches.  The weight of the chamber was 1,000

pounds.  A detailed drawing of the chamber can be found in Appendix A, drawing 1.

### 3.2.2 Scissor Press

The main component requiring the most careful design was the scissor press.

This assembly was responsible for actuating the plunger the entire length of the

stroke and transmitting a vertical force of up to 10,000 pounds.  The press was a

double-scissor design as shown in figure 3.3.  The two scissors were connected to

each other at the top of each scissor arm by two traveling collars as seen in figure

3.4.  The two collars traveled along an acme threaded rod 1 inch in diameter with 6

threads per inch.  The acme rod was composed of two separate rods, one a left-

hand thread, one a right-hand thread coupled together by a one piece clamp-on

shaft coupler.  The shaft coupler, 3 inches in length, forced precise alignment

between the two halves of the acme rod and could transmit up to 4,000 inch-pounds

of torque once the clamping screws were tightened.  More details of the acme rod

assembly can be found in drawing 2 of Appendix A.

Each scissor was composed of two equal length scissor arms pinned together at the midpoint by a 5/8 inch hex head cap screw. The bolt incorporated large washers on either side of each scissor arm to ensure proper alignment, spacing and reduced friction. Since the scissor arms had to be able to rotate relative to each other, the cap screw could not be completely tightened and therefore a nylon insert lock nut was used to ensure the assembly would not become loose.



Figure 3.3: Double scissor press
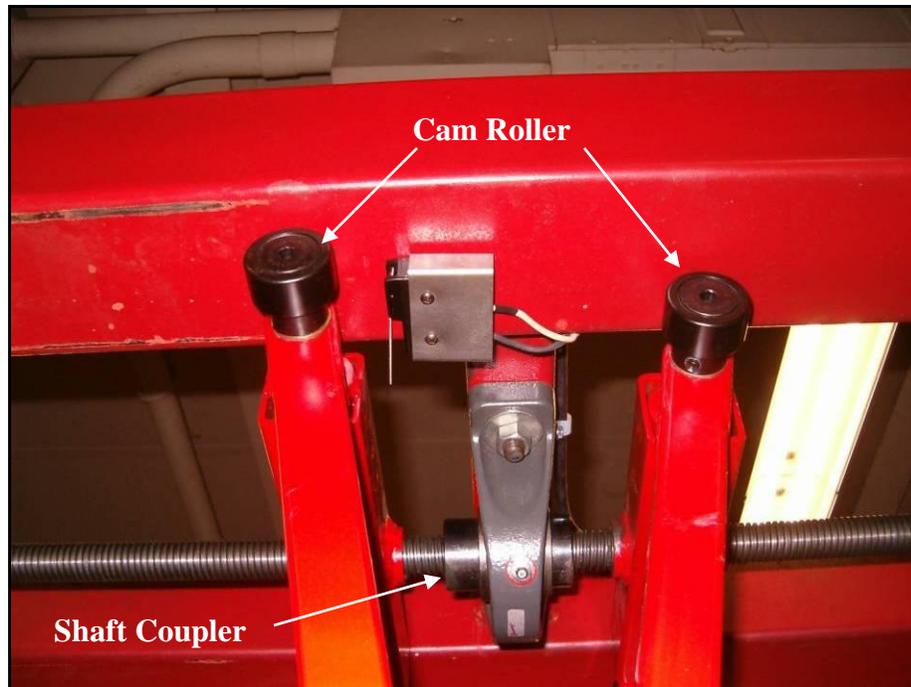
Figure 3.4: Traveling collars with integrated acme threaded nuts.

The weight of the press was supported by three pillow-block bearings holding the acme rod in place. The acme rod was supported on each end by a 1 inch bore eccentric lock bearing shown in figure 3.5 and 3.6. The clamp-on shaft coupler was located at the midpoint of the acme rod. The shaft coupler was inserted through a 1-3/4 inch pillow block bearing that supported the press at its midpoint as shown in figure 3.4. The three pillow block bearings were mounted to the baler frame and along with the rollers on the end of the traveling collars, transmitted the vertical reaction force felt during pressing to the frame.

Figure 3.5: Eccentric lock pillow block bearing.
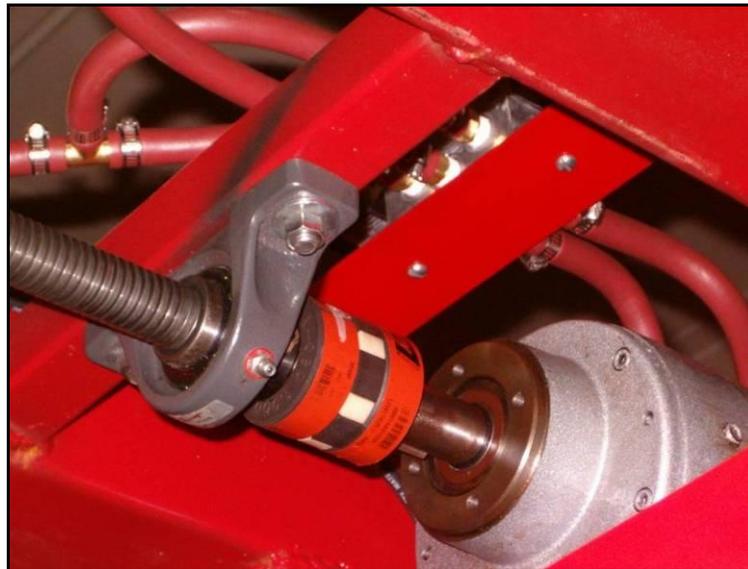


Figure 3.6: Eccentric lock pillow block bearing and lovejoy® shaft coupler.

### 3.2.2.1 Plunger Actuation

The traveling collars had integrated acme nuts of the same thread as the acme rod on which they traveled.  Details of the two traveling collars can be found in drawing 3 of Appendix A.  Depending on the rotation of the acme rod, the collars either traveled toward or away from each other.  This feature is what caused the press to move up and down.  The connection of the scissor arms to the traveling collars was a pivot point that allowed the arms to rotate as the collars traveled horizontally.  Each scissor arm had an integrated, maintenance free, oil-impregnated, bronze sleeve bearing that reduced friction as the scissor arms rotated on the pivot.  Brass washers were installed on either side of the scissor arms to reduce friction and wear on the steel members.   At the ends of the traveling collars were 1-5/8 inch cam rollers, which rode under the baler frame.  These can also be seen in figure 3.4.  This prevented the entire scissor press from trying to rotate as the acme rod was rotated.  Another 1-5/8 inch roller was attached to the bottom of each scissor arm.  These rollers traveled in box tracks attached to the free-floating plunger.  The roller tracks were made from overhead door track and allowed the rollers to press down on the plunger during pressing and pick up on the plunger during retraction.  Each roller incorporated needle bearings that allowed it to transmit up to 4,840 pounds while rolling.  The plunger was constricted in lateral movement by the sides of the chamber and the only degree of freedom was in the vertical direction.  As the scissor press was actuated, the plunger would move vertically in the baling chamber.

28

The scissor press was designed to maximize the plunger stroke in order to maximize the uncompressed chamber volume. This feature reduced the overall number of presses required to reach target bale weight. The plunger stroke was limited by the length of the scissor arms. Because the unit was situated inside the bale chamber, the scissor arm length was limited to the chamber cross section dimensions. As discussed in section 3.2.1, the chamber cross section was a 41-1/4 inch square. The scissor arms were 40 inches in length from the centers of the rollers located at each end. The geometry and limits of the scissor press resulted in a total plunger stroke of 30 inches as shown in figure 3.7 and 3.8. In order to achieve full stroke, each traveling collar translated 16 inches horizontally. The range of rotation between the scissor arms and the horizontal plane was from 13 degrees at full retraction to 80 degrees at full press. Due to the geometry of the scissor press, a constant linear velocity of the traveling collars results in a constant angular velocity of the scissor arms but a vertical deceleration of the plunger. Figure 3.9 describes the relationship between scissor arm rotation and plunger distance. The decreasing slope of the curve demonstrates a deceleration of the plunger. The magnitude of this deceleration is more pronounced at the end of the plunger stroke.

Figure 3.7: Plunger fully extended.



Figure 3.8: Plunger fully retracted.
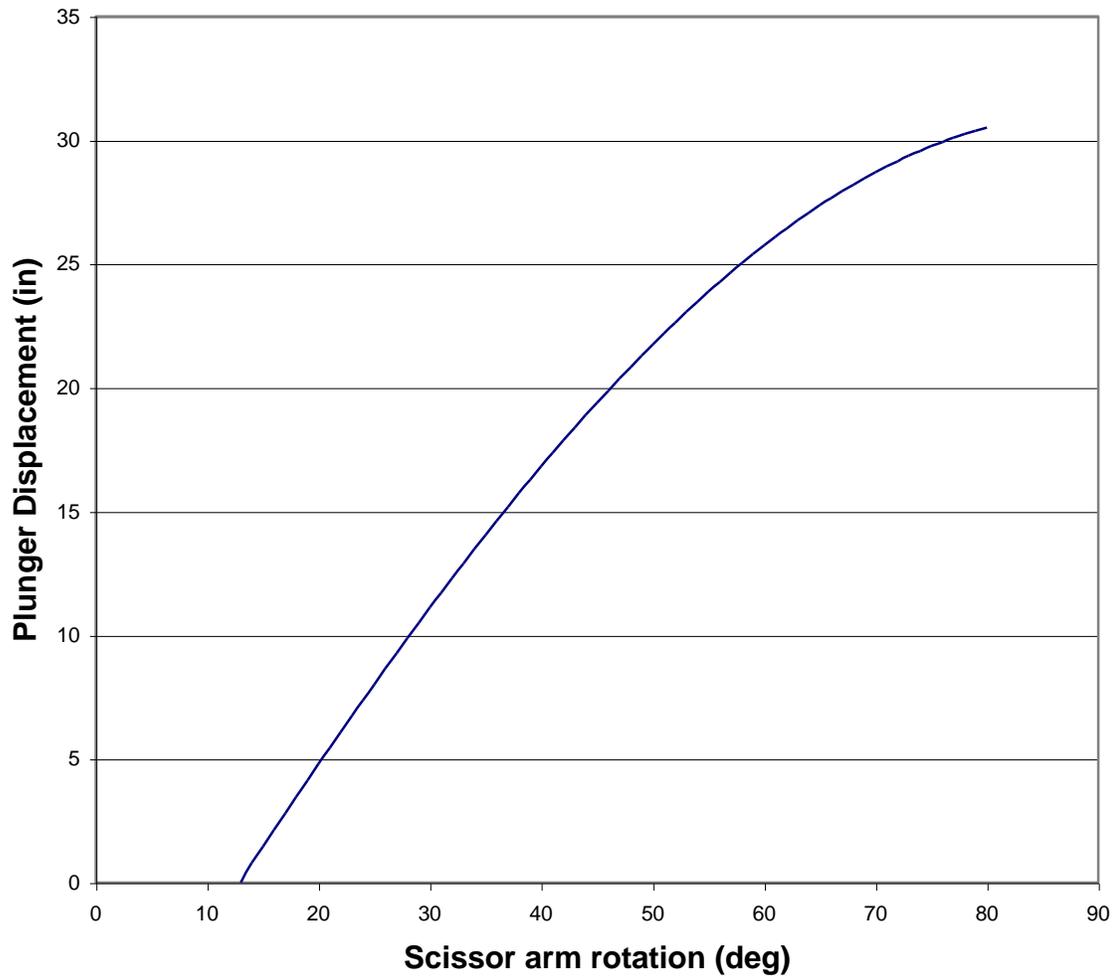
Figure 3.9: Plunger distance as a function of scissor arm rotation from 13 to 80

degrees.

### *3.2.2.2 Horizontal Force Calculations*

A scissor press is unique in that it translates a horizontal force into a vertical

force. Equation 3.1 is used to calculate the horizontal force produced by the

actuator to lift a weight as shown in figure 3.10 (Engineers Edge, 2006)

$$(3.1) \qquad F = \frac{W + \frac{Wa}{2}}{\tan(\Phi)}$$

where F is the horizontal force produced by the actuator, W is the combined weights of the payload and the load platform, Wa is the weight of the two scissor-arms, and Φ is the angle between the scissor arms and the horizontal.

For the bale press application shown in figure 3.3, the horizontal force equation changes slightly. Equation 3.2 describes the horizontal force produced by the acme rod during bale formation

$$(3.2) \qquad F = \frac{Ft - Wp - Wa}{\tan(\Phi)}$$

where F is the horizontal force produced by the acme rod, Ft is the vertical resistive force from the tobacco, Wp is the weight of the plunger, Wa is the weight of one scissor, and Φ is the angle between the scissor arms and the horizontal.
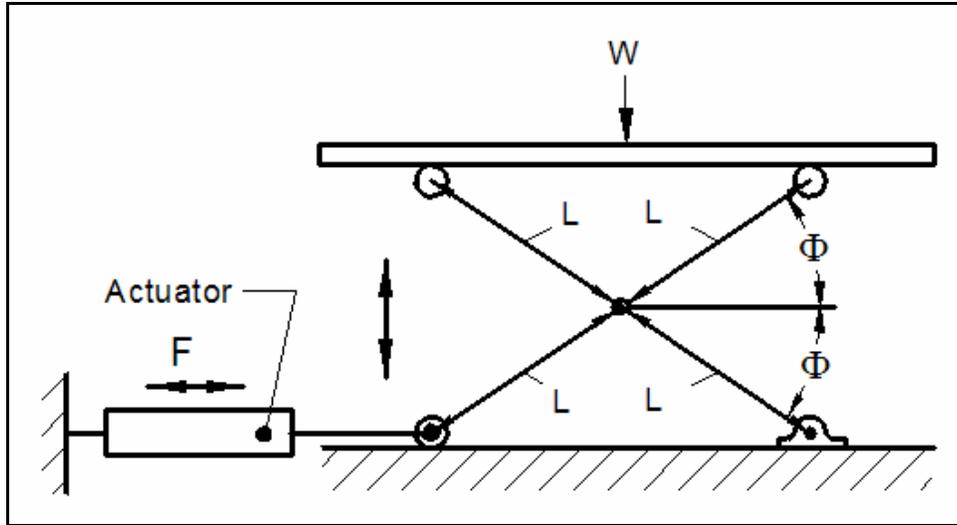
Figure 3.10: Scissor lift with straight, equal-length arms.

(Engineers Edge, 2006).

As mentioned in Section 3.2.2.1, full plunger stroke occurs when the scissor arms are 10 degrees from vertical ($\Phi$). Full stroke is where the scissor press would experience the maximum resistive force from the tobacco. Substituting in a value of 250 pounds for the plunger, 13 pounds for the scissor, and 10,000 pounds for the vertical tobacco resistive force into equation 3.2 gave an axial force of 1,717 pounds. This demonstrates a nearly 6:1 mechanical advantage produced by the scissor press.

Figure 3.11 demonstrates how the scissor press would behave during plunger retraction. Equation 3.3 was used to determine that it would take an axial force of 1,139 pounds to fully retract the plunger for leaf loading of the chamber. In order to

meet design criteria, the acme rod must be able to produce 1,717 pounds of horizontal force.

$$(3.3) \qquad F = \frac{Wp + Wa}{\tan(\Phi)}$$
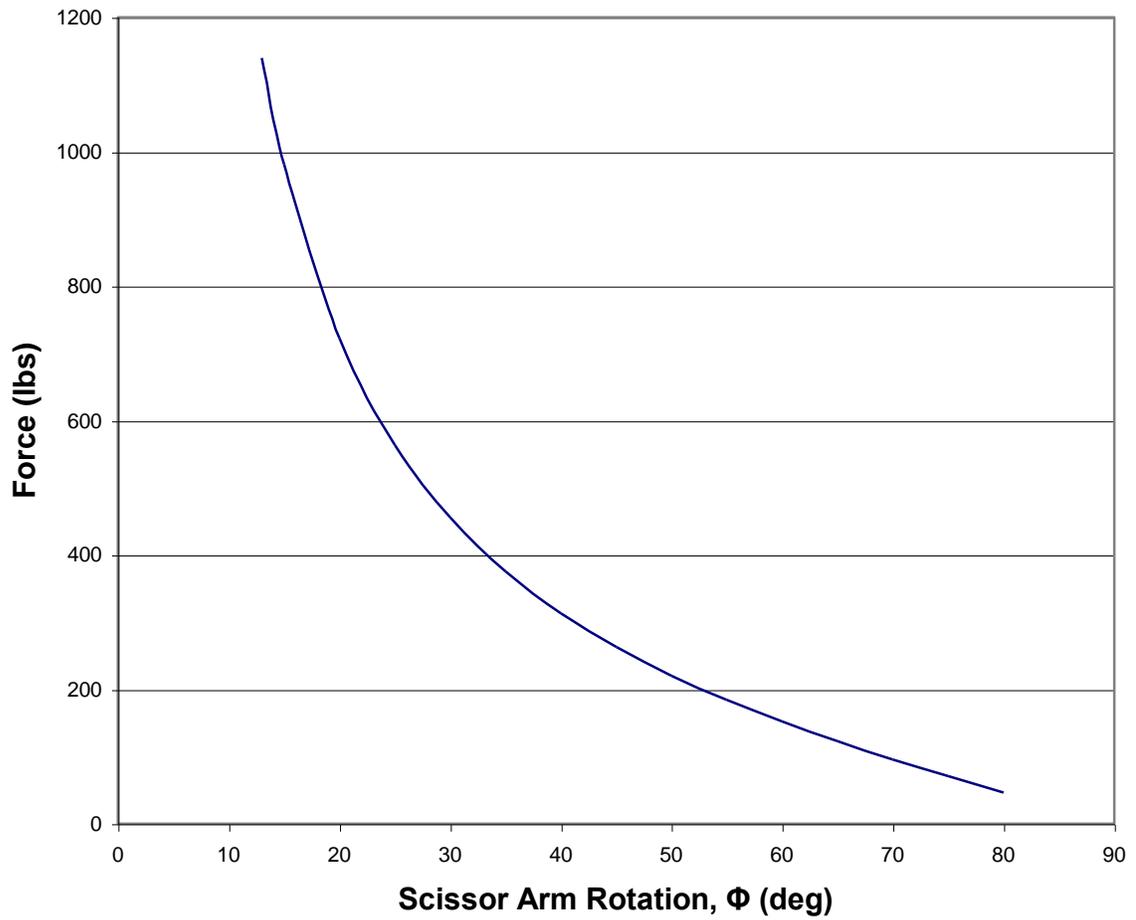
Figure 3.11: Axial force required to retract plunger.

### 3.2.2.3 Torque Calculations

The rotation of the acme rod, the driving force behind the scissor press, is a basic example of a power screw. A power screw converts rotation to linear motion and is often used when heavy loads are moved or when precision positioning is needed. Equation 3.4 is used to calculate the torque that an acme threaded power screw requires to move a given load (Norton, 2000).

$$(3.4) \qquad T = \frac{F*dp}{2} * \frac{\mu*\pi*dp + L*\cos(\alpha)}{\pi*dp*\cos(\alpha) - \mu*L} + \mu*F*\frac{dc}{2}$$

where T is the torque required, F is the axial load, dp is the pitch diameter, μ is the coefficient of friction, L is the thread lead, α is the thread radial angle, and dc is the mean collar diameter. For a 1 inch acme rod with 6 threads per inch, the pitch diameter is 0.9 inches and the lead is 0.167. The radial angle is 14.5 degrees for acme threads. The mean collar diameter was 1.47 inches for the acme nuts used in the traveling collars. Table 3.1 lists the torque values required to move a 1,717 pound axial load at various coefficients of friction using equation 3.4. The coefficient of sliding friction in threaded steel pairs using no lubrication is suggested to be between 0.15 and 0.25 (Bazoune, 2007). Using the upper limit, it can be seen from table 3.1 that 564 inch-pounds of torque will need to be applied to the acme rod to produce a horizontal force of 1,717 pounds.

Table 3.1 Torque requirement at different friction coefficients using equation 3.4.

| Torque, T (in-lbs) | Coefficient of sliding friction, μ |
|:---:|:---:|
| 356.2 | 0.15 |
| 460.2 | 0.20 |
| 564.4 | 0.25 |

### 3.2.2.4 Self-Locking Power Screw

Due to the arrangement of the scissor press as shown in figure 3.8, it would also be advantageous for the power screw to be self-locking.  If a power screw were self-locking, no magnitude of axial force would allow the screw to rotate.  Without this feature, the gearmotor would be responsible for holding the plunger in place during the leaf loading process when the plunger is retracted to the top of the stroke and during bale wire tying when the bale reaction forces are pushing against the plunger at full stroke.  Relying on the gearmotor to do this would quickly deplete the supply of air in the air tank and cause the air compressor to operate at a higher duty cycle increasing wear and tear and causing parts to fail quicker.  It would also result in an input of work into the system with no work output from the system.  Equation 3.5 is used to determine if a power screw will self-lock (Norton, 2000)

$$(3.5) \qquad \mu \geq \frac{L}{\pi * dp} * \cos(\alpha)$$

According to equation 3.4, the power screw driving the scissor press will self-lock as long as the coefficient of friction between the collar and acme rod is greater than 0.06.  In this case, the power screw is self-locking with a 0.25 coefficient of friction.

### 3.2.2.5 Pneumatic Gearmotor

An acme rod coupled to a pneumatic gearmotor via a flexible spider shaft coupling hub drove the scissor press.  The coupling hub, more commonly known as a lovejoy®, allows two rotating shafts to be coupled together and compensates for minor shaft misalignment.  This device is pictured in figure 3.6.  The motor output shaft was a 1 inch diameter, keyed shaft.  The acme rod was also a 1 inch shaft but a keyway had to be cut into the threads to accommodate the lovejoy.  The gearmotor chosen to power the scissor press was model number 4AM-RV-75-GR25 manufactured by GAST®, (Benton Harbor, MI).  As shown in figure 3.12, the motor has a parallel output shaft and can be face or foot mounted.  It has a rotary, 4-vane design with reversible output shaft rotation and can be operated horizontally or vertically with the output shaft facing down.  This model incorporates a 15:1 spur gear reducer to produce higher torque at lower speeds.  Lubrication to the spur gear reducer was provided by an oil-bath of SAE Grade 20 turbine quality oil.  According to specifications, at the maximum working pressure of 80 psi and 200 rpm, the motor consumed 60 CFM of air and produced 400 inch-pounds of torque and 1-1/4 horsepower.

Figure 3.12: GAST® pneumatic gearmotor with parallel output shaft

(www.gastmfg.com).

As seen in figure 3.13, the torque production ramps up as the load on the motor is increased until it peaks at 630 inch-pounds at 20 rpm and 80 psi.  Using TK Solver®, equation 3.4 can be used to calculate the maximum axial force the gearmotor is capable of producing.  An input of 630 inch-pounds of torque on the acme rod would cause it to produce an axial force of 1,916 pounds.  The TK Solver sheets used for this calculation can be viewed in Appendix B.
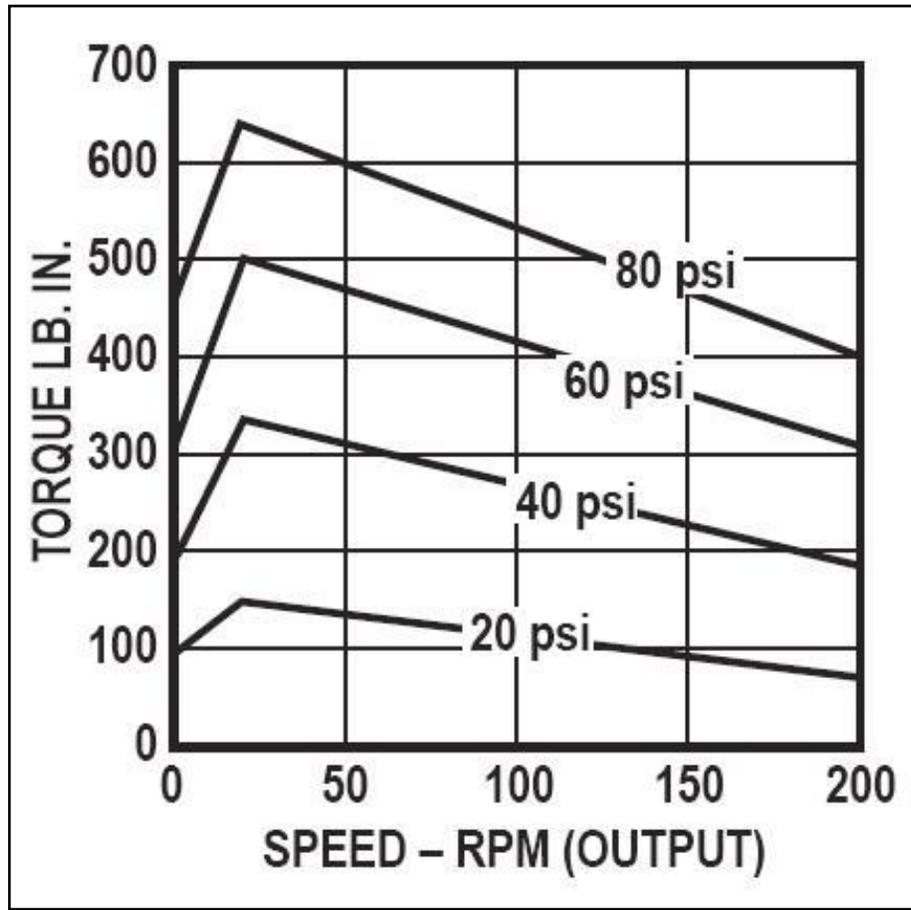
Figure 3.13: Torque specification chart for GAST® pneumatic gearmotor

(www.gastmfg.com).

*3.2.2.6 Press Cycle Calculation*

The linear speed of the traveling collars can be determined using equation 3.6

$$LS = \frac{RPM}{TPI}$$

(3.6)

where LS is the linear speed of the collars in inches per minute, RPM is the rpm of the acme rod and TPI is the number of threads per inch on the acme rod. The time in minutes can be determined by dividing the distance the collars traveled by the linear speed of the collars. In order to determine the shaft rpm, something would need to be known about the load on the plunger in order to determine the axial force and eventually the required torque. Knowing the load experienced during pressing and retraction would allow equation 3.2 to be used to determine the axial force produced. Once the axial force is known, equation 3.4 is used to determine the torque. The torque vs. rpm curve in figure 3.13 would allow the shaft rpm to be determined.

The process is simple for retraction. The load is simply a function of the weight of the plunger and scissor itself. Figure 3.11 gives the axial forces required to retract the plunger. Substituting these force values into equation 3.4 shows that at no point during retraction would the shaft rpm drop below 200 rpm since no torque above 400 inch-pounds would be required to fully retract the plunger. With this being known, equation 3.6 is used to show that the linear speed of the collars during retraction is 33.3 inches per minute resulting in a little over 1/2 minute for the collars to travel 16 inches to fully retract the plunger.

The problem is more difficult to solve when the scissor press is pressing. The press is not experiencing a constant reaction force during bale formation; rather, the resistance from the tobacco varies quite considerably. The compressive force required to bale tobacco depends on many factors. The moisture content, type of

40

tobacco and density are some of the most important factors.  To simplify the

problem, it was assumed that the resistive force would increase exponentially from 0

to 10,000 pounds as the plunger stroked from 0 to 30 inches.  Microsoft Excel was

used to generate the anticipated press cycle scenario.  Using the exponential force

scenario, the press time would take slightly more time than retraction, resulting in a

total press cycle time of 65 seconds.  While this situation did come very close to

meeting design criteria, the actual performance was not known until testing.

### 3.2.3 Frame

The frame was a unique feature to this baler.  The frame encompassed the

chamber with three load cells placed between the chamber and the frame bottom

section.  The frame top section contained the bearings that held the power screw in

place.  Due to this setup, the weight of the chamber plus its contents as well as the

force from the scissor-press registered as compression forces on the load cells.

The frame bottom section was made of 3 inch tubing welded at the ends onto

rectangular 2" x 4" tubing to allow forklift access to maneuver the machine.  The

bottom section formed a base for the machine.  The depth was equal to that of the

chamber.  The width was wider to accommodate for the frame columns.  The frame

top section and columns were made of 4 inch thin-walled tubing.  The depth of the

column sections and the top section were equal to the depth of the scissor press.

The columns and top section were centered over the chamber.  Cross bracing

provided stability and created mounting points for the bearings and pneumatic

components as well as routing for electrical wires and air hoses.  Socket access

41

holes were made in the tubing members to prevent soft joints where the pneumatic motor, load cells, and bearings were mounted.  The frame defined the overall machine dimensions which were 8ft - 4-1/4in high, 5ft - 3-1/2in wide, and 3ft - 6-3/4in deep.  Details of the frame can be found in drawing 4 in Appendix A.  The total frame weight was 550 pounds.

### 3.2.4 Plunger

The plunger design was common to what is typically seen in multi-stroke balers.  It consisted of 3 inch c-channel steel members welded in a configuration with gaps for placement of the 4 bale wires.  The ten channel members were a mirror of the chamber floor with the exception of the outer two members.  There was a 1-1/4 inch gap between eight of the members.  The outer two members were offset only by 1-1/8 inch.  This arrangement resulted in a total width of 41 inches making a 1/8 inch of clearance between either side of the plunger and the chamber wall.  The plunger was 40 inches in length, which resulted in 5/8 inches of clearance between the door and rear walls of the chamber and the plunger.

The channels were welded together with two pieces of tubing placed on top of and perpendicular to the channel.  The roller track mentioned in section 3.2.2.1 was attached to the top of each tubing member.  Due to the tubing being placed on the open part of the channel, small sheet metal squares were used to provide more contact area for welds between the channel and the tubing.  This orientation made the plunger more structurally stable and not as dependent on small precision welds between the tubing and the top of the channel flanges.  Thin sheet metal strips were

welded to the upper portion of the channel to serve as guides for the bale wires.  Not only did these form a groove for the wires to travel along, but they also helped to mark the gaps where wires were to be placed, making them easier to spot by eye.  Details of the plunger can be found in drawing 5 in Appendix A.  The total plunger weight was 250 pounds.

### 3.2.5 Bale Removal

Bale removal was achieved through the use of two strips of 1 inch nylon webbing.  One end of the webbing was attached to the chamber floor forward section while a hook was attached to the other.  Each webbing occupied one of the empty floor slots immediately adjacent to slots occupied by bale wires.  The webbing was hooked to points located in the back wall so that it was kept out of the way during normal pressing.  The idea was the webbing would be attached to the plunger before retraction and would force the bale out of the chamber using the webbing as the plunger retracted.  The nylon webbing was intentionally cut longer than needed and steel buckles were incorporated to ensure adjustments were available during testing.

As mentioned in section 3.2.2.1, the box roller track on the plunger was responsible for absorbing the force encountered during retraction.  The track flanges were responsible for handling the vertical forces encountered during bale removal.  The problem was the track was only rated to handle 400 pounds.  Because the plunger weighed 250 pounds, the track was reinforced with sheet metal to ensure it was capable of accommodating these vertical forces.  The difficulty faced was the

reinforcements could not interfere with the normal travel of the rollers across the track.

## 3.3 Pneumatic System

The pressure and air flow needed by the pneumatic gearmotor was provided by a Speedaire® 60 gallon, single stage air compressor manufactured by Dayton Electric®, (Chicago, IL).  Because the gearmotor was capable of reversible rotation, a 4-way, 3-position, directional control valve was used to change the direction of air flow to the motor.  The control valve contained a closed center position and was mounted to a 5-port manifold.  Two ports connected the valve and motor, 2 ports were connected in parallel to the exhaust muffler, and the last port was for the air supply line.

A stacked filter/regulator/lubricator (FRL) manufactured by Norgren®, (Littleton, CO) was used to condition the air before entering the control valve and gearmotor.  The filter removed particles as small as 5 microns and separated water from the air.  An automatic drain emptied the FRL as water accumulated in the separator.  The regulator was a relieving style and contained a pressure gage and a manual adjustment knob.  The FRL was set to regulate the air pressure to 80 psi due to specifications of the gearmotor.  The lubricator was filled with air tool oil and can be monitored overtime through a built-in sight gage.  The lubricator also contained a transparent bowl that allowed the operator to view and manually adjust the rate of oil drop into the air line.  The FRL was adjusted to provide one drop of oil

for every 10 CFM of air consumed.  At no load, the lubricator would provide roughly one drop of oil every 10 seconds.

An industrial-shape, quick-disconnect hose coupling allowed the machine to separate from the air compressor for maneuvering.  For safety, a quarter turn shutoff valve was located downstream from the hose coupling which permits the operator to shutoff flow to the system in the event of an emergency.  All components used 1/4 inch NPT fittings.  The air hose utilized was oil resistant and had a 5/16 inch inner diameter and a 5/8 inch outer diameter.  The pneumatic schematic diagram of the system using ISO standard fluid power symbols is shown in drawing 6 of Appendix A.

## 3.4 Electrical System

The control valve of section 3.3 provided electric-over-pneumatic control of the baler.  It featured double solenoid valves operating on 120 VAC to determine the valve position.  The engaged solenoid would hold its position as long as power was supplied.  An interruption in current would return the valve to the closed center position.  A single-pole, double-throw, 3-position (on-off-on) toggle switch was used to select the control valve solenoids.  The switch and wiring were oriented so that the switch in the upward position would result in the plunger rising and the switch in the downward position would cause the plunger to lower.  Normally closed limit switches were placed in each circuit between the toggle switch and the solenoids as shown in drawing 7 of Appendix A.  Their physical placement on the baler was such

that the traveling collars of section 3.2.2 would contact the limit switch, thus breaking the circuit and returning the control valve to the center position.

Safety was a top priority and there were many safety features involving the electrical system.  A fast-acting, 1-1/2 amp, panel-mount fuse was placed in the circuit to prevent damage to the control valve or persons near the machine in the event of an electrical short.  A UL listed, NEMA 4X enclosure was mounted to the baler frame.  The toggle switch and panel-mount fuse were mounted to this enclosure.  The location of the switch was within easy, unobstructed reach of anyone loading the baler.  This weatherproof enclosure contained electrical wire-nut connections.  Liquid-tight cord grips provided waterproof passage into and out of the box for the electrical cord.  The cord itself contained a ground and 2 insulated copper conductors protected in a rubber jacket.  The ground was connected to the baler frame to prevent electrical shock to anyone contacting the baler in the event of an electrical short causing the frame to become electrically charged.  Covers were fabricated over the limit switches to avoid potential contact between the screw terminals and the frame.  The electrical connections to the control valve were contained in weatherproof enclosures that snapped into place allowing easy disconnection of the electrical cord from the control valve.

## 3.5 Data Acquisition

A Gateway® notebook computer was interfaced with a National Instruments SCXI data acquisition series module for data acquisition.  The module consisted of a

National Instruments DAQ® card-6036E PCMCIA card and an SC-2043-SG signal conditioning board.  Output voltage excitation leads and input signal leads from a linear position transducer and load cells were attached to the screw terminals on the eight channel analog signal conditioning board.  An onboard excitation voltage of 2-1/2 volts was provided to all channels.  The DAQ card was configured using NI-DAQ software, which converted incoming voltage signals to physical values.  LabVIEW®, a graphical programming software package, was used to record the force and distance measurements and elapsed time into an Excel spreadsheet.  LabVIEW allows the user to custom design a virtual instrument to accomplish a task.  For the baler application, the program was developed to continuously scan each board channel 100 times per second.  Each loop iteration averaged 10 recorded samples and returned the average value to the spreadsheet file.  The LabVIEW program can be seen in Appendix C.

### 3.5.1 Force Measurement

Omega® model LCCA-5K 5000 pound capacity S-Beam tension and compression cells were placed between the baler frame bottom section and the chamber.  Rated output was 3mV/V and accuracy was 0.037% full scale (1.85 pounds).  Due to the orientation, the chamber weight and any vertical force applied to the bale were transferred as compression on the load cells.  This design allowed for the weight of the tobacco to be monitored as the chamber was loaded.  This setup also allowed for measurement of the vertical force applied to the tobacco during bale formation.

### 3.5.2 Displacement

A Patriot® model P200-A Linear Position transducer was mounted to the baler frame top section.  The sensor cable was attached to the plunger directly below the transducer.  During a press, the transducer would track the distance traveled.  Because the cable was spring loaded, there was constant tension on the cable and it was retracted with the plunger.  The transducer range was 0-200 inches with an accuracy of 0.1% full scale (0.2 inches) and a rated output of 4.97 mV/V/inch.

# Chapter 4

## Results and Discussion

The machine performance was tested with burley tobacco grown and cured during the 2007 growing season at Central Crops Research station near Clayton, NC. The tobacco was stripped by workers at the research station and placed into burlap sheets denoted by the leaf grade. The burlap sheets were transported to NCSU and stored in the Advanced Curing and Drying Structure located at Weaver Laboratories. Prior to baling, moisture was reintroduced to the tobacco to prevent the leaf lamina from shattering during handling and baling. The moisture reintroduction process for flue-cured is known as bringing the leaves into order, or just simply ordering. Casing is the equivalent term used for burley tobacco.

The tobacco was brought into case using a variable rate humidifying fan. The flow rate of water into the humidifier was controlled through a variable rate flow meter. The burlap sheets were placed on the structure floor, separated by grade and opened. The leaves were spread evenly over the sheet area with a pitchfork. The structure door was closed and the humidifier was set to add 5 gallons of water per hour to the air inside the structure. The leaves were mixed periodically to ensure a more even distribution of cased leaves throughout the pile. After casing, the tobacco was re-bundled and the sheets were tied and weighed. The weight of each grade was summed to gage the accuracy of the load cells on the baler. Table 4.1

lists the weight of each grade of tobacco baled.  Due to poor growing conditions

during the summer of 2007 and losses during curing and stripping, none of the

grades would be capable of producing a bale meeting the target weight of 550

pounds.  In order to fully test the machine, grades would have to be mixed in order

to meet target bale weight.

Table 4.1:  Weight of burley tobacco grades produced during 2007.

| Grade | Weight (lbs) |
|---------|--------------|
| Tips | 398 |
| Leaf | 349 |
| Lugs | 323 |
| Flyings | 348 |

## 4.1 Testing Procedure

The baler was tested in the Research Shop located in Weaver Labs room

130.  The sensor wires from the load cells and linear position transducer were

connected to the signal conditioning board of section 3.5.  The linear position

transducer sensor cable was connected to the plunger.  The LabVIEW program was

initialized and the chamber weight on the load cells and position of the transducer

were displayed on the laptop screen and written to an Excel spreadsheet data file.

The signal conditioning board contained potentiometers to trim the signal for each

channel.  With the plunger in the fully retracted position, the trim pot of the linear

position transducer channel was adjusted to show a value of zero on the screen. The load cells were tared through the LabVIEW program.

Both baler doors were opened and a pre-formed corrugated fiberboard slipsheet was placed upright and against the side of the baling chamber prior to leaf loading. The chamber was loaded from the front with burley leaves. During loading, four samples of 2 to 3 whole leaves each were taken randomly. These samples were individually double bagged, labeled, and stored in a cool, dark area for moisture content analysis. The process was repeated for each bale produced by the machine. The ASABE standard for moisture measurement in tobacco (ASAE S487) was used to find the wet basis moisture content for each bale. The moisture content utilizes the oven-dry method and is calculated as follows by equation 4.1:

$$(4.1) \qquad MC(\% wb) = \frac{WSW - DSW}{WSW} * 100$$

where MC(%wb) is the wet basis moisture content described as a percentage, WSW is the wet sample weight and DSW is the dry sample weight.

An AND® Model GX-6100 precision weighing digital mass balance was used to weigh each sample before and after oven dry. The samples were exposed to 101 degrees C for 24 hours to remove moisture. Table 4.2 summarizes the moisture content data. It should be noted that these moisture content values are considerably

high for baling tobacco.  The desirable moisture content for burley bales ranges from 18-24%wb.

Table 4.2:  Moisture content for burley tobacco bales produced in 2007.

| Bale | Container weight (g) | Wet weight (g) | Dry weight (g) | Moisture content (%wb) |
|------|------------------------|------------------|------------------|--------------------------|
| 1 | 1355.37 | 1450.11 | 1425.26 | 26.23% |
| 2 | 1406.44 | 1499.98 | 1462.01 | 40.59% |
| 3 | 399.56 | 495.79 | 463.66 | 33.39% |
| 4 | 403.97 | 499.01 | 465.55 | 35.21% |
| 5 | 2490.42 | 2581.8 | 2553.02 | 31.49% |
| 6 | 2472.44 | 2562.43 | 2536.21 | 29.14% |

### 4.1.1 Bale Formation

After a sufficient amount of tobacco was loaded into the chamber, the lower door was closed and loading continued through the top opening as shown in figure 4.1.  Once the baling chamber was full of loose leaf burley, the top door was closed and the operator opened the shutoff valve and toggled the switch mentioned in section 3.4 to the down position.  This procedure actuated the control valve, allowing air to enter the pneumatic gearmotor and rotate the acme rod.  This rotation forced the plunger to press the tobacco into a bale.  At full stroke, one of the limit switches of section 3.4 interrupted the electric circuit, allowing the control valve to return to the closed center position, thus shutting off airflow to the gearmotor.  At this point, rotation of the acme rod stopped and the self-locking power screw held the plunger

in the fully stroked position.  The operator toggled the switch to the up position,

reversing airflow to the gearmotor, resulting in reversed shaft rotation and raising of

the plunger.  The other limit switch stopped the plunger at the fully retracted position.

At this point, loading commenced again until the chamber was full.  This process

was repeated until the laptop display showed the amount of tobacco in the chamber

was at target bale weight.  Typically, 3 to 4 cycles were required to reach a final bale

weight of 550 pounds.  During this time, the data acquisition system was constantly

recording the force present on the load cells and the position of the linear position

transducer.  These values, along with the elapsed time, were written to an Excel

spreadsheet data file.



Figure 4.1: Baler chamber loading through top door opening.

### 4.1.2 Bale Wire Tying

The compressed bale was secured with 4 steel tie wires. The wires were 150 inches long with pre-formed loops on each end. After the terminal press, the plunger was held at full stroke to facilitate wire tying. Both chamber doors were opened and a wire was inserted between the offset channels in the chamber floor. Thin gage sheet metal added to the floor slots during fabrication ensured that the wires slid all the way through the grooves to the rear of the chamber. At the rear of the chamber, the wires were pushed back through similar slots in the rear wall and plunger to the front of the chamber. The ends of each wire were looped together at the front of the bale to firmly secure the perimeter of the compressed bale.

### 4.1.3 Bale Removal

After the wires were hooked, the plunger was retracted slightly. The hooks at the end of the nylon webbing of section 3.2.5, which were previously hooked to the rear wall, were now placed over bolts assembled in the plunger. At this point, the plunger was retracted fully and the nylon webbing was made taut. During this process, the nylon webbing forced the bale forward and eventually out of the chamber. The webbing was then unhooked from the plunger and placed back on the rear wall to remain out of the way during bale formation. The bale was then rotated 90 degrees onto the fiberboard slipsheet.

## 4.2 Results

The baler produced 6 bales with force and displacement measurements recorded for 5 of them. An error in the data acquisition system resulted in no data being saved for the third bale. A tabular summary of the data is shown in table 4.3. The first two bales consisted of mixed grades in order to produce bales near the target bale weight of 550 pounds. During leaf loading, a burlap sheet was placed in between the different grades in order to keep the grades separate for later baling. These mixed grade bales were formed solely to record the force and displacement data for a full bale. After formation, the leaves from the different grades were removed from the baling chamber manually.

Table 4.3: Bale data summary.

| Grade | Mixed Grades | | | | Mixed Grades | | | Leaf | Flyings | Tips |
|---|---|---|---|---|---|---|---|---|---|---|
| Bale | 1 | | | | 2 | | | 4 | 5 | 6 |
| Press | 1 | 2 | 3 | 4 | 1 | 2 | 3 | 1 | 1 | 1 |
| Tobacco Weight (lbs) | 260 | 343 | 424 | 500 | 421 | 523 | 590 | 349 | 348 | 393 |
| Peak Force (lbs) | 317 | 544 | 947 | 1446 | 459 | 845 | 1046 | 1373 | 1308 | 965 |
| Pressure (lb/in$^2$) | 0.19 | 0.32 | 0.56 | 0.85 | 0.27 | 0.50 | 0.61 | 0.81 | 0.77 | 0.57 |
| Density (lb/ft$^3$) | 6.95 | 9.17 | 11.3 | 13.4 | 11.3 | 14.0 | 15.8 | 19.9 | 19.9 | 16.1 |
| Press Time (sec) | 71 | 81 | 106 | 201 | 85 | 180 | 190 | 132 | 305 | 233 |
| Moisture Content (%) | 26 | 26 | 26 | 26 | 41 | 41 | 41 | 35 | 31 | 29 |

A graphical summary of bale 1 formation is displayed in figure 4.2. The chart is organized with plunger distance on the y-axis and compressive force on the x-axis

to visually display the force increasing as the plunger moves vertically down.  As shown in figure 4.2, four presses were required to complete the bale.  The general trend of the increasing rate of compressive force required as distance increases was expected.  For the first and second press, the slope changes abruptly when the plunger reaches 25 inches.  Upon inspection of the inside chamber surfaces, noticeable paint scraping occurred on the chamber sides at approximately the 25 inch mark for plunger displacement.  The degree of freedom for plunger movement is laterally within the chamber.  By design, the sides of the chamber are what restrict movement of the plunger laterally.  The paint scrape marks at 25 inches along with the evidence in figure 4.2 suggest the friction between the sheet metal sides and the plunger increased at this point requiring more compressive force.  The third press shows a smooth increase in force for increasing distance.  The fourth press shows a discontinuity at 15 and 20 inches that resulted in the gearmotor stalling.  The press was paused for a short amount of time while the tobacco relaxed and then restarted.
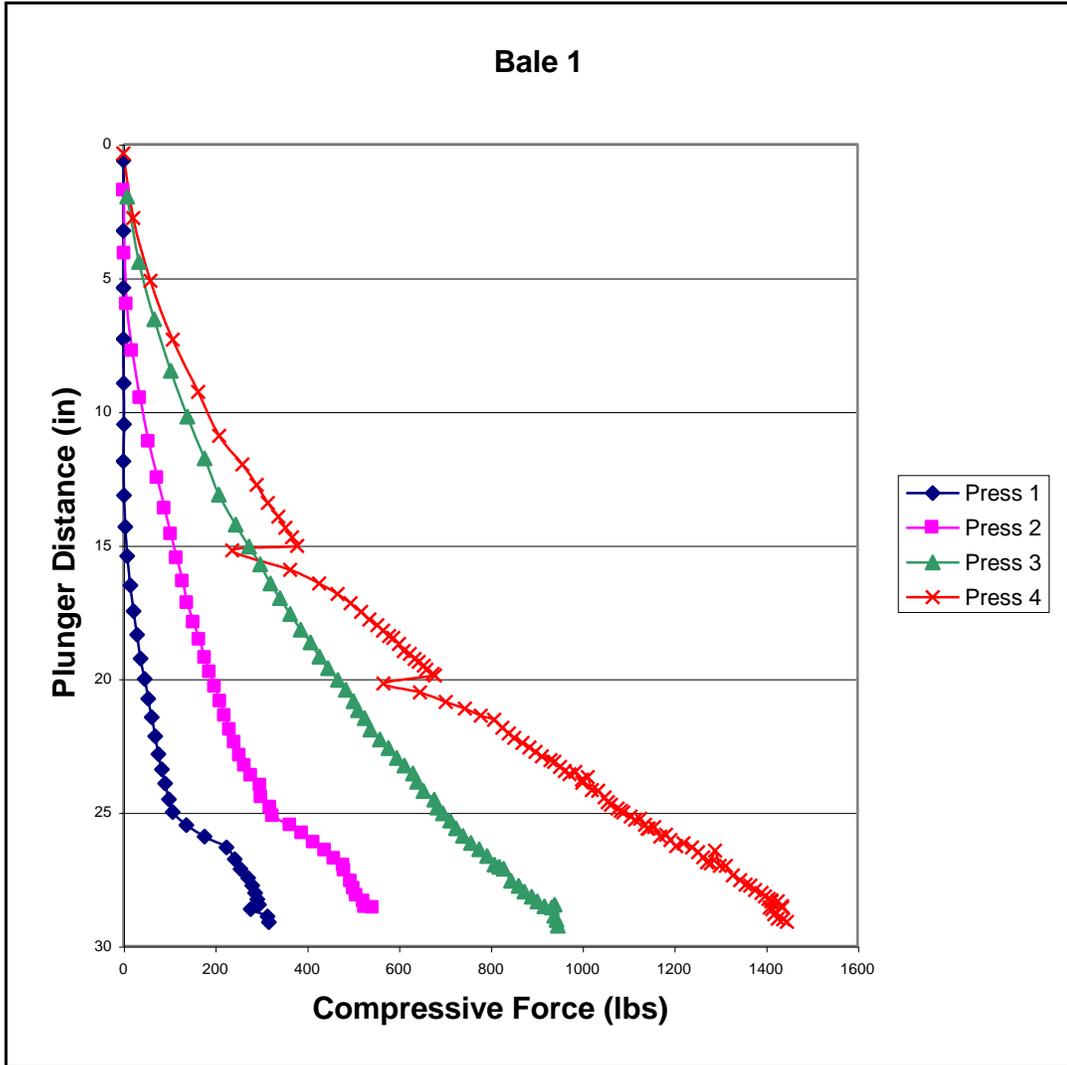
Figure 4.2: Bale 1 force and displacement results.

Table 4.3 shows that 260 pounds were initially loaded into the chamber and then approximately 80 pounds were added between each successive press for a final bale weight of 500 pounds. The average wet basis moisture content for this bale was 26% and was the lowest of all bales tested. Figure 4.3 is the compression

force response as a function of compressed bale density. It should be noted that the density would decrease once the bale is removed from the chamber and allowed to expand. The peak force recorded for bale 1 formation was approximately 1,450 pounds at a compressed bale density of 13.4 pounds per cubic foot.
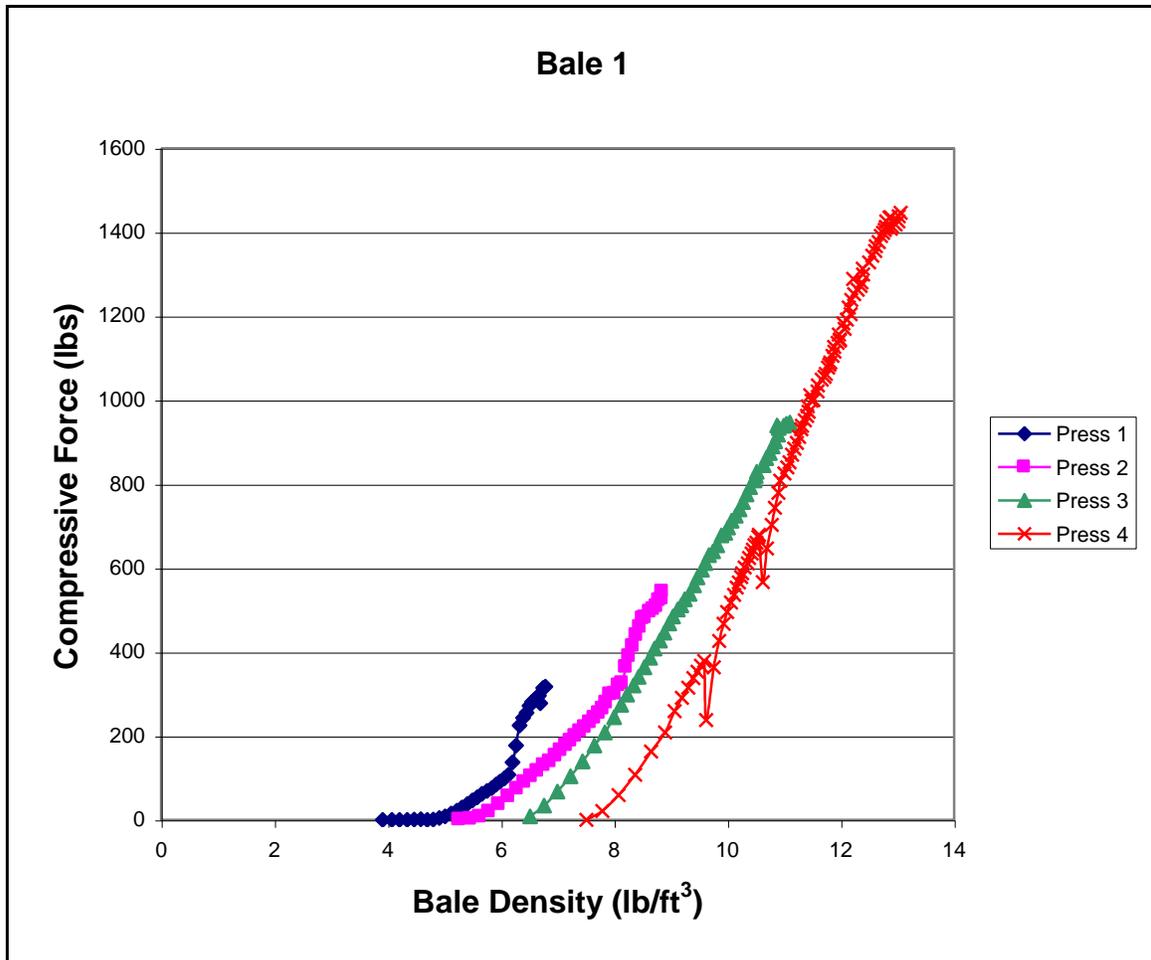


Figure 4.3: Bale 1 force vs density results.

Figure 4.4 is the force and distance curve for the formation of bale 2. As shown, 3 presses were required. From table 4.3, 421 pounds of tobacco were loaded into the chamber initially with 100 pounds being added before the second press. For the final press, only 70 pounds were added, which did not fill the entire volume of the chamber. Thus, the plunger traveled about 6 inches before being loaded by tobacco. This observation also explains why the final press curve did not overtake the other press curves in force required until late in the stroke. It should also be noted that the moisture content of this bale averaged 41% on a wet basis. This moisture level was the highest of all bales tested. Figure 4.5 represents the compression force as a function of bale density. The peak force produced during formation of bale 2 was approximately 1,050 pounds at a density of 15.8 pounds per cubic foot. The peak force for bale 2 was lower than the peak force for bale 1 even though it was baled to a higher density. This outcome was expected based on results by Ellington (2000), who demonstrated that less force is required to bale tobacco at a higher moisture content.

After collecting data on full sized bales, the flyings, lugs, leaf, and tip grades were baled individually. Due to the low amount of tobacco for each of these grades, the baler would have to be modified in order to form bales of a desirable density. The bales would have to be compressed to a distance beyond the capabilities of the machine as designed and built. A stroke of at least 43 inches would be required to compress the 323 pounds of bale 3 to at least 13 pounds per cubic foot. A spacer made out of wooden pallets and blocks was placed in between the plunger and the

tobacco.  The spacer thickness of 20-1/4 inches provided a sufficient amount for compressing the individual grade bales and limited the distance traveled by the plunger at zero load.  The bales produced were about half the height of a normal bale.  Because of this, the bale wires had to be modified in order to wrap around the bale to hold its shape.  The wires were cut to approximately 1/2 to 2/3 their original length and new loops were formed on the end so the wires could be attached in their normal manner.
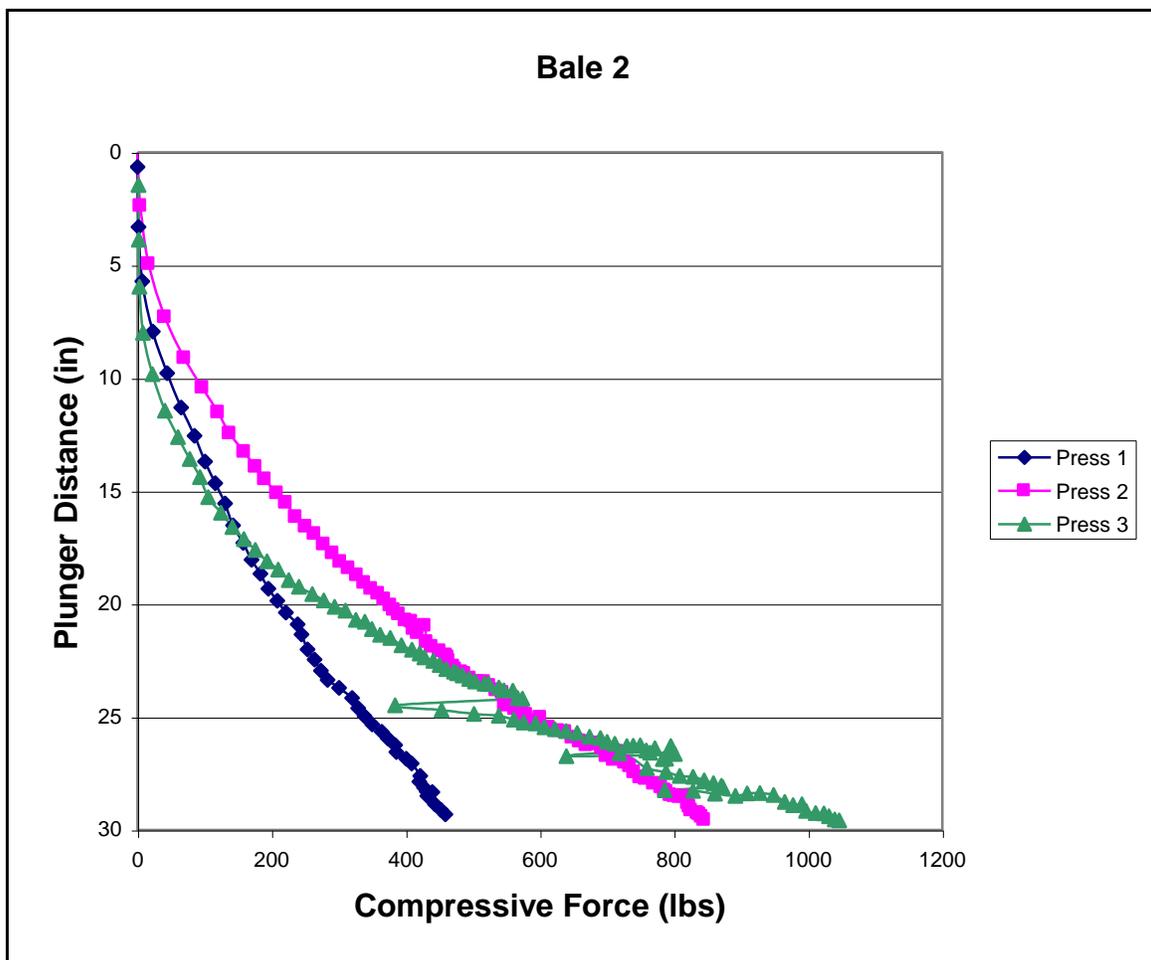


Figure 4.4: Bale 2 force and displacement results.

60
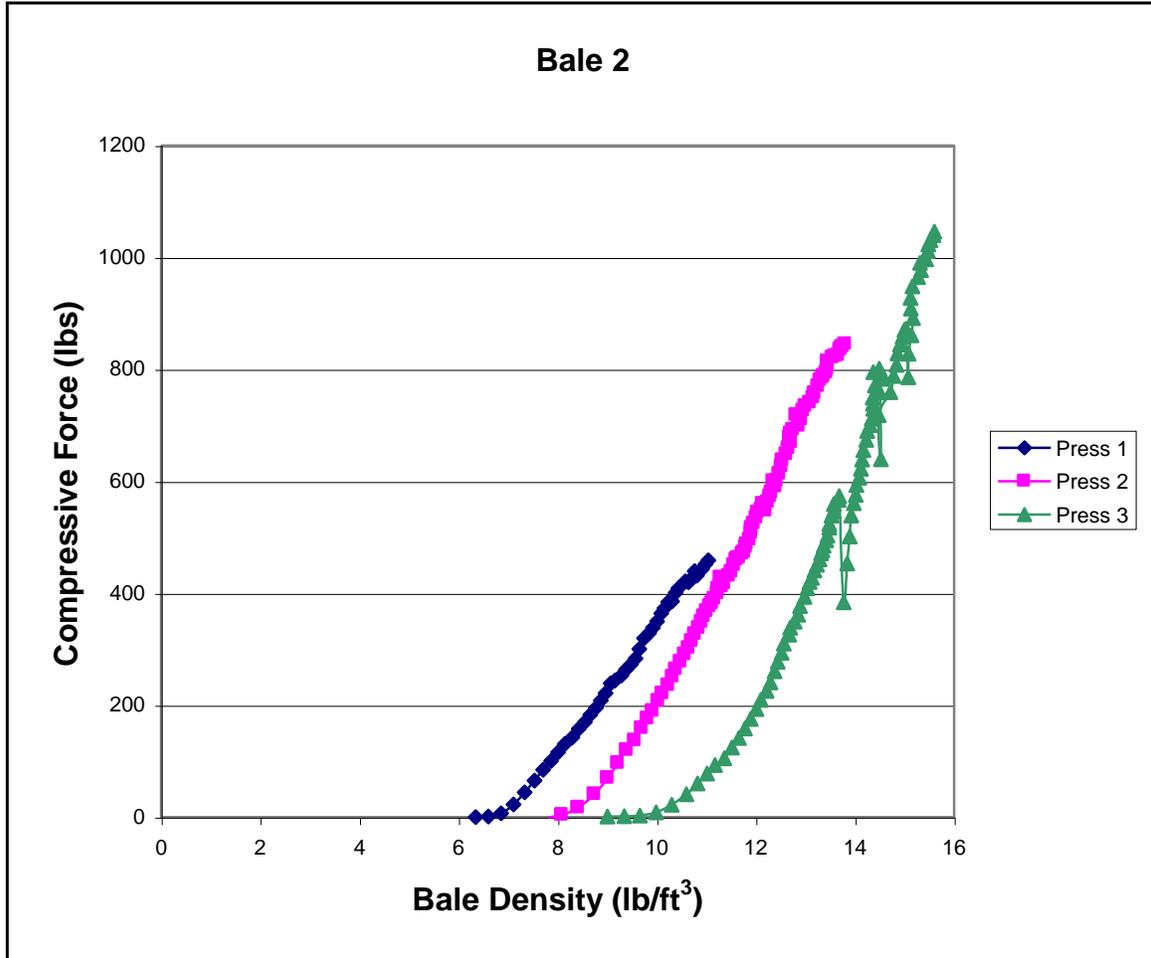
Figure 4.5: Bale 2 force vs density results.

A summary of the force and displacement measurements for these bales is shown in figure 4.6 and the force as a function of density is shown in figure 4.7. As mentioned earlier, data was not recorded for bale 3 due to a problem with the DAQ system. The peak force recorded for bale 4 was 1,373 pounds compressed to 19.9 pounds per cubic foot at a wet basis moisture content of 35%. Bale 5 was also

baled to 19.9 pounds per cubic foot and required a peak force of 1,308 pounds at a wet basis moisture content of 31%.  Bale 6 had a wet basis moisture content of 29% and was compressed to a density of 16.1 pounds per cubic force.  The peak force required for this bale was 965 pounds.  Once started, the plunger traveled about 12 inches before meeting the spacer and becoming loaded.  This occurrence is evident in figure 4.6 and figure 4.7 as the initial zero reading on each chart.  The plunger stopped on bale 6 and was not restarted because an adequate density had already been achieved and stroking the full 30 inches was unnecessary.  It should be noted that some leaves used for data collection of bales 4, 5, and 6 did have previous compression history from bales 1 and 2.
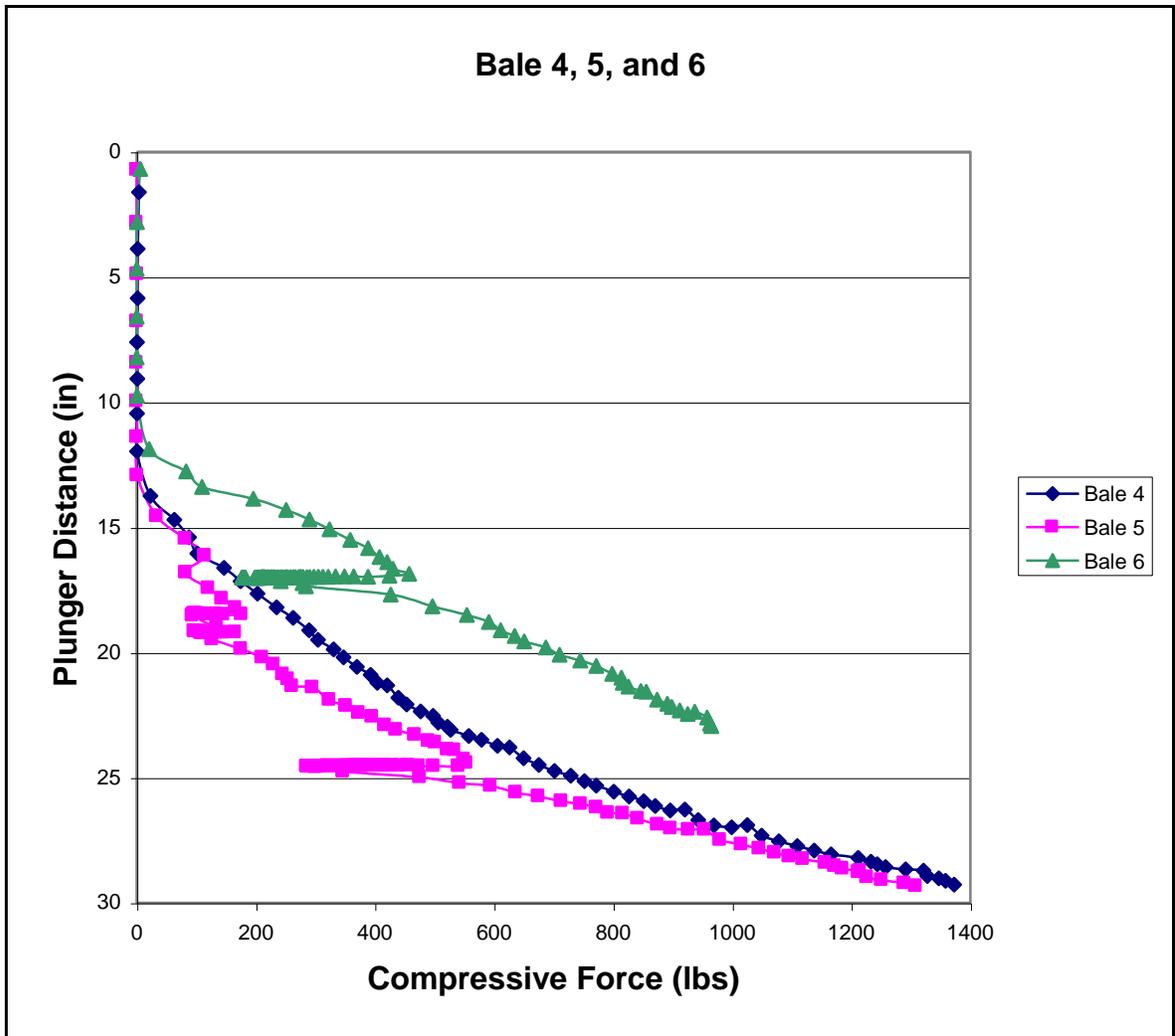
Figure 4.6: Bale 4, 5, and 6 force and displacement results.
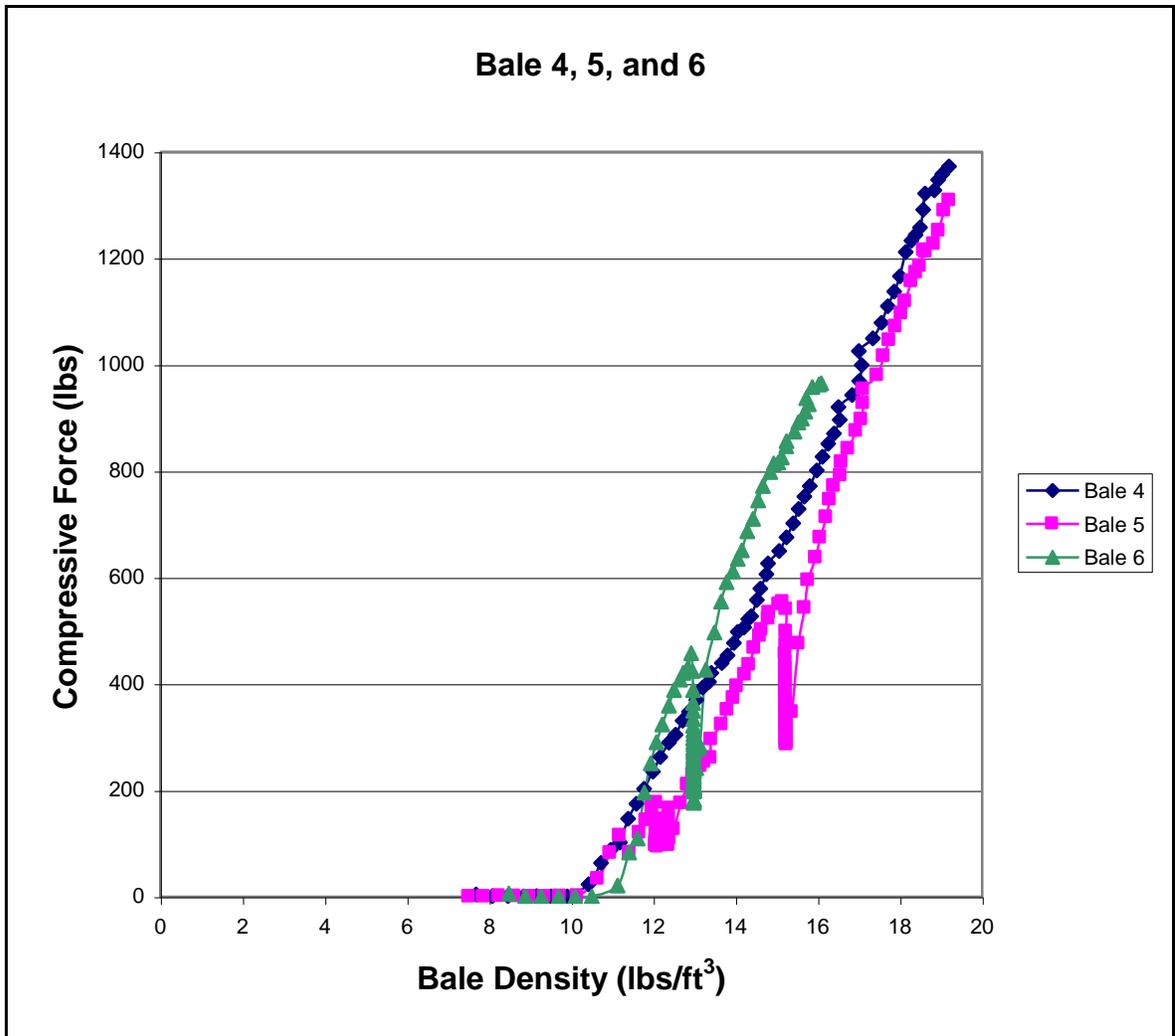
Figure 4.7: Bale 4, 5, and 6 force vs density results.

Table 4.3 lists the press time for each press in seconds.  Press time is the

time it took for the plunger to travel from fully retracted to fully extended.  The

retraction times averaged 70 seconds to fully retract from full stroke.  With this

information, the quickest press cycle occurred during press 1 of bale 1 and took 2

minutes and 21 seconds.  The longest press cycle was 6 minutes and 15 seconds during the formation of bale 5.

The retail raw material cost associated with fabrication and assembly of the baler is shown in table 4.4 broken down by category.  The overall material cost of the machine was $6,200.  It should be noted that the three load cells accounted for $1,335 of that cost along with the air compressor and pneumatic gearmotor at $900 and $855 respectively.

Table 4.4: Pneumatic baler retail raw material cost summary.

| Category | Retail Cost |
|---|---|
| Steel | $2,110 |
| Electrical | $1,410 |
| *Load cells* | *$1,335* |
| Power Transmission | $1,405 |
| *Pneumatic gearmotor* | *$855* |
| Pneumatics | $1,275 |
| *Air compressor* | *$900* |
| **Total Retail Raw Material Cost** | **$6,200** |

## 4.3 Discussion

The design equations did not accurately predict machine performance in the area of vertical force production.  Equations 3.2 and 3.3 were used to show that 1,717 pounds of axial force at maximum gearmotor torque of 630 inch-pounds

should force the scissor press to produce a vertical force of 10,000 pounds. The maximum peak force experienced during bale formation was 1,446 pounds during the terminal press of bale 1. One explanation for this performance is friction. The scissor press has many areas where friction loss occurs. The main sources of friction loss are in the threads between the acme nut and acme rod, the pivot point between the ends of the traveling collars and the tops of the scissor arms, the pivot point at the center of the scissor arms and the rollers at the bottom of the scissor arms. However, the most critical area is friction between the steel acme rod and the steel acme nut inside the traveling collars. This point is where the rotational motion is transformed into linear motion.

During the review of literature, there was no consistent value given for the coefficient of friction between sliding steel surfaces. Some sources listed values below the 0.25 coefficient of friction used in the design equations of section 3.2.2.3 and some listed values as high as 0.57 for sliding steel surfaces using no lubrication. The value for the coefficient of friction between the acme nut and acme threaded rod is extremely difficult to estimate. Sliding friction is typically lower than static friction but as the velocity between the two surfaces decreases, the sliding friction coefficient begins to approach the static friction coefficient. Knowing this maximum friction coefficient value is key for the selection of the gearmotor. For example, a value of 0.57 for the coefficient of friction in equation 3.3 results in a torque requirement of nearly 1,240 inch-pounds. This value is almost double the capability of the selected gearmotor, just to produce 10,000 vertical pounds at the plunger.

Further complicating the issue is the pressure increase between the threaded pairs as the scissor press is loaded during compression of the bale. As the plunger experiences added resistance from the bale, there is more resistance between the acme rod and the acme nuts. This resistance is due to the normal force increasing between the acme threads of the nut and rod. An increase in normal force increases the friction force. The friction loss may start out rather low during baling, but due to the increase in load and resultant decrease in gearmotor shaft rotation, the friction loss may increase exponentially. Attempts to lower the friction loss during testing were in vain. Even after application of Moly Lube®, an industrial strength aerosol lubricant, the change in machine performance was negligible.

In order to test the true maximum force the system was capable of producing, four wooden 2 x 4 studs were cut approximately 38 inches in length and placed in the corners of the baling chamber between the plunger and floor. At this length, the plunger would contact the wood at nearly full stroke (80 degrees vertical) representing the maximum mechanical advantage of the system. The powerscrew was activated until the gearmotor stalled. The maximum force recorded on the load cells was 4,000 pounds. Assuming a 15% loss due to all other frictions points between the traveling collars and the plunger, the axial force produced by the acme rod is about 780 pounds at full stroke from equation 3.2. Equation 3.3 and TK Solver were used to determine the friction coefficient in the threaded pair. At maximum motor torque of 630 inch-pounds, an axial force of 780 pounds, and all

other values remaining constant, the coefficient of friction is estimated to be 0.64 between the acme nut and acme rod threads at full stroke.

Increasing friction can also be attributed to the increased press cycle times. The quickest pressing time happened during press 1 of bale 1 at 71 seconds and the plunger retraction times averaged 70 seconds with a total press time of 2 minutes and 21 seconds. The scenario discussed in section 3.2.2.6 suggested the press and retraction times to be around 30 seconds each. It was expected that increased friction causing increased force would also result in a longer press cycle.

It was also expected that the machine would have no trouble compressing the leaves during bale formation. As shown in figures 4.2-4.7, this was not the case. Several times during pressing, the gearmotor stalled and had to be restarted. The scissor press design equations were based on performance at maximum tobacco resistance, which for the scissor press was experienced at 80 degrees vertical. However, this only describes a snapshot of the system. It was estimated that the press would experience maximum resistance from the tobacco at this point. While this statement is true, the system performance prior to this point should be discussed.

Equation 3.2 and TK Solver were used to develop the theoretical mechanical ratio of the scissor press as the scissor arm angle changed during the stroke. As shown in table 4.5, the scissor press is in a state of mechanical disadvantage when fully retracted (denoted by the orange color). It is not until the scissor arms are 45 degrees vertical does the system enter a state of mechanical advantage (denoted by

beige). As seen in table 4.4, this occurrence happens at a plunger distance of approximately 20 inches. It is only in the last 1/3$^{rd}$ of plunger stroke does the system exert a mechanical advantage during bale formation. This reality means whatever vertical resistance the plunger encounters from 0 to 20 inches, the axial force produced will have to exceed this value to avoid stalling. Every instance of the gearmotor stalling occurred in the mechanical disadvantage state of the scissor press with the exception of press 3 for bale 2 and the formation of bale 5.

Table 4.5: Mechanical ratio of scissor press stroke.

| F (lbs) | Ft (lbs) | Theta (deg) | Plunger Distance (in) | Ratio |
|---|---|---|---|---|
| 572900 | 10000 | 1 | --- | 57.3 |
| 114301 | 10000 | 5 | --- | 11.4 |
| 56713 | 10000 | 10 | --- | 5.7 |
| 43315 | 10000 | 13 | 0 | 4.3 |
| 37321 | 10000 | 15 | 1.5 | 3.7 |
| 27475 | 10000 | 20 | 4.8 | 2.7 |
| 21445 | 10000 | 25 | 8.0 | 2.1 |
| 17321 | 10000 | 30 | 11.1 | 1.7 |
| 14281 | 10000 | 35 | 14.1 | 1.4 |
| 11918 | 10000 | 40 | 16.8 | 1.2 |
| 10000 | 10000 | 45 | 19.4 | 1.0 |
| 8391 | 10000 | 50 | 21.8 | 1.2 |
| 7002 | 10000 | 55 | 23.9 | 1.4 |
| 5774 | 10000 | 60 | 25.8 | 1.7 |
| 4663 | 10000 | 65 | 27.4 | 2.1 |
| 3640 | 10000 | 70 | 28.7 | 2.7 |
| 2679 | 10000 | 75 | 29.7 | 3.7 |
| 1763 | 10000 | 80 | 30.5 | 5.7 |
| 875 | 10000 | 85 | --- | 11.4 |
| 175 | 10000 | 89 | --- | 57.3 |

Bale 5 was unique in that the motor stalled at a lower force than what was produced at the same plunger distance during bale 4.  This can be seen in figure 4.6.  If the motor produced 650 pounds at a distance of 24 inches, why did it stall at 550 on the next press?  The answer is due to the capacity of the air tank.  The air compressor used to power the baler also handled the entire air demand for the Weaver Labs Research Shop as well as much of the rest of the building.  Due to many pneumatic operations going on during this time, the tank reached the low-pressure point sooner during this press resulting in a lower force stalling the pneumatic motor.  During this particular case, the motor was not restarted until the air compressor recharged the air tank, resulting in a long pause during the press.

Another result of the mechanical disadvantage is the undesirable performance of the bale removal device.  As explained in section 3.2.5, bale removal was achieved via the retracting plunger.  During retraction, the scissor press is losing its mechanical advantage and is moving into the area of mechanical disadvantage.  As a result, the normal force between the threaded pair increases which increases the frictional force.  An increasing amount of axial force is needed to retract the full weight of the plunger and as a result the gearmotor slows down.  As the motor slows down, the velocity between the sliding threads decreases which also increase the friction coefficient.  This compounding problem caused the machine to struggle to remove the bales.  The bales had to be helped out in order to fully remove them from the baling chamber.

The criteria to produce 10,000 pounds was an educated estimate based on previous work, however, a similar operation to current balers was the ultimate design criteria.  It was desired to produce force and displacement data for bale formation from a multi-stroke, vertical baler in order to optimize the design for future machines. Force and displacement data was produced for each bale along with the compressed bale density, average moisture content (% w.b.), and the time it took to compress the bale.

The retail raw material cost was $6,200.  Assuming a wholesale material cost of 70% of retail, the wholesale material cost would be $4,340.  This would leave a 57% markup to cover labor, overhead, and profit margin for full production of the prototype machine if the list price were $10,000.  The cost for commercially produced balers currently is as high as $55,000.

**4.4 Data Comparison**

Several variables determine the amount of force required for bale formation. The most influential variables are moisture content, plunger velocity, and compressed density.  A few interesting comparisons regarding interaction of these variables can be made from the data.  As would be expected, the bales formed in the quickest amount of time had the higher plunger velocities and typically required the highest force.  A lower plunger velocity means the press took longer and the tobacco was able to experience more of a force decay.  This occurrence resulted in less force required to form the bale.  Bales with higher plunger velocities did not

71

have as much time for the decay response to take place and thus required higher forces.

Bale 2 and bale 6 were baled to similar densities. Bale 2 required more force even though the average moisture content was higher than bale 6. Bale 2 had a higher moisture content and required less force initially which resulted in a higher speed. This higher speed resulted in a higher peak force to form bale 2. Density is the difference when comparing bale1 and bale 6. Both bales had similar moisture content. Bale 1 was baled to a lower density but required more force to form. The lower density allowed the bale to be formed faster resulting in more force due to less force decay.

Previous work has suggested that stalk position may also affect force requirement. After comparing bale 4 (leaf), bale 5 (flyings), and bale 6 (tips), there is not enough consistency to definitively say that stalk position is that significant. The leaf and flyings were baled to the same density. The flyings were slightly drier than the leaf but the leaf had the highest plunger velocity of all three bales and required the most force. The tips had the lowest moisture content of all three bales but required the least force. A comparison of the tips and flyings show that the tips were drier and baled faster, but the flyings were baled to a higher density and required more force. The tips were drier than the leaf and the leaf was baled faster and to a higher density, requiring the higher peak force.

# Chapter 5

## Summary and Conclusions

It was desired for the operation of the prototype pneumatic baler to be similar to current multi-stroke, vertical balers which typically operate using hydraulic components. In order to produce such a machine, design criteria were developed to establish the foundation of the baler design. The machine was designed to produce burley and flue-cured tobacco bales utilizing a pneumatic gearmotor-driven power screw and a scissor press.

### 5.1 Pneumatic Baling Summary

Within the tobacco industry, the design of a tobacco baler is affected by the tobacco producer and the tobacco buyer. For a producer, the most important aspect of performance is how quickly the machine can form a bale and the cost of the machine. For the buyer, the quality of the product coming out of the machine is very important. This pneumatic baler shows that tobacco can be baled using "zero-risk" components. Pneumatics poses no potential risk for contaminating the product or safety hazards for workers using the machine. While the machine did under-perform in press cycle timing compared to hydraulic balers, it can be improved through modifications to future machines. With the lower cost pneumatic components, this problem can also be alleviated through lower total machine cost allowing a producer

to own multiple machines at roughly the same cost as a single hydraulic machine. Multiple machines would increase the overall throughput and would be especially appealing to burley growers.  Burley is harvested, cured and stripped whole stalk.  In a burley stripping room, containers holding stripped leaves often pile up while the machine is baling a different grade.  Multiple machines would allow multiple grades to be baled simultaneously.  Considering the fact that each of these machines could operate off a single dedicated air compressor makes the lower cost machine very feasible even though it underperforms against a single hydraulic baler.

## 5.2 Recommendations for Future Work

For future work involving baling, it would be very desirable to instantly know the leaf  moisture content just prior to baling.  A limitation to the oven-dry method is that the samples are taken, the tobacco is baled, and then the moisture content is measured at least a day later.  The sample cannot be tested before baling and remain accurate to the tobacco going into the baling chamber unless that tobacco can be held at constant moisture content.  Instantly knowing the moisture content would allow for tests where certain variables could be controlled.  For example, a true comparison between stalk positions could be made if moisture content, plunger velocity, and compressed density were all controlled variables.  Controlling certain variables to determine which variable had the most significant control over plunger force would help to increase the efficiency in the design of tobacco balers.

Much can be recommended to future work involving pneumatic balers. First of all, the friction coefficient can be lowered in the traveling collars by using a recirculating ball screw similar to that shown in figure 5.1. A recirculating ball screw has a much lower and more predictable friction coefficient. Reducing friction would prevent motor torque from being lost. This feature would result in a smaller and less costly motor being able to produce the same results as a larger and more expensive motor. Friction is also lost when the plunger contacts the insider chamber surface during bale formation. Two plastic strips lining each inside surface of the bale chamber would prevent metal-on-metal contact and reduce friction.
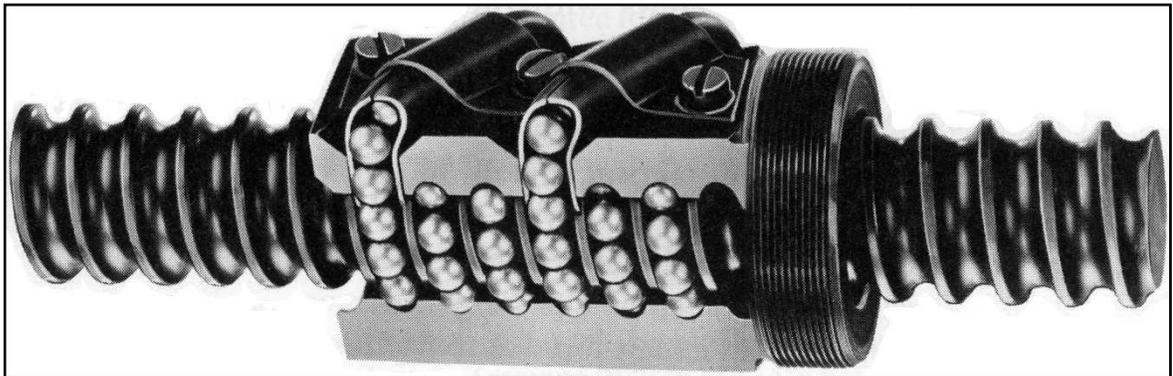


Figure 5.1: Typical recirculating ball screw cutaway view (www.mech.uwa.edu.au).

Having a dedicated compressor close by the baler would be very beneficial. A dedicated compressor would ensure the compressor is working to meet the need of the baler and not other pneumatic users connected to the air tank. The gearmotor

would have the ability to run longer before losing pressure and the pause to wait for the compressor to shutoff would be eliminated.  Several times during testing, the operation was paused to allow the compressor to fully recharge the air tank.  A compressor located close by the baler would reduce the air drag experienced in hoses and fittings and allow the machine to operate more efficiently.  There were 30 ft of 5/16 I.D. airline connected between the air tank and baler fabricated for this project.  Long runs of air hose increase air drag and result in a lower flow rate to the gearmotor.  Minimizing the runs of airline or increasing the hose inner diameter would be very beneficial to the baler performance.  The low flow rate to this baler made press cycle times much longer.  A no load test of the pneumatic gearmotor and a handheld RPM gage indicated that the motor was receiving 30 CFM according to the gearmotor specifications charts.  The gearmotor was rated at 200 rpm's at 60 CFM and was only turning 100 rpm's during the no load test.

Perhaps the most untapped potential remains with the scissor press.  A scissor device is used to translate horizontal motion into vertical motion within a very small footprint.  With the ability to use compound scissors, the potential is almost limitless.  Single-stroke balers could achieve the long strokes needed through a compound scissor press rather than with long and bulky hydraulic cylinders.  Tuning the scissor press to match the requirements for baling tobacco would also be beneficial.  Simply changing the position where the two scissor arms pivot to form the "X" shape would modify the behavior of the device.  Moving this pivot point up or down would affect the mechanical advantage of the device as well as the stroke.  It

has been found through testing of this baler that more force is needed during the early part of the stroke where the press experiences a mechanical disadvantage but has ample force towards the end of the stroke with an increasing mechanical advantage.

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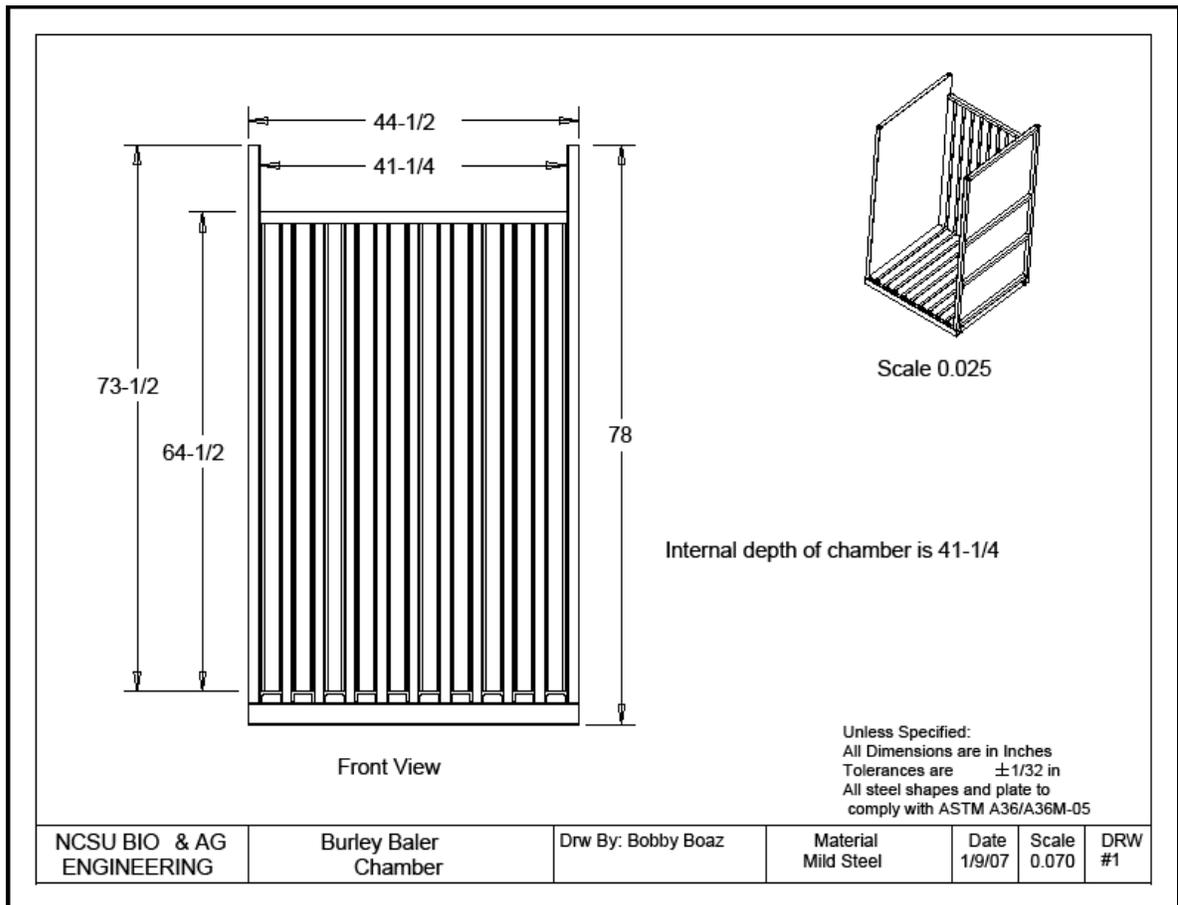
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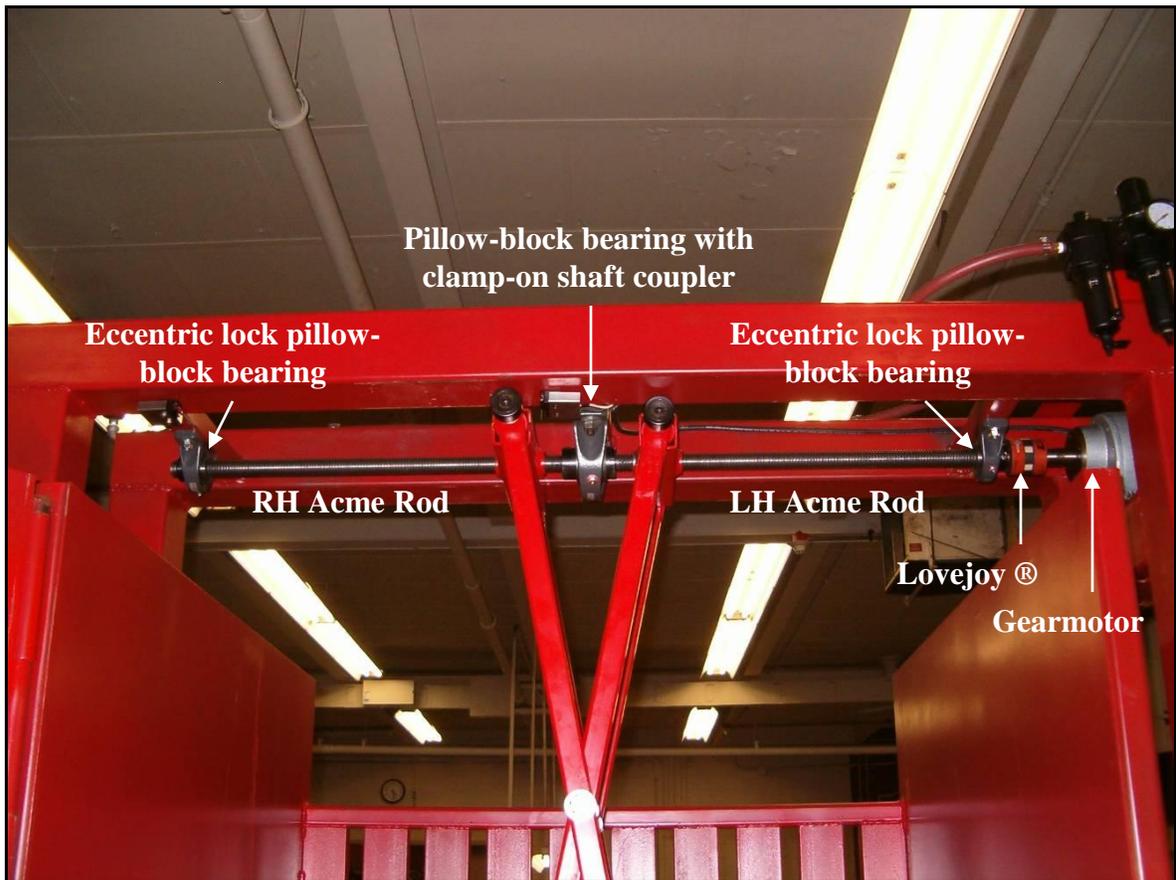
# APPENDICES

# Appendix A – Tobacco Baler Detailed Drawings

Drawing #1: Baling Chamber



Scale 0.025

Internal depth of chamber is 41-1/4

44-1/2

41-1/4

73-1/2

64-1/2

78

Front View

Unless Specified:
All Dimensions are in Inches
Tolerances are     ±1/32 in
All steel shapes and plate to
  comply with ASTM A36/A36M-05

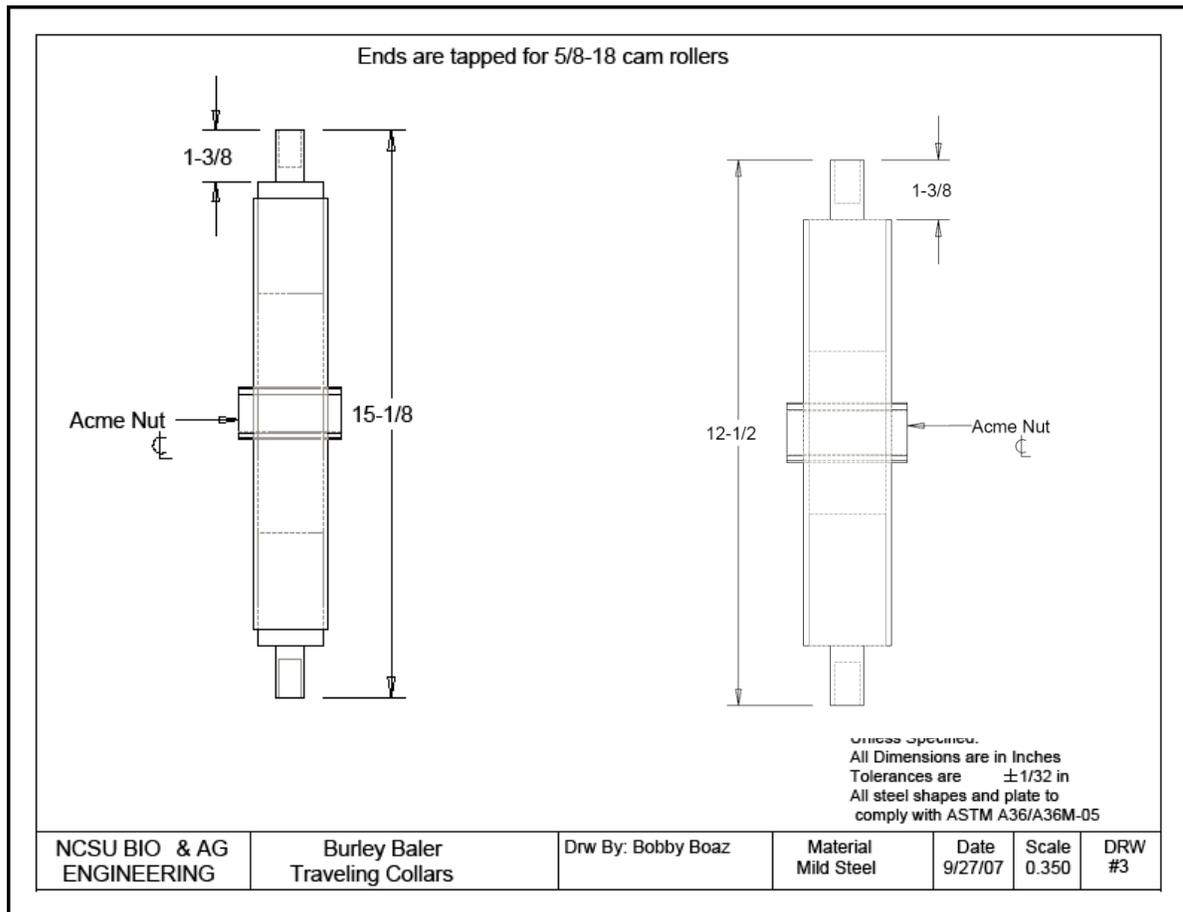| NCSU BIO & AG ENGINEERING | Burley Baler Chamber | Drw By: Bobby Boaz | Material Mild Steel | Date 1/9/07 | Scale 0.070 | DRW #1 |
|---|---|---|---|---|---|---|

# Appendix A – Tobacco Baler Detailed Drawings

Drawing #2: Acme Rod Assembly

# Appendix A – Tobacco Baler Detailed Drawings

Drawing #3: Traveling Collars



Ends are tapped for 5/8-18 cam rollers

1-3/8

Acme Nut

15-1/8

1-3/8

12-1/2

Acme Nut

Unless Specified:
All Dimensions are in Inches
Tolerances are     ±1/32 in
All steel shapes and plate to
comply with ASTM A36/A36M-05

| NCSU BIO & AG ENGINEERING | Burley Baler Traveling Collars | Drw By: Bobby Boaz | Material Mild Steel | Date 9/27/07 | Scale 0.350 | DRW #3 |

# Appendix A – Tobacco Baler Detailed Drawings

Drawing #4: Frame



| NCSU BIO & AG ENGINEERING | Burley Baler Frame | Drw By: Bobby Boaz | Material Mild Steel | Date 7/25/07 | Scale 0.030 | DRW #4 |

Unless Specified:
All Dimensions are in Inches
Tolerances are ±1/32 in
All steel shapes and plate to comply with ASTM A36/A36M-05

Front View     Right View

63-1/2    100-1/4    20    42-3/4
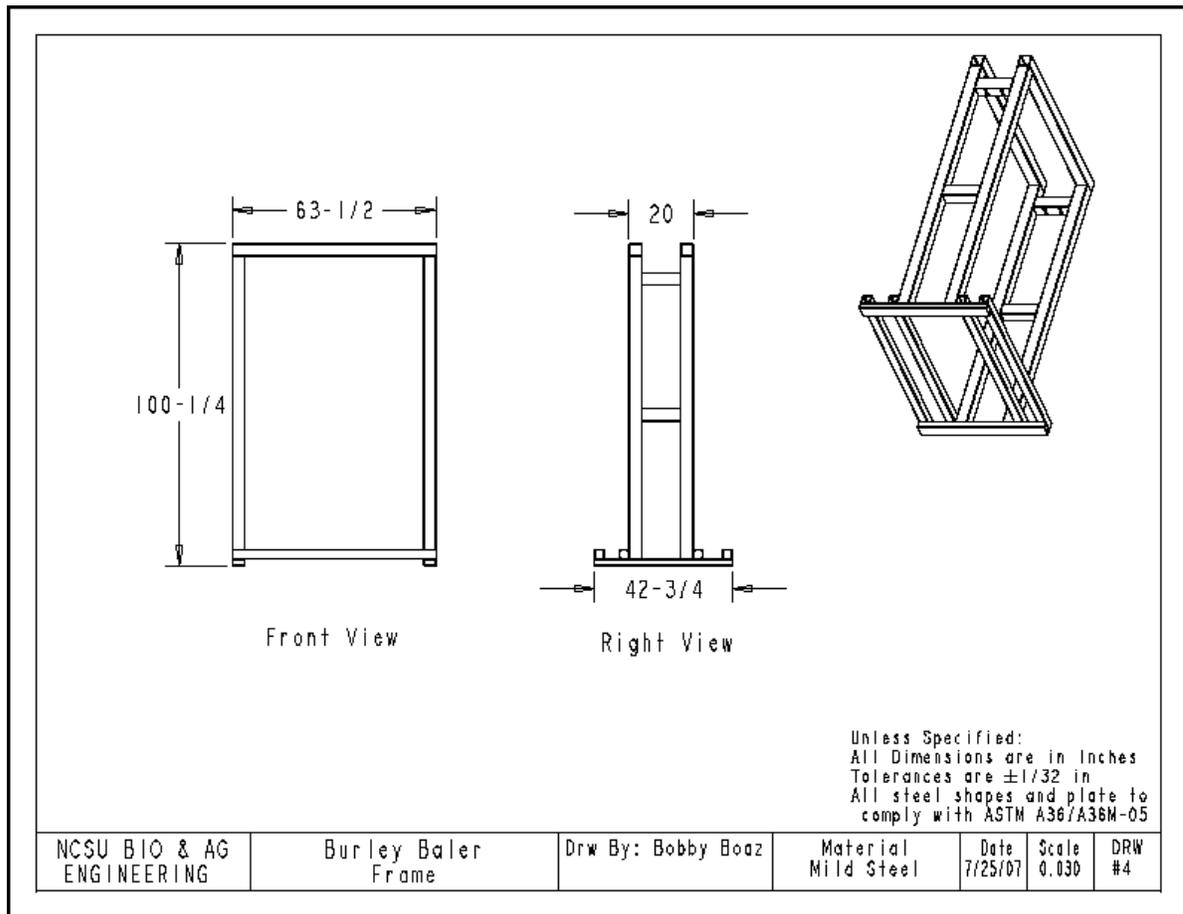
# Appendix A – Tobacco Baler Detailed Drawings

Drawing #5: Plunger

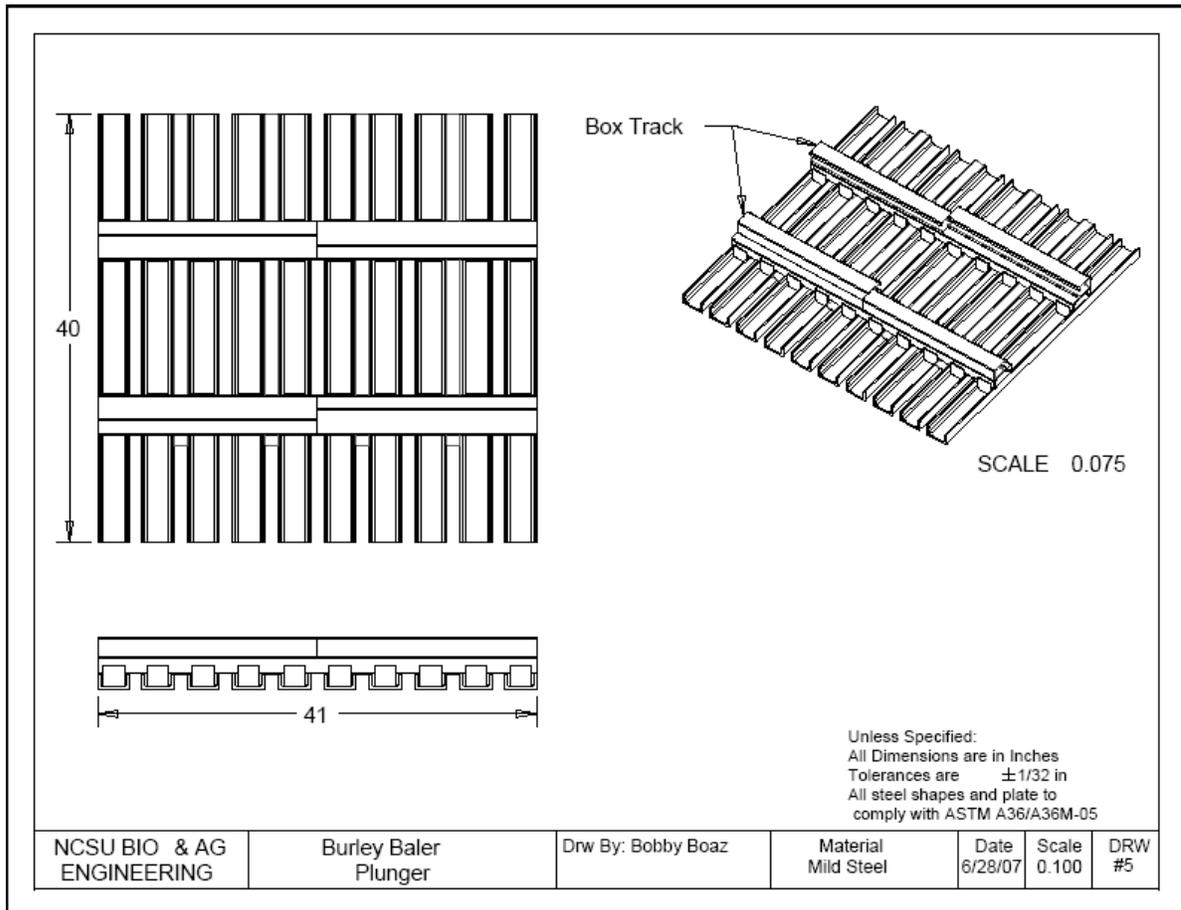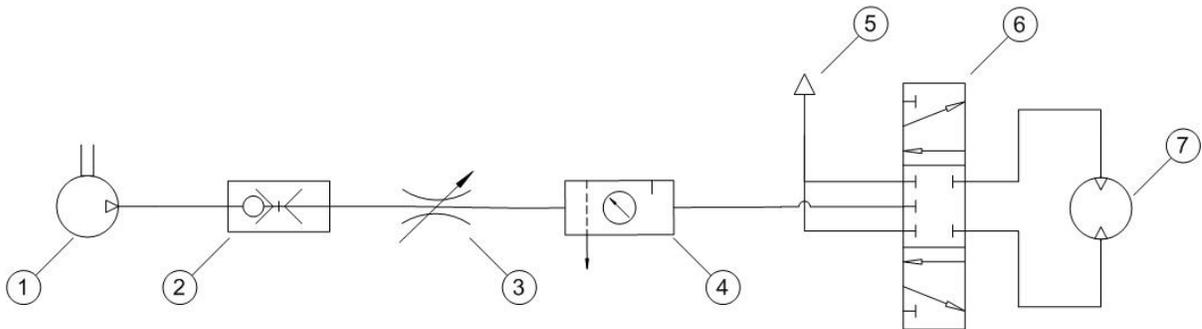# Appendix A – Tobacco Baler Detailed Drawings

Drawing #6: Pneumatic Schematic Diagram

| No. | Component |
|-----|-----------|
| 1 | Air Compressor |
| 2 | Quick Disconnect Coupling |
| 3 | Quarter Turn Shutoff Valve |
| 4 | Filter/Regulator/Lubricator |
| 5 | Exhaust Muffler |
| 6 | Directional Control Valve |
| 7 | Pneumatic Gear Motor |

# Appendix A – Tobacco Baler Detailed Drawings

Drawing #7: Electrical Diagram
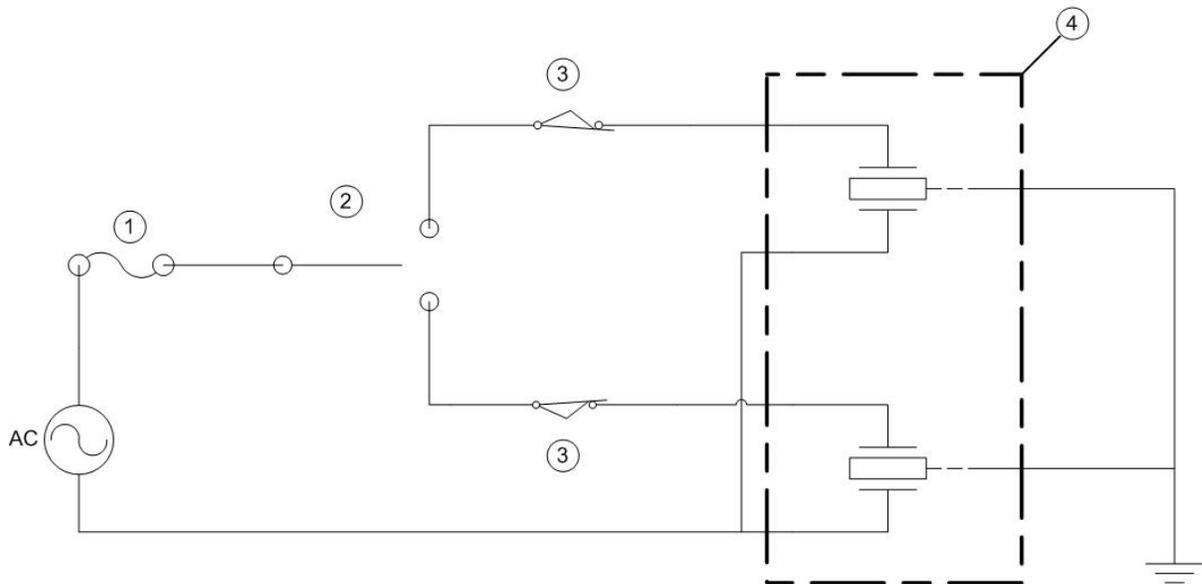


| No. | Component |
|-----|-----------|
| 1 | Panel Mount Fuse |
| 2 | SPDT On-Off-On Switch |
| 3 | N.C. Limit Switches |
| 4 | Double Solenoid Control Valve |

# Appendix B – TK Solver Worksheets

## Rules and Variables Sheet

| | |
|---|---|
| TK Solver - force_acme_rod.tkw | |
| File  Edit  Commands  Options  View  Window  Tools  Applications  Help | |

**Rules**

| Status | Rule |
|---|---|
| Satisfied | T=(F*dp/2)*((mu*pi()*dp+L*cosd(a))/(pi()*dp*cosd(a)-mu*L))+mu*F*dc/2 |
| | |

**Variables**

| Status | Input | Name | Output | Unit | Comment |
|---|---|---|---|---|---|
| | 630 | T | | in-lbs | torque: motor max output is 630 in-lbs |
| | | F | 1916.399 | lbs | axial load: 10,000 lb vertical = 1717 lb horizontal. Initial guess required. |
| | .9 | dp | | in | pitch diameter of 1" rod = 0.9 |
| | .25 | mu | | | coefficient of friction: steel acme nut on steel acme rod, dry sliding contact = .15–.25 |
| | .167 | L | | | lead: equals pitch for single start; 6 threads per inch = 0.167 thread pitch |
| | 1.47 | dc | | in | mean collar diameter: average of 2 measurements of the dia. at right angles. (1.56+1.38)/2=1.47 |
| | 14.5 | a | | | radial angle of the thread: always 14.5 for acme threads. |

(2s) Status:                                                                 OK  /701M    122M    5:48 PM

# Appendix C – LabVIEW Program

Front Panel
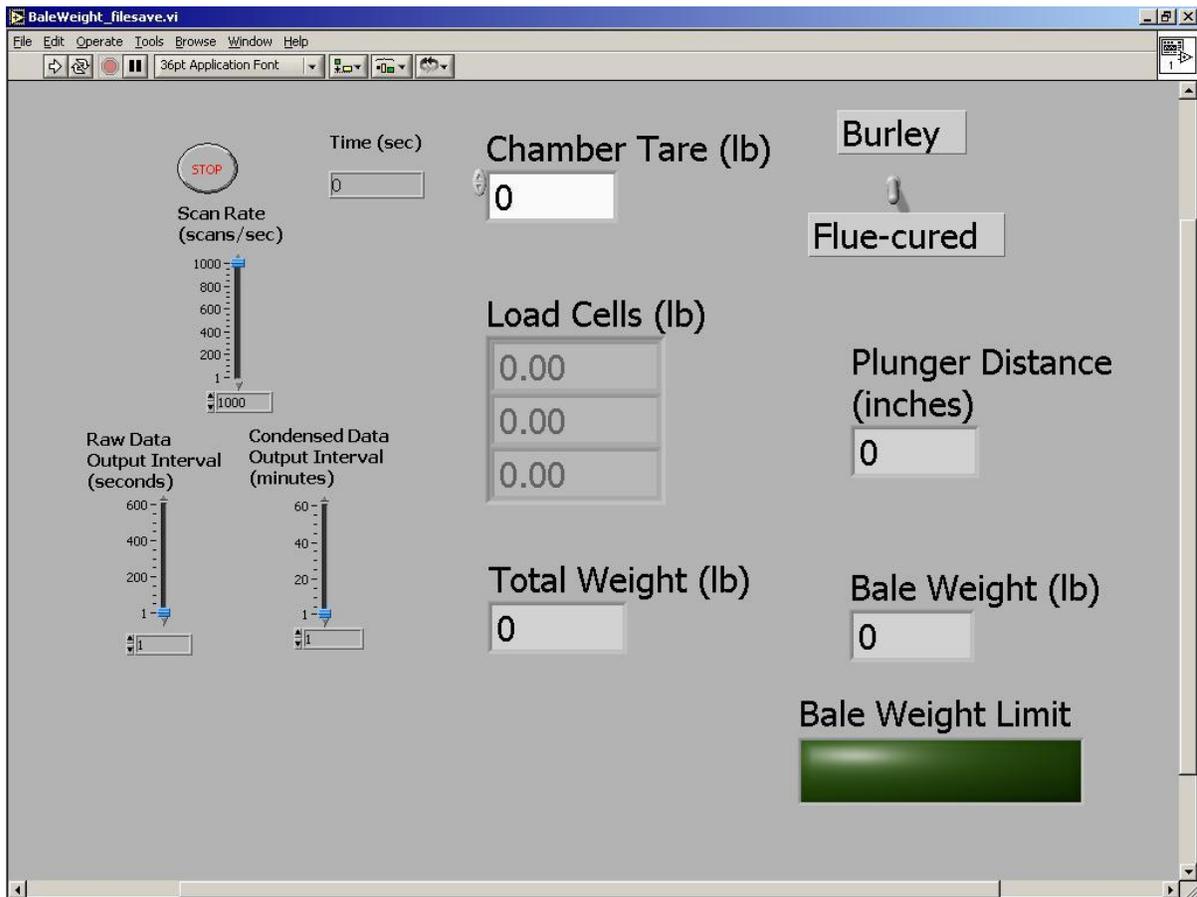
# Appendix C – LabVIEW Program

## Block Diagram