A New Concept in Side Delivery Rakes For Forage Crops

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Title: A New Concept in Side Delivery Rakes for Forage Crops

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Foreward

This design paper is a senior engineering project paper done by the authors during the 1995-96 academic year. The advisor of the project was Professor E.R. Norris, who helped inspire the idea. Research was examined on the subject, and an analysis of machines currently on the market was undertaken to arrive at the final design criteria.

The harvesting of forage crops is very important in the province of Quebec, and in North Eastern North America in general. In the province of Quebec, 68 % of the arable land is in forage crops. The beef and dairy industries play pivotal roles in the provincial economy. The climate is relatively humid, so the harvesting of the crops is sometimes difficult.

The harvesting of forage crops is not nearly as advanced as the harvesting of grain. Many losses are incurred during the harvesting process. For this reason, there is great potential for the development of new machinery to harvest forage crops. The purpose of this project was to develop a machine that could increase the harvesting efficiency of forage crops.

A New Concept In Side Delivery Rakes For Forage Crops

by

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&

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Executive Summary

The purpose of this project was to propose a new design of rake for forage crops. The main objective was to develop a machine whose use would increase the field harvesting efficiency. Field losses consist of machine losses, plant respiration losses, and losses caused by rain.

The new rake design possesses two characteristics that should serve to increase harvesting efficiency of forage crops. To reduce leaf losses caused during raking, a new raking operation was developed. The initial concepts for the operation were taken from a machine that was developed in the 1950's that achieved little success. Theory concerning rake parameters was adopted to the design. It results in a machine with a very smooth action that could have a relatively high operating speed.

The main component of the machine is a 22 inch wide flat belt. Similar to the way pickups on modern combines are made, tines are bolted to the belt. Three 8 inch diameter rollers hold the belt, with one of them maintaining the tension by spring loaded tighteners. The main support frame resembles one of a conventional side deliver rake. Three adjustable links connect the raking apparatus to the frame. The rear two links are attached close to the raking apparatus, and maintain the level of the rake to the ground. The front link is located farther away, and serves to adjust the tilt of the belt.

When the machine is adjusted so the front tines are significantly higher than the rear, the rear tines will be carrying most of the greener wet hay at the bottom of the swath and placing it on top of the windrow. This will increase curing rates and thus reduce plant respiration losses and losses caused by wetting from rain.

The cost of purchase and maintenance will undoubtedly be higher than other rakes. A sensitivity analysis determined that an increase in harvesting efficiency of 1-3 percent would be required to offset additional expenditures, depending opon the size of operation.

Tests would have to be done to determine if the machine functions as planned and to improve some aspects of the design.

Acknowledgements

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Introduction and Background

Literature Review

For dairy farms to maintain an acceptable level of profitability, it is important for forage crops to be harvested efficiently. Buckmaster et al (1990) quoted that during the mid 1980's, farmers in the United States produced an average of 89 million tonnes of alfalfa hay worth at least 6.1 billion dollars(US) annually. However, the harvesting of forages is often very inefficient. Typically, only 75 percent of the crop is available for animal feed (Rotz and Sprott, 1984). Therefore an average of 30 million tonnes of alfalfa crop were lost during harvest, storage, and feeding worth approximately 2.1 billion dollars (US) (Buckmaster et al, 1990).

Losses occur during the harvest, storage, and feeding processes. In cases where the hay is rewetted by rain, losses are significantly higher. In addition, the nutrient content of the losses is higher than the average nutrient content of the plant, so greater than 25 percent of the feeding value is lost. Harvest losses occur during machine operations from when the crop is standing to when it is removed from the field. The machine operations can possibly consist of mowing, tedding, inverting, raking, baling, and chopping. Many researchers have investigated the phenomena of forage losses (Buckmaster et al, 1990; Buckmaster, 1993; Dobie, 1961; Elliot, 1950; Giles and Routh, 1951; Koegel et al, 1985; Rotz and Sprott, 1984; Rotz and Savoie, 1991; Rotz and Abrams, 1988; Rotz et al, 1993; Rotz et al, 1990; Savoie, 1988; Savoie et al, 1982; Savoie and Marcoux, 1985; Shearer et al, 1992). Figure 1 illustrates the machine losses found in a study done by Rotz and Abrams (1988).

Forage crops are harvested in two different ways. They are either harvested as dry hay or as chopped haylage. Hay must be field cured to a moisture content of no more than 25 percent (wet basis) while haylage can be removed at a moisture content as high as 60 percent. In some instances, forages are directly cut and removed from the field in one operation. The high content moisture (greater than 70 percent wet basis) feed is suitable for feeding shortly after removal (i.e. 24 hours), however chemicals must be added for preservation and there is a significant environmental problem with the large quantity of effluent produced. Rotz et al (1993) concluded that the higher production costs in a direct cut system do not offset the benefits.

When forages are harvested as dry hay or haylage, it is desired to have them dry to the desired moisture content as quickly as possible so as to reduce the chance of weather damage and to reduce the losses caused by plant respiration. Forage plants continue to respire during the curing process until a wet basis moisture content of 40 percent is reached. This loss accounts for 5-10 percent of crop dry matter (Rotz and Abrams, 1988).

Generally the crop is cut with a mower-conditioner and deposited in a wide swath or narrow windrow. Crops that are left in windrow formation can be removed from the field with a forage chopper or baler without any intermediate crop manipulations. However windrowed forages dry much slower than crops left in a swath, because of inefficient use of solar radiation and poor aeration. Forages left in swaths require manipulations before they are removed from the field. Studies have been conducted to determine the efficiency of drying and the costs involved with different combinations of swath manipulations (Dobie et al, 1961; Koegel et al, 1985; Rotz and Savoie, 1991; Rotz and Abrams, 1988; Savoie et al, 1992; Shearer et al, 1992). Table 1 demonstrates the effect on drying and feed cost of the different combinations of manipulations at Quebec, Canada.

Studies by Rotz and Savoie (1991) indicate that the most cost effective method of curing hay in Quebec is to leave it in swath formation and rake once. To achieve minimum curing time, a combination of raking and tedding or inverting is needed. The economic efficiency of the swath manipulations involving raking could have been increased if a tandem rake setup (handle two swaths at once, figure 3) had been used. Windrow inverters can not be doubled to handle two swaths with one pass (figure 6).

Raking can be done to place forage into a windrow or to invert an existing windrow to increase the rate of drying. Dry matter losses from raking can range from 1 percent for flipping a previously formed windrow to 48 percent for a swath that is raked when it has reached 10-15 percent moisture content (figure 2). An average dry matter loss of 5 % is considered normal. Losses from raking are the most significant caused by a machine during the harvesting process. Buckmaster (1993) stated that the quantity of dry matter losses was dependent on rake type, yield and moisture content of the crop. Crop yield is dependent on agronomic practices and weather, and moisture content at which raking is done is dependent on management, and are both beyond the control of the engineer. The only factor that can be controlled by the engineer is the rake type, or the design of the machine.

Presently three different types of rakes are commonly used in North America. These are of the parallel bar rake(or oblique reel head, figure 3), the wheel rake(figure 4) and the rotary rake(figure 5). Rake design factors that affect leaf loss are the total distance the rake moves the crop, the speed at which it moves the crop, and the number of times and magnitude of accelerations and decelerations (Giles and Routh, 1951). In a parallel bar rake the number of bars on the rake could also affect leaf loss.

Different researchers have performed theoretical analyses of motion of side delivery rakes to relate rake design to total dry matter lost (Bainer, 1951; Elliot, 1950; Giles and Routh; 1951, Richey, 1943). The theoretical total distance that the hay is moved during the raking process can be found by adding up the vectors of forward displacement and rake displacement. Bainer (1951) analyzed five different models of rakes. The wheel rake usually possesses an advantage over the parallel bar rake in that the total distance that the crop is moved is smaller, which translates into less leaf loss. Figures 9a-d illustrates some the vector diagrams presented in the analysis by Bainer. In figures 9a-d, tooth path is analogous to the distance the crop is moved, as forward motion is to forward displacement and reel component is to rake displacement. In observing these figures, it is evident that the tooth component for a wheel rake (figure 9b) is smaller than the tooth component for a parallel bar rake (figure 9a)

Recently, the rotary rake has gained a lot of popularity. Research done on rotary rakes is sparse, however Savoie et al (1982) did compare the effects of a rotary rake with a parallel bar rake. No analysis of motion has been performed on a rotary rake likely because of the simplicity of its motion. The shortest distance that the crop can be moved is $\pi d/2$ where d is the diameter of the path the tines follow. The total distance that the crop is moved increases with increased forward speed. With a rotary rake the hay is moved very fast, is under constant angular acceleration, and initial and final magnitudes of tangential acceleration are very large. These factors would seem to induce a high quantity of dry matter loss. Tests by Savoie et al (1982) indicated that the use of a rotary rake resulted in higher dry matter losses than did the use of a parallel bar rake. The advantage of the rotary windrower was that it generally produced a drier windrow.

In the analysis carried out by Bainer (1951), several different rakes were analyzed. The rakes analyzed used four different working principles. The only ones to become popular were the wheel rake and the parallel bar rake. One of the rakes that did not achieve wide acceptance was the Curry rake (figure 8). This rake was mounted to the front of a tractor and consisted of a pair of chains with 26 inch cross pieces connected between the chains at 23 inch intervals. Coil spring tines are attached to the cross pieces every four inches to form a sort of drag conveyor. The rake is driven by a power take off and operated at right angles to the direction of travel. While the machine appears cumbersome and has many moving parts, it appears that it would produce a low density, untwisted windrow which seems to be one of the factors making the rotary rake popular with farmers.

The calculated length that the hay travels when forming a seven foot windrow with the Curry Rake swath was 11.1 feet. The vector diagram for the Curry rake is presented in figure 9c. It was pointed out that by orienting the rake at a rearward angle and correctly correlating the conveyor speed, it would be possible for the hay to move a distance of seven feet when a seven foot swath is raked. The vector diagram for such a machine is presented in figure 9d. Other attractive features of such a rake would be that the crop will only be engaged by the teeth once and thus will be accelerated only once and decelerated once.

Machine Analysis

Parallel bar rakes have attained a great deal of popularity in the last thirty or so years. It is evident that the dependability, ease of operation, ability to adapt to a tandem rake setup, and speed of operation have made this rake attractive. For this project several different models of parallel bar rakes were examined. Rakes manufactured over the last 20 years by New Holland (figure 3), International, John Deere, Massey Furguson, and New Idea were studied. On these rakes particular attention was paid to the drive train of the machine, the main frame used to suspend the raking basket and connect the rake to a tractor, and the mechanisms used to suspend the raking mechanism from the main frame. Some of the rakes were ground driven, some by power take off, and some hydraulically. To maintain an exact ratio of raking speed to ground speed it would be desirable to use a ground drive.

The parallel bar rakes consist of the raking basket, a main frame connecting the rake to the tractor and supporting the raking basket, a means of suspending the basket from the frame, and a means by which power is transferred to the reel of the rake. On New Holland ground driven models, power is transferred from the wheels by a 1 1/4 inch square extendible shaft to a gearbox. In the gearbox the shaft speed is increased by a ratio of 1.55:1 by means of a pair of bevel gears at 90 degrees. A spring loaded interlocking clutch on the main gear serves to disengage the raking operation during transport. A short shaft from the pinion gear drives the reel on the rake. Both wheels are used to drive the rake. The wheels are connected by a shaft and two universal joints. Each wheel has an overriding clutch so it does not skid on corners when the other wheel is turning slower. A main frame made of 4 inch channel beams connects the driving wheels to the tractor. The raking basket is connected to the main frame by three main suspending links. A fourth link stabilizes the basket. The back two links serve to adjust the height of the raking basket while the link in front of the basket controls the tilt or pitch of the raking basket A labeled figure of a side delivery rake is illustrated in figure 8.

Rotary rakes and wheel rakes were also studied. On a wheel rake, a set of finger wheels are placed on a simple frame at an angle to the direction of travel (see figure 4). The wheels have radial fingers which contact the ground, causing the wheels to turn. The turning wheels move the crop into a windrow.

Rotary rakes are driven by a power-take-off. Power is transferred from the power-take-off shaft to a rotor by a set of bevel gears with an approximate gear ratio of 6:1. Radial tine arms are connected to the rotor. Models by New Holland, Miller Pro, and Khun were studied. New Holland Rotary rakes offer the advantage of being able to ted the crop as well as rake it (see figure 5). Some rotary rakes have been built which can handle two swaths at once. These machines are very large and have a large mass. Many attach to a three point hitch, and a heavy tractor is needed so the front end of the tractor does not lift off the ground when the rake is being transported.

Project Objectives

The objective of this project is to develop a preliminary design of a side delivery rake the use of which would increase the overall harvesting efficiency of forage crops. Primarily, this is to be accomplished by minimizing dry matter losses. A secondary objective to be incorporated into the design is to have the rake produce the best windrow for drying. An increase in the windrow drying rate would reduce losses caused by rain wetting and by plant respiration.

To attain these objectives, information from previous studies will be analyzed. Mechanisms will also be studied on existing rakes. The new design will incorporate information from previous studies into the better components of existing machines. The scope of this project will be limited to the analysis and design of the mechanism used to place the crop into a windrow. Components serving to support and drive the raking mechanism will be similar to those used on existing machines.

Final Design

Design Guidelines

Information examined suggests that a good rake design must minimize the distance the crop is moved in order to form the windrow, and minimize crop acceleration. Thus impacts upon the crop should also be minimized. If possible, it seems desirable to produce a windrow similar in consistency to one created by a rotary rake. The rotary rake produces a light, fluffy windrow while other rakes twist the crop into a denser mass. A lighter windrow would seem to have benefits over the tightly wound one in that the hay in the windrow would, under similar conditions, cure more quickly.

From a theoretical point of view, the wheel rake is currently the best machine because of the short distance the crop is moved and the small number of impacts imparted to the crop during the raking process. In the analysis by Bainer (1951) it was established that it may be possible to improve on this by modifying the Curry rake (figure 7). A correct correlation of forward speed with raking speed and the angle at which the rake moves with respect to the direction of travel could yield an ideal raking distance (the shortest possible distance to move the crop in a windrow). With the Curry rake, the drag conveyor was placed perpendicular to the direction of travel. In this situation, it is impossible to have an ideal raking distance, because the rake would have to be driven at infinite speed. Downfalls of the Curry design were that the chains could break and lead to a major inconvenience in the field, and many moving parts make the design complicated. It was directly mounted on the tractor which would cause it to be a nuisance to connect and disconnect.

The basis of the design for a new rake will be to use the basic structure of a parallel bar rake (see labeled photo in figure 8). The raking basket is removed and replaced with a drag conveyor with a drive speed ratio correlated to forward velocity and angle between the conveyor to the direction of travel. The relationship between these parameters is illustrated in figure 10. In the new rake the drag conveyor will have a similar appearance to that of a Curry, but will be constructed of a material comparable to the rubber aprons used on combine pick-ups. This design will minimize the number of parts used on the rake. The conveyor will resemble a large flat belt (22 inches wide) supported by three 8 inch diameter rollers(see figures A10-A15). Rake tines will be connected to the belt (figure A8a-b,9). A metal frame will be inserted on the inside of the belt to support the rollers and to attach suspension members from the main frame. The driving mechanism will be essentially the same as the one used on the ground driven New Holland parallel bar rakes (figure 8, A17a-b). The exception will be that the gear ratio will be readjusted to give a correct correlation with ground speed. An overhead main frame similar to the one used on all parallel bar rakes will be designed to fit over the raking apparatus (figure 8). The mechanism to suspend the raking apparatus from the main frame will be adapted from the New Idea side delivery rake, as it appears that it will suit the design the best. As in other parallel bar rakes the rear two suspension links will control the height of the rake relative to the ground . Height adjustment will be made from a simple crank. The suspension components are presented in figures A24 - A25. A vertical crank fixed to the main frame will make the position of the front link adjustable. The position of attachment of the front link to the frame will adjust the tilt of the raking apparatus. A large tilt angle will result in the front tines being higher than the rear tines. The advantage of this configuration could be that the front tines will sweep the top portion of the swath into the windrow first, and then the rear tines will rake the wetter green hay on the bottom of the swath and deposit it on the top of the windrow. This action could result in crop drying characteristics superior to all side delivery rakes currently being manufactured.

The design life of the machine components is 15 years, with annual use being estimated at 500 acres. An increase in annual use would correspondingly reduce the design life.

Design Specifications

Capacity

To begin the process of designing the rake, it was first necessary to determine the overall size and capacity of the machine. Most side delivery rakes currently sold are able to handle 9 1/2 - 10 feet in one pass. It was desirable to keep the capacity similar to other rakes currently available. The ratio of rake speed to forward speed and angle of the raking apparatus to the direction of travel were decided on using figure 11. There is obviously a practical trade-off between rake angle and the ratio of forward speed to hay speed. These parameters are illustrated on a vector diagram in figure 10. From the graph it was decided that a realistic value for the angle of the rake would be 45 degrees and thus the ratio of hay speed to forward speed would be 1:1 and the ratio of rake speed to forward speed would be $2^{0.5}$:1.

Rake Tines

From other machines, it was observed that one tine could practically handle a width of four inches. On the rake, double tines will be used. A double tine consists of two separate tines four inches apart connected by coil springs at the base. Figure A8a-b demonstrates the configuration of the double tine to be used, along with the dimensions. Figure A9 demonstrates the arrangement of tines on the belt. Tines will be bolted to the belt. Across the belt, tines will be 4 inches apart, or double tines will be eight inches on center, in a staggered arrangement. Along the belt there will be 18 inches between tines. According to calculation *I* in Appendix C, this arrangement will result in a coverage factor of 1.88, meaning that each piece of ground will be covered by a tine 1.88 or approximately 2 times. The belt width is 22 inches, so that there is one inch between the edge of the belt and the edge tine.

Belt

Different materials were considered for belts, however, a reinforced rubber belt will provide the characteristics necessary for this application. The belt will have one joint. There is a considerable advantage in cost with reinforced rubber as opposed to alternatives (i.e. nylon). To determine if a belt of this material would suit our design, a performance test was done to determine some material properties of the belt. It was desired to find the tension vs. elongation characteristics of the belt, along with the tension vs. deflection characteristics with different moments applied on a pin bolted on the belt. Figure 12 represents a schematic of the test setup. Tension versus elongation and pin deflection values were found for different sizes of washers used to support the pin on the belt. The specimen tested was 2 inches wide and 1/8 inches thick. Results of deflection vs. moment tests are presented in figures B1-B5. In these figures the moment values represent the estimated force of the crop on the rake tines. Tension vs. Elongation characteristics are presented in figure B6.

By initial inspection, it was evident that a thicker belt than the tested specimen would be needed for the rake. However, it is necessary to provide a minimum tension on the belt to be able to transmit the necessary power for raking. Force on the structural members increases linearly with thickness, so additional thickness would require stronger frame construction. The deflection of the belt is proportional to the thickness cubed. From an analysis of the data, a 1/4 inch belt would appear adequate. Doubling the thickness of the belt means that the deflections occurring under the application of a moment would be 1/8 of the values found in the performance test. It appears that the a washer size of one inch would provide enough support to the belt. For the design, no washer will be used on the side of the tine because the a large portion spring coil between the double tines will be in contact with the belt (figure A9). On the opposite side of the belt, a bolt with a narrow flat head will be used whose diameter is one inch will be used. It is necessary to minimize the thickness of the bolt head so that when the bolt is tightened, the head of the bolt will be drawn flush with the edge of the belt, so not as to cause interference when the belt is in contact with the roller.

Tension of Belt

A value of 75 lb./in was chosen for the initial tension of the belt. The tension values change when the machine is in operation. The raking operation is performed by the tight side of the belt. This tension is calculated as 2150 lb. (97.7 lb./in) in part *II* of Appendix C. The belt thickness will be doubled, so deflection/elongation characteristics will be similar to values for 50 lb./in tension. In the performance tests, all elongation values were measured relative to 50 lb./in tension. For the 1/8 inch thick belt specimen, elongation was difficult to measure until that value of tension was reached. From the data it would be logical to assume that the 1/4 inch belt would elongate no more than 0.5%. To obtain an idea of the deflection caused by the application of a moment, it would be logical to take the values obtained when testing one inch washers for 50 lb./in and divide deflection values by 8, because deflection is proportional to $(\text{thickness})^{-3}$. The expected deflections are plotted in figure B7. Belt deflections should be small.

Power Transmission

From the belt specifications determined above, it is possible to calculate the maximum power and pull that can be developed. This calculation is presented in part *III* of Appendix C . In part *IV* of Appendix 3 estimates are made for raking power requirements. The force required for raking is estimated at 60 lb. This is based on an assumption of 60 tines engaged in the raking action at one time with an average force of 1 lb. on each tine. Force requirements for raking will be many times less than the force that can be delivered by the belt (1000 lb.).

The force that can be supplied by the tires of the rake can be estimated using ASAE Agricultural Machinery Management Data D497.2, figure 1. Assuming worst field conditions would be similar to tilled soil, the estimated ratio of drawbar pull to static wheel load at 10% wheel slippage is about 0.32, and at 5% wheel slippage is 0.16. To supply the rake with enough power with only 5% wheel slippage, 375 lb. of vertical force is needed on the rear wheels. The actual force on the rear wheels is estimated to be at least 1000 lb., so the machine should easily be heavy enough, even when inefficiencies in power transmission are considered. When surges in power demand occur, increased wheel slippage will increase the power transmitted to the rake.

Frame design for Raking Apparatus

The frame built to support the three rollers is presented in figure A10-A13. In order to assure that the crop disengages easily from the tines and is not subject to a large acceleration upon release, three rollers are used to support the belt. A third one is added on the delivery side so the crop is not kicked out as it may be if only one roller was used on the delivery side. The tines will withdraw gradually from the crop as they travel between the two rollers on the delivery side of the rake. The tines have a rearward angle to facilitate release of the crop. A set of main supporting links between rollers on periphery of structure are formed of C3 x 5 channel steel. The main supports are also cross braced for strength (figure A14a-b). Bearings used to support the rollers are mounted on metal plates installed between ends of the mains channels. These metal plates also serve to connect main frame members.

Each roller is made of 3/16 inch mild steel, 8 inches in outside diameter and 22.5 inches in width (figure A15). Through the central axis of the roller, a 1.3125 inch shaft is installed 24 inches in length. The shaft is supported by flange bearings at each end. Calculations necessary for choosing bearings are presented in parts *V* and *VI* of Appendix C. The design life the machine is assumed to be 15 years and estimated yearly use is 500 acres. All six bearings used will be the same. The total force acting on each bearing can be approximated by $2F(\cos((180-\phi)/2)/2$, where F represents the initial tension force on the belt and ϕ is the angle of belt contact on the roller. The bearing force will be largest on the roller on the pick-up side of the machine, due to the large angle of contact. This was the force used for bearing design. Bearing specifications are given in figure B10.

To insure that the crop is released properly, the tines gradually disappear through medal slats somewhat similar to a mechanism used on forage chopper and baler pick-ups. Figure A12 illustrates the location and design of these slats.

Belt Tightener

All belts stretch with age. For this reason, a mechanism was needed to hold a certain level of tension on the belt. It is possible attach two of the three roller shafts solidly to the frame of the raking apparatus. The two rollers attached directly to the frame are on the delivery side of the mechanism. The roller on the pick-up side of the rake is not attached directly to the frame. It is connected via a spring loaded tightener. For an illustration see figure A16. Various levels of tension can be set on the belt by adjusting the force on of the spring by changing its length. The spring was designed to support the same force as the bearings. The spring specifications are calculated in part *VII* of Appendix C. The variable adjustment is provided by means of a sliding plate. (figure A16a-b) . The bearings for the roller are mounted at ends of narrow rectangular plates and slide in a guide track over another plate on the frame of the raking mechanism.

The sliding plate is connected to the rest of the frame by a threaded rod. The spring is installed around this rod (figure A16a-b). The thread strength of the nuts on the rod are calculated in part *VIII* of Appendix C. The factor of safety for the shearing of the threads on the nut is very high. Although it is not necessary to have a safety factor this large, it should be twice the rod safety factor because some thread deformation could have occurred in the nut, and all of the threads in the nut will not be supporting the load. Buckling calculations for the threaded rod are made in part *IX* of Appendix C. The maximum free length of the rod is estimated to be 6 in, which is the maximum length that could exist between the spring and the bracket on the sliding plate. The portion of the rod inside the spring will not be subject to any significant loading.

Drive train

As stated in the objectives, the scope of the project did not include designing a complete means of power transmission for the rake. Nevertheless, these components are presented in the design to make it complete. The pieces are also included to demonstrate that providing the raking mechanism with a power from the wheels would not be a problem.

It is necessary to use both wheels to drive the rake to ensure continuous operation around corners. No mechanical calculations or exact specifications are determined for the driving shafts between the wheels and the gearbox. These parts are quite similar on all parallel bar rakes and the same basic mechanism will be used on this machine. The setup is illustrated in figure A17a-b. A typical setup of the driving components in a parallel bar rake is illustrated in figure B8. In the hub of each wheel there will be a clutch so wheels do not skid on corners when not turning at equal speeds. The wheels are connected by means of a shaft with universal joints at bends. The raking mechanism must be driven by the highest roller on the delivery side so maximum power can be transferred to the belt, and so the tight side of the belt is performing the raking operation.

An extendible shaft from the wheel on the delivery side connects to the gearbox, which is mounted on the end of the high roller shaft on the rear delivery side of the rake. The gearbox is illustrated in figure A18a-b. A typical setup for a side delivery rake is illustrated in figure B9. In the gearbox, a set of bevel gears is used. As previously mentioned, the gear ratio will be 4.6:1. The gear will have 55 teeth, and the pinion 12. It is necessary to increase the rotational speed of the shaft to have a correct correlation between hay and forward velocities. The gear and pinion geometry and tooth strength have been calculated, and are presented in part *X* of Appendix C.

Similar gearboxes are used on all parallel bar rakes, however the gear ratio is never as large. For this reason, the bending and fatigue strength of the gear teeth have been calculated. As expected, the factor of safety in bending is large (4.4), but the factor of safety in fatigue is 1.33, for 99% reliability. Design life for fatigue calculations was taken as the same as for the bearing calculations. It is possible that the gears may fail in fatigue late in the machine's life.

Main Frame

A main support frame was designed to carry the raking mechanism, and hold the wheels of the rake and to hold the hitch for towing. Different views of the rake's main frame are presented in figures A19 and A20. Two large members are in the frame are made from 4 inch channel beams. From the top the members appear to be arranged in a "V" formation, with the point of the "V" located at the hitch, and the ends attached to the hub of each wheel (figure A20). Each 4 inch member is bent twice to fit over the raking apparatus. The members are braced together for strength. A tractor hitch similar to one normally found on implements of this type was designed for the front of the frame (figure A21a-c).

Suspension of Raking Apparatus

The mechanisms that are used to control the level of the raking apparatus with respect to the ground have been adopted from other machines. A three point attachment system has been used. Figure A22a-b illustrates the location of these points of attachment on the raking mechanism, and figure A23a-c illustrates all suspension components. There are two links supporting the rear of the apparatus, and one link supporting the front. The level of the rear can be adjusted by one crank, and the front link by another. The adjustment of these cranks not only raises and lowers the raking mechanism, but also controls the tilt angle of the raking mechanism to the ground. A rearward angle will have the front tines located farther from the ground than the rear tines. The tines on the front of the machine will first sweep the top of the swath into the windrow, and the rear tines pick up the green crop on the bottom of the swath and deposit it on top of the widow. This would reduce curing time.

The two rear suspension links are hung off short lifting arms welded to ends of a pipe. Figure A24a-b illustrates the pipe and components attached to it. This pipe is set in brackets which are bolted on the main support frame. The pipe will be required to rotate freely in these brackets, so lubrication will be needed. The design force used for each of the rear suspension links was 750 lb. It is believed that this value is quite high, and the actual force will likely be in the area of 300-500 lb. The mass of the raking mechanism is not well balanced under the main frame, and so the suspension bar on the delivery side of the machine will be required to support more than the link on the pickup side. A design load of 750 pounds was used because a moderate level of loading uncertainty in the members.

Rotating the pipe will change the position of the lifting arms welded on the ends, and the level of the raking mechanism will be changed. At the ends of the arms at the ends of the large pipe a rod is connected via a pin joint. The bottom end of this rod is fixed to the frame members of the raking apparatus. A rear suspension rod is illustrated in figure A25a-b. The large horizontal pipe will have another radial arm welded in the center of it which will serve to control the rotation of the pipe. This control arm will be connected to a crank which extends toward the front of the rake. A view of the crank and associated components is presented in figure A26a-c. The crank will be in two pieces threaded together. By turning a handle at the front of the machine, the bar is extended or contracted. The position of the part of the crank is kept constant by supporting it with a bracket (refer to figure A26a-c) so that turning the handle will control the rotation of the pipe and thus the elevation of the raking apparatus. The control arm on the pipe is designed to be shorter than the lifting arms so the number of turns of the crank to lift the rake can be reduced. No analysis was done to determine how much crank torque will be required to lift the rake. Modifications may need to be made if the amount of torque required is too great.

The rear suspension link between the lift arm and the raking mechanism will be spring loaded so that shocks and vibrations of the raking mechanism are not entirely transmitted into the main frame. Refer to figure A25a-b for an illustration of the components of the rear suspension link. The lifting arms will be connected by a pin joint to a short vertical section of pipe. The safety factor of the pins is calculated in part *XII* d of Appendix C. The short section of vertical pipe has a plate welded on the bottom. A small hole is cut in the plate so a suspension rod can pass through (figure A25a-b). A spring is inserted inside the pipe. Calculations for specifications of the springs used here are presented in part *XI* of Appendix C. The metal rod is connected to a bracket on the raking apparatus. The top part of the rod is threaded, so a nut and washer can be installed. The force of the spring between the bottom of the pipe and the washer holds the raking apparatus level. Safety factor calculations for the threads on the nuts supporting the suspension rods, the tensile strength of the rod and the pins connecting the rod to the frame of the machine are presented in part *XII* a-c of Appendix C. All of these parts are greatly overdesigned, however, these parts are relatively cheap. It is likely that there will be some shock effects and metal fatigue effects acting on these parts which are difficult to estimate.

Calculations for strength of the lift and control arms is presented in part *XII* e-f of Appendix C. The size of these links need to be relatively large due to the large moments they support.

Stresses on the horizontal supporting pipe are calculated in part *XII* -h of Appendix C. Figure A27 illustrates the nature of the stresses on the pipe. There are both torsional and bending stresses on this beam. There is a large moment due to the length of overhang of the pipe on the delivery side of the machine. It was desired to have the pipe overhang as much as possible because the raking mechanism was not balanced under the main frame. For this reason the pipe had to be quite strong.

Calculations are made for the threads in the crank in part *XII* -g of Appendix C. As in other cases, thread strength is always many times more than needed. Usually some threads are deformed so a conservative estimate would be that only one half of the threads are actually supporting the load. A calculation for the strength of the long part of the crank arm (a square tube) in buckling is presented in part *XII* - i of Appendix C.

The lone suspension link in front of the raking mechanism will be connected to the main frame far from the raking mechanism itself. The configuration of the front suspension link is illustrated in Figure A28a-c. A large piece of square tubing will be extended forward from the raking apparatus. The large tube will be connected to the frame via a screw jack similar to the mechanism used on other side delivery rakes. It is assumed that a suitable jack could be obtained from numerous manufacturers.

To insure that the rake does not move relative to the main frame during the raking operation, a sway bar has been added between the raking apparatus and main frame in the rear (figure A29a-b). This will pivot as the raking apparatus is lifted and lowered.

Economic Analysis

The impetus for designing a new side delivery rake was to increase forage harvesting efficiency. The new design may increase efficiency in two ways. The more gentle raking action will reduce the shatter losses caused by raking. Tilting the raking mechanism at a rearward angle would bring all green hay to the top, thereby increasing drying rates. Increasing the drying rate could reduce plant respiration losses and losses caused by rain damage.

At this stage of development, it is difficult to provide an exact figure representing the savings that may be obtained by using this machine. The true operating characteristics of the newly designed rake design are unknown. In any case, the increase of harvesting efficiency actually attained would be variable. Harvesting efficiency is not only a function of machine design, it is also a function of other parameters such as crop variety, crop yield, and time of raking during the curing process. For this reason, a sensitivity analysis has been done. The analysis is designed to compare the operation of the new machine to a conventional side delivery rake, under similar conditions. For the analysis, inflation and tax effects are ignored.

The results of the sensitivity analysis are presented in Appendix D. Table D1 presents the present value of the benefits based on a 15 year machine life, while table D2 presents the same information for a 12 year machine life. For the analysis, interest rate is assumed to be 10 percent and price of forages is assumed to be 75 dollars per tone dry matter. In maintaining a conservative outlook, it is estimated that the new machine would require 100 dollars a year in additional maintenance, and would have an initial purchase price of $3000 more, as compared to a conventional side delivery rake. It is assumed that all other operating parameters (i.e. fuel, labor) would be similar to those of other rakes, and would not affect the analysis. The effect of salvage value was also ignored, as it would have little effect.

Variable parameters for the analysis were total yearly crop production and the increase in harvesting efficiency resulting from the use of the new machine. It is obvious that the use of the machine would be more justified if the amount of production and the relative increase in harvesting efficiency is higher. For a relatively small farm with an annual forage harvest of 270 tonnes dry matter, an increase in harvesting efficiency of 3 percent is needed to justify the acquisition of the machine as opposed to a conventional rake with a machine life of 12 or 15 years. However, for a farm with a annual forage harvest of 630 tonnes, an increase in harvesting efficiency of only 2 percent is needed for a machine life of 12 or 15 years. It is possible that two machines may be purchased for a farm with a very large production.

If the use of such a machine would result in an average increase in harvesting efficiency of 4 percent, an average farm with a production of 450 tonnes dry matter would realize a present value benefit of over 6500 dollars if the machine life is 15 years and over 5500 dollars if the machine life is 12 years.

Discussion

Overview

The mechanism to windrow the crop on the new design was developed by adapting the use of newer materials to theory which has existed for some time. The new design presents two possible advantages over existing machines. First, the raking operation is smooth and efficient, with the crop being engaged only once and disengaged once. The design also minimizes the distance the crop is moved by correctly correlating the rake angle and belt speed with forward speed. Second, by tilting the raking apparatus, the front could be set higher than the rear tines. When in operation, the front tines should place the top of the swath into the windrow first, with the rear teeth taking the bottom of the swath and placing it on top of the windrow. This may effectively invert the windrow, and increase field curing rate relative to conventional side delivery rakes.

This machine may be needed in future years. This would result from widespread adoption of a forage crop mowing machine commonly referred to as a macerator. The macerator uses multiple sets of knurled conditioning rolls to shred forages. Conventional mower-conditioners crack stems along their length or break them at regular intervals. Maceration has been proven to increase drying rates dramatically. The only downfall is that harvest losses are very high. Shredded pieces of forage material have a tendency to fall into crop stubble. In future years machine designers may seek to develop a more gentle rake to combat this problem. This design may present a viable means to reduce losses from macerated forages.

The action of tilting the raking mechanism to place green hay on the top of the windrow and reduce this curing time may be a more important marketing advantage for manufacturers. Producers are always interested in machines that will help to reduce curing time, and reduce exposure to rain.

Drawbacks/ Possible Modifications

Despite the fact that this design appears to have some important advantages over existing designs, further modifications would have to be made before it could be marketable. However, any modifications should not significantly alter any of the mechanisms that would give it an advantage over other conventional machines.

One significant drawback of the machine is that the raking mechanism is not very well balanced on the main frame. The delivery side of the machine will be heavier than the pickup side. This is partially caused by the extra roller that is inserted on the delivery side of the machine. The driving mechanism does not allow the optimum placement of the raking apparatus under the main frame. It is not known if the degree of unbalance will be large enough to cause problems. Tines may scrape the ground on the heavy side of the machine.

There are different modifications that could be made to fix the problem. One is that the extra roller on the delivery side could be removed. This would significantly reduce the weight of the delivery side of the raking mechanism. The reason this roller was inserted was to have a more gentle crop release, as it was feared that the rapid acceleration of the end of the tine as the crop was released would increase crop leaf loss. Tests should be performed to determine if the extra roller is necessary. A simple but less elegant solution may be to attach a gauge wheel on the frame to the raking apparatus on the front of the delivery side. The wheel could be installed so it does not run on the swath being raked. A third solution may be to redesign the means of power transmission so the center of gravity of the raking apparatus is better placed under the main frame.

The rake may be too large. The main frame of the new raking apparatus may not suit the this design. As the machine is currently set up, the two large channel beams in the main frame span a large distance. Perhaps the frame could be designed so the wheel on the delivery side of the machine is located in front of the raking apparatus as opposed to behind it. In this case the driveshaft between the wheels would have to run through the frame of the raking apparatus. Another option may be to use a large "gooseneck" pole as is used on many large mower conditioners. For this setup one large steel member would connect to the tractor from the rear of the rake. Another modification that could be made to reduce the size is to increase the rake angle. The main drawback of this is that hay speed will be increased, as will the magnitude of the impact of the tines on the crop. An angle of 60 degrees between the belt and the direction of travel will reduce the conveyor length by approximately 22 percent. If the angle of the rake was changed without increasing the gear ratio, the distance traveled by the hay would increase by 25 percent. This is significant.

Other small modifications could be made to the machine. Hydraulic cylinders could be used to adjust machine height instead of cranks. This would increase the cost of the machine. The number of tines on the belt and the size of the belt could also be changed, but the chosen design should be adequate.

Conclusions

1. A new design of side delivery rake was developed. The advantages of the new design are a potential reduction in dry matter losses and a reduction in curing time.
2. The new design of rake would move the crop shorter distances and impact it fewer times than existing machines.
3. The crop could be inverted as well as windrowed by tilting the raking mechanism. The front rake tines may place the crop from the top of the swath on the bottom the windrow. The rear rake tines may place the crop on the bottom of the swath on the top of the windrow.
4. For most farms, an increase of harvesting efficiency of 1-3 percent would be needed to offset an estimated higher initial cost and increased yearly maintenance costs. This was determined with a sensitivity analysis.
5. Before building the designed machine further modifications should be made. The design of the main frame and power train was not within the true scope of this project. They are included but improvements should be made, chiefly to improve the balance of the raking apparatus on the main frame.

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Machine Treatment	Feild Curing Time (days)		Hay Quality	Production Cost
	Cut 1	*Cut 2*	*% Crude Protein*	*$/t Dry Matter*
Narrow swath, no manipulaiton	8.2	6.6	18.7	112
Wide swath, ted after rain, raked	6.3	6.1	18.8	103
Wide swath, raked	6.5	6.3	18.9	102
Wide swath. ted after first day, raked	6.1	6.1	18.7	106
Wide swath, ted after second day, raked	6.0	6.1	18.7	107
Wide swath, ted twice, raked	5.9	6.1	18.5	110
Narrow swath, invert	8.0	6.4	18.6	124
Wide swath, invert narrow	6.5	6.3	18.9	103
Wide swath, invert, rake	6.4	5.8	19.0	112
Wide swath, invert after rain, invert narrow	6.4	5.9	19.0	109
Wide swath, invert, invert narrow	6.4	5.8	18.9	111
Wide swath, invert twice, invert narrow	6.3	5.7	18.8	115

Table 1. Effects of Different Combinations of Swath Manipulation on Curing Time, Hay Quality, and Cost of Production of the Crop

Data Taken From Rotz and Savoie (1991)

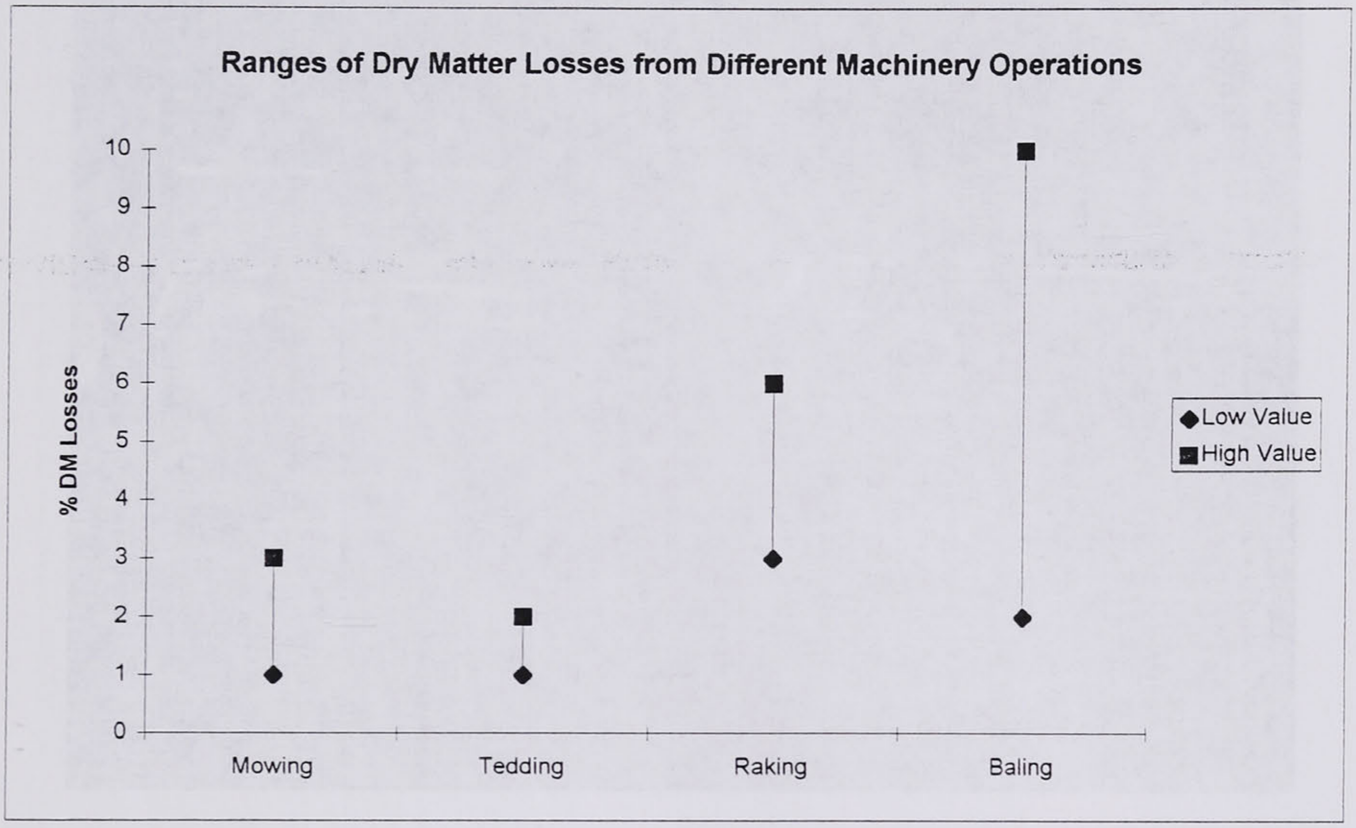


Figure 1. Dry Matter Losses For Different Machinery Operations
Data Taken From Rotz and Abrams (1988)

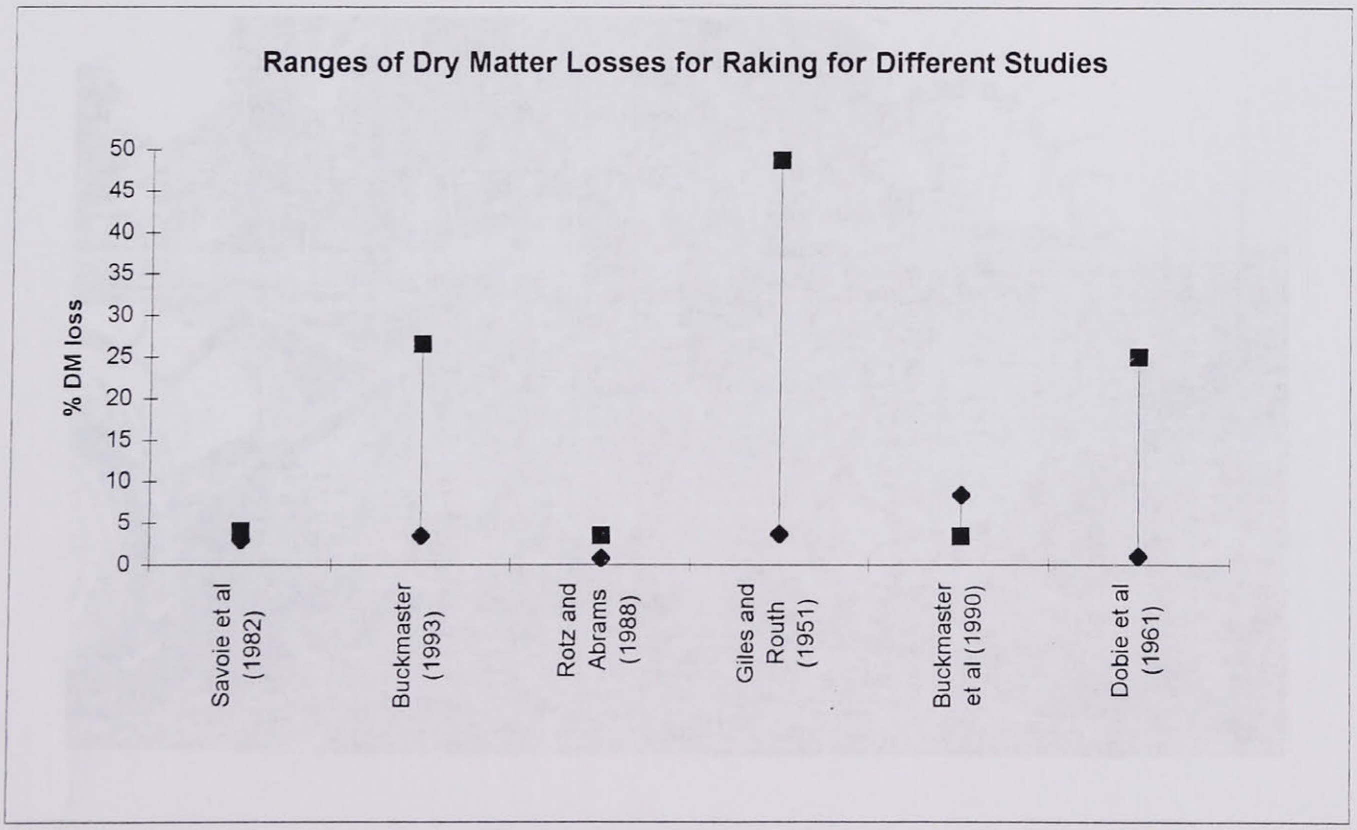


Figure 2. Raking Dry Matter Losses From Different Studies

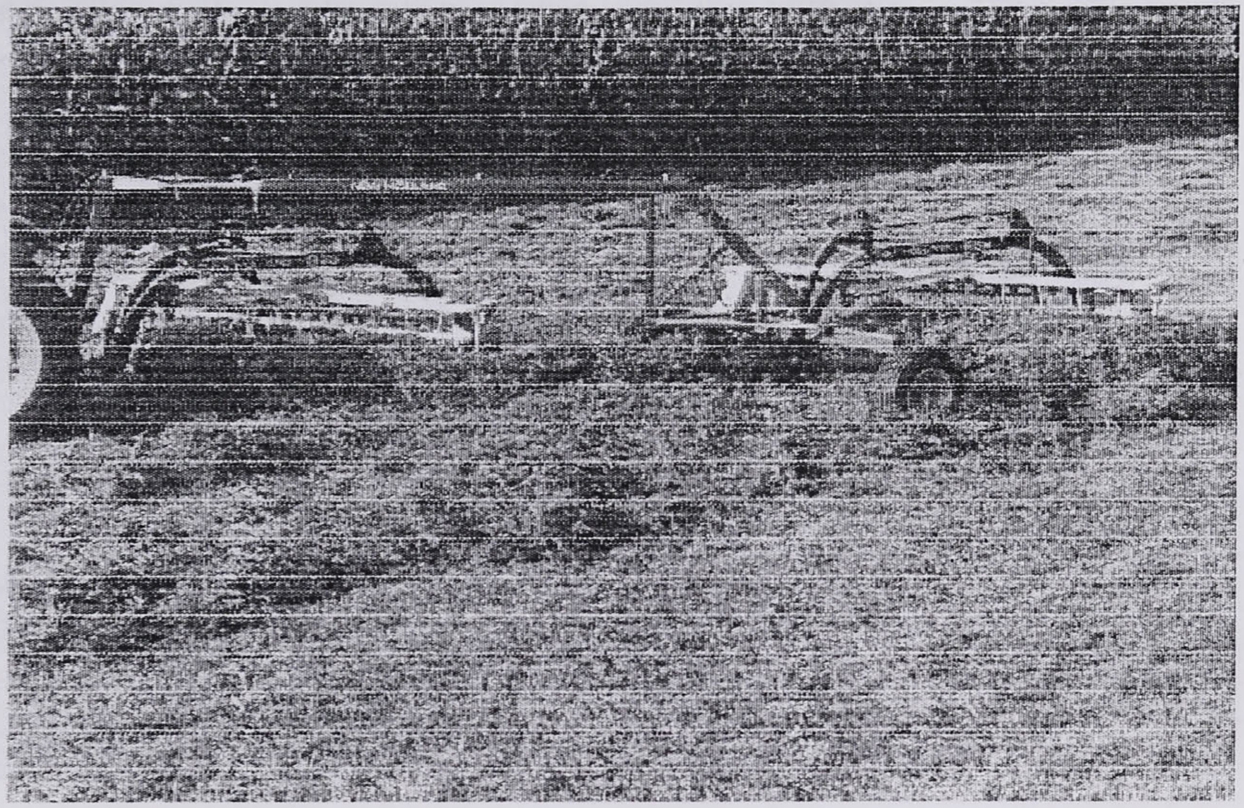


Figure 3. Parallel Bar Rakes in Tandem Setup
New Holland Limited

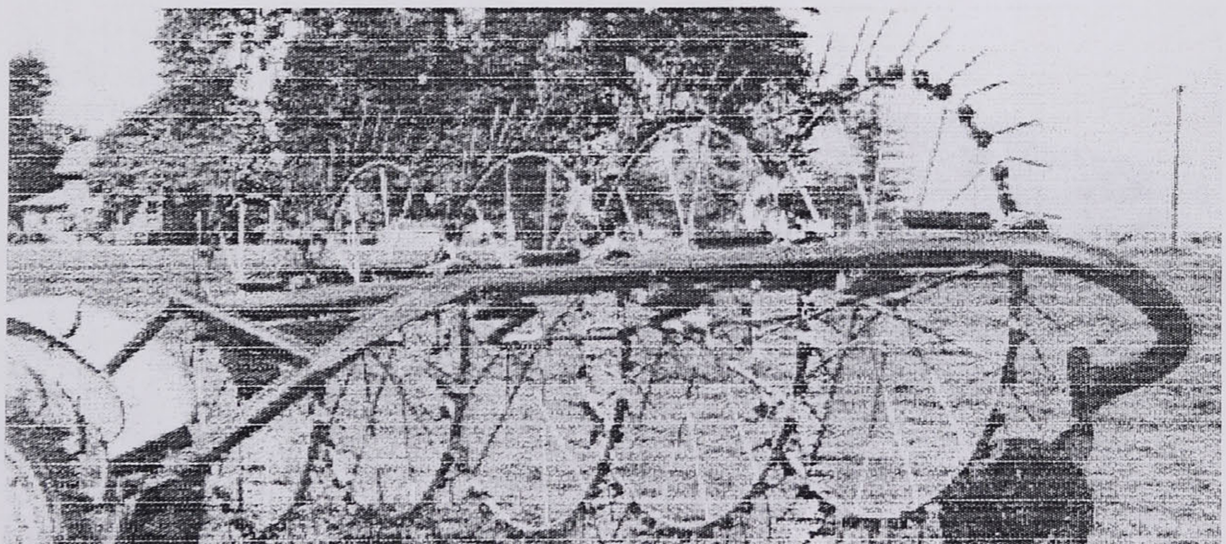


Figure 4. Wheel Rake
Stone and Gulvin (1977)

Figure 5. Rotary Rake
New Holland Limited

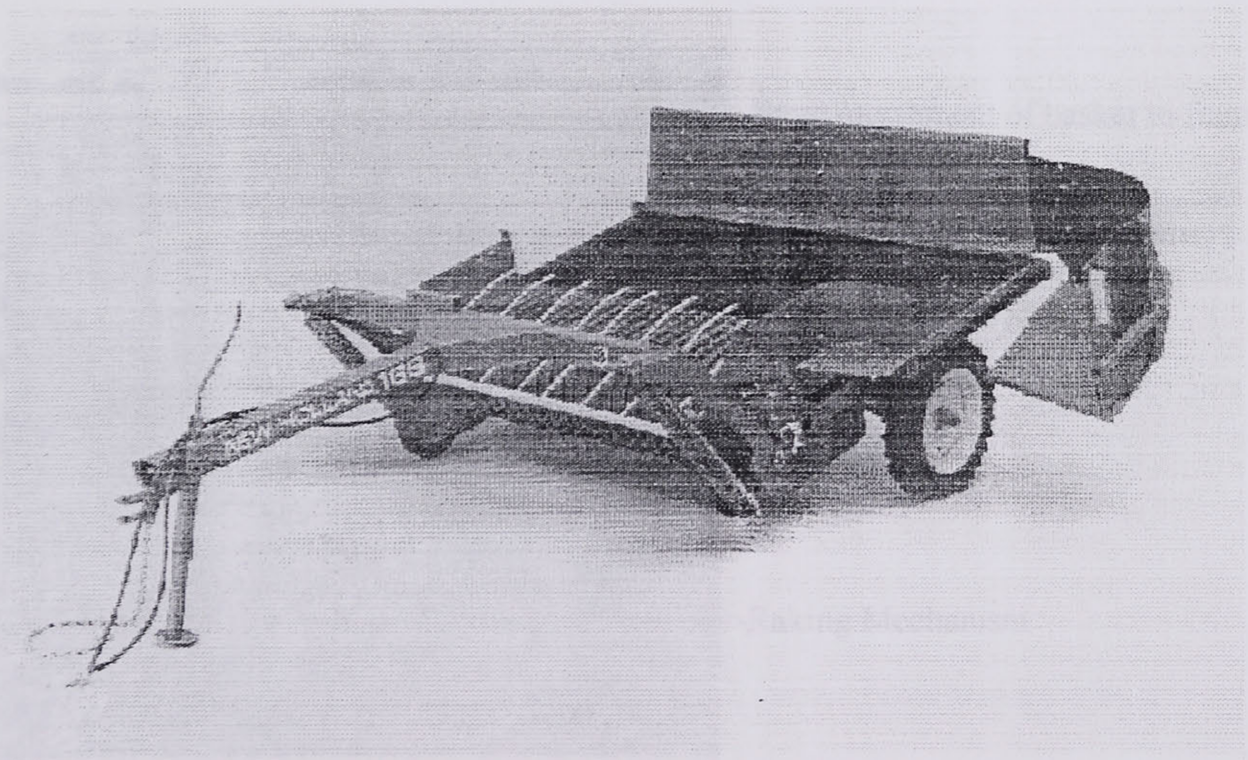


Figure 6. Windrow Inverter
New Holland Limited

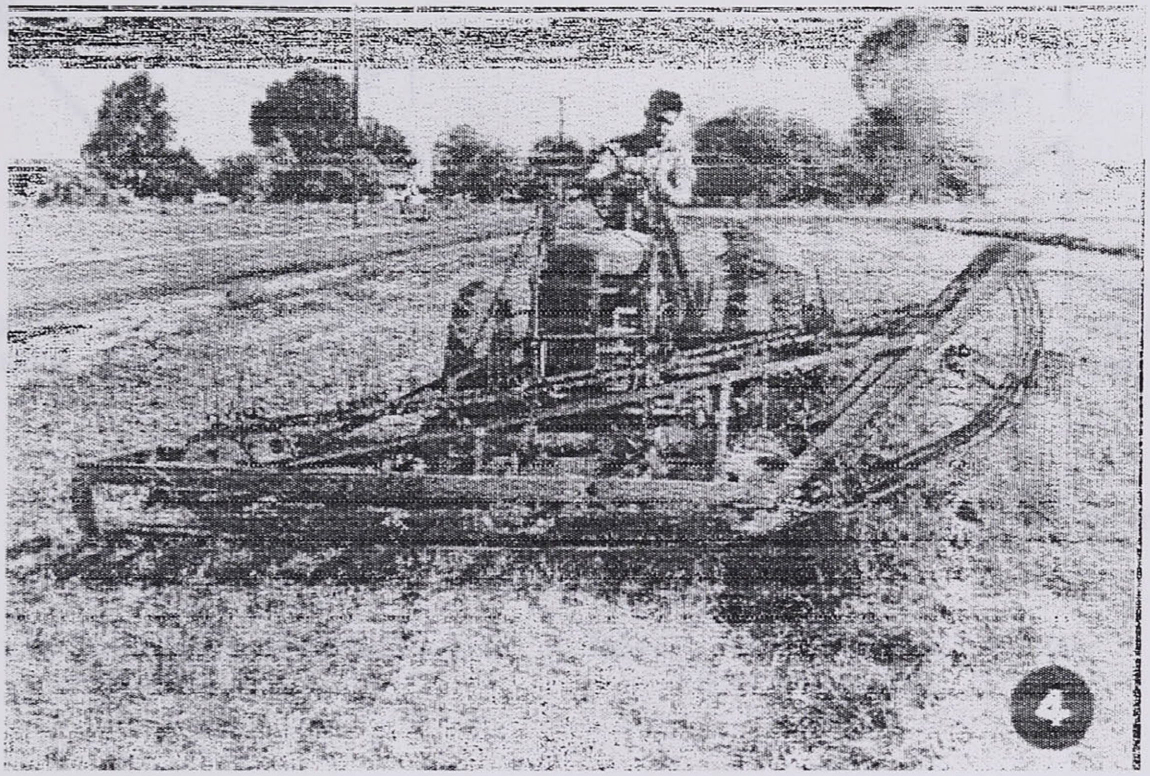


Figure 7. Curry Rake
Taken From Bainer (1951)

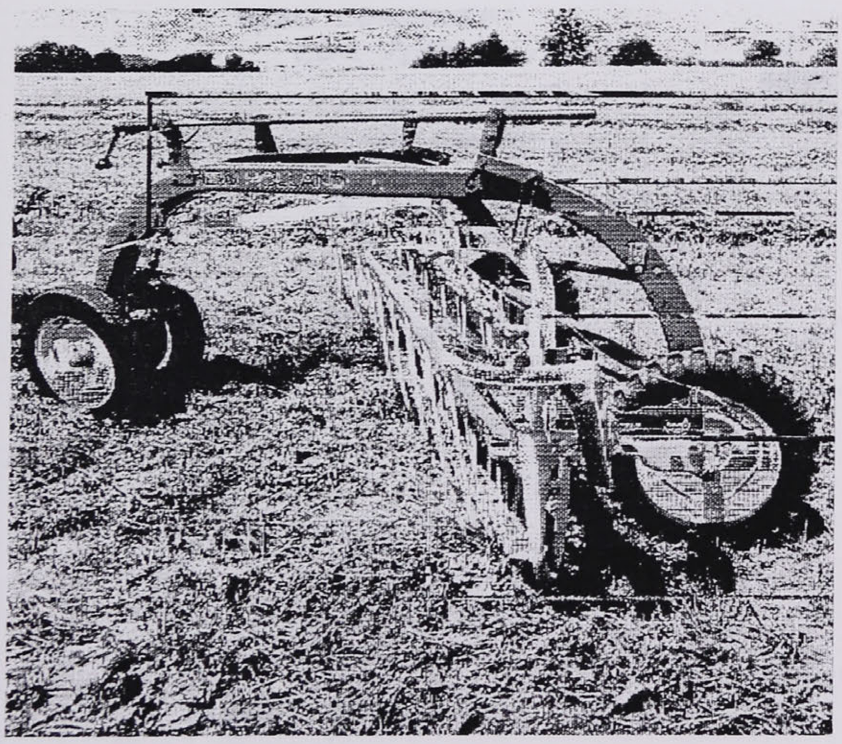


Figure 8. Side View & Description of New Holland Side Delivery Rake
New Holland Limited

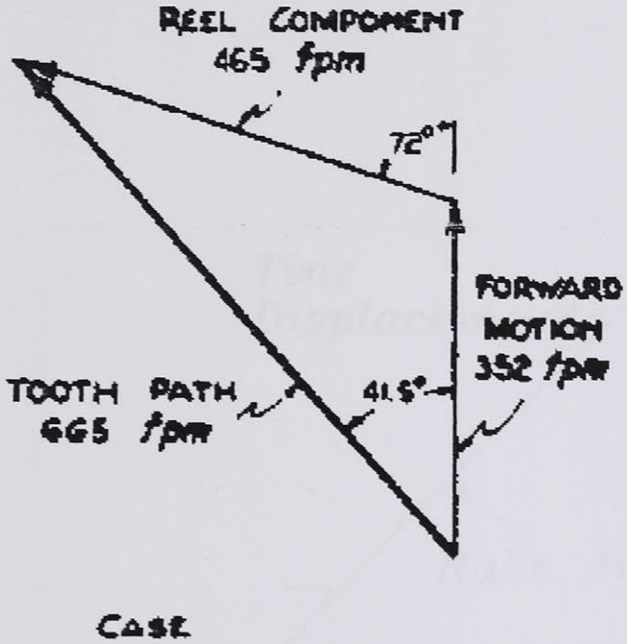


Figure 9a. Vector Analysis of Parallel Bar Rake
From Bainer (1951)

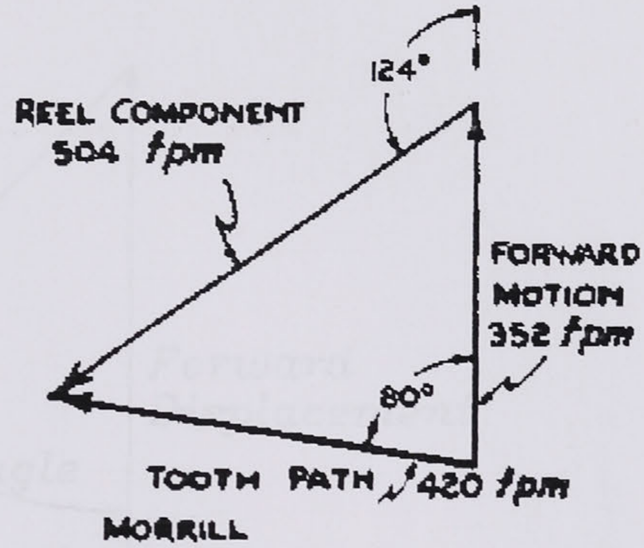


Figure 9b. Vector Analysis of Wheel Rake
From Bainer (1951)

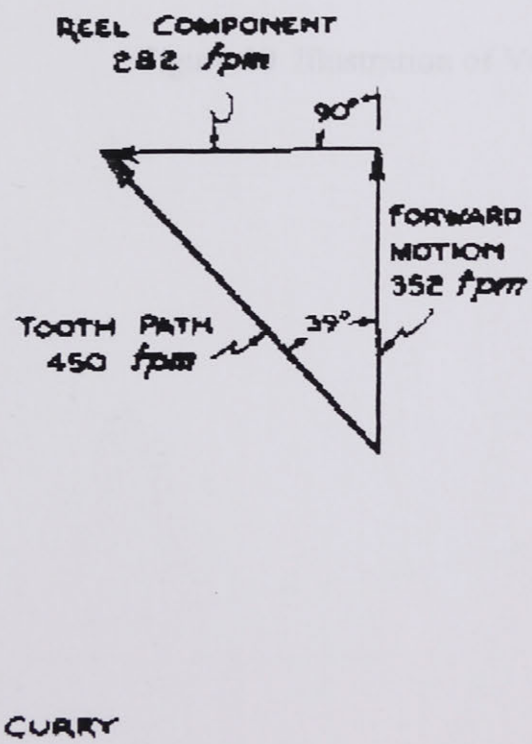


Figure 9c. Vector Analysis of Curry Rake
From Bainer (1951)

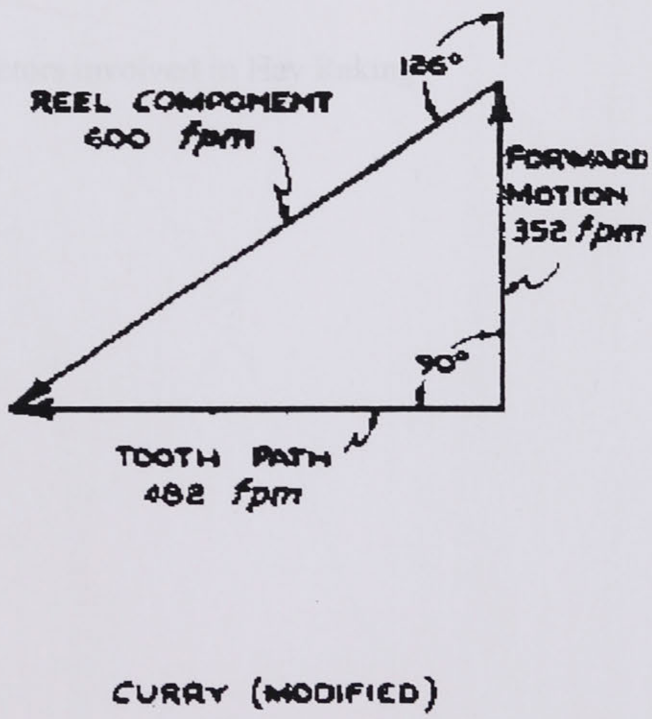


Figure 9d. Proposed Modifications to Curry Rake
From Bainer (1951)

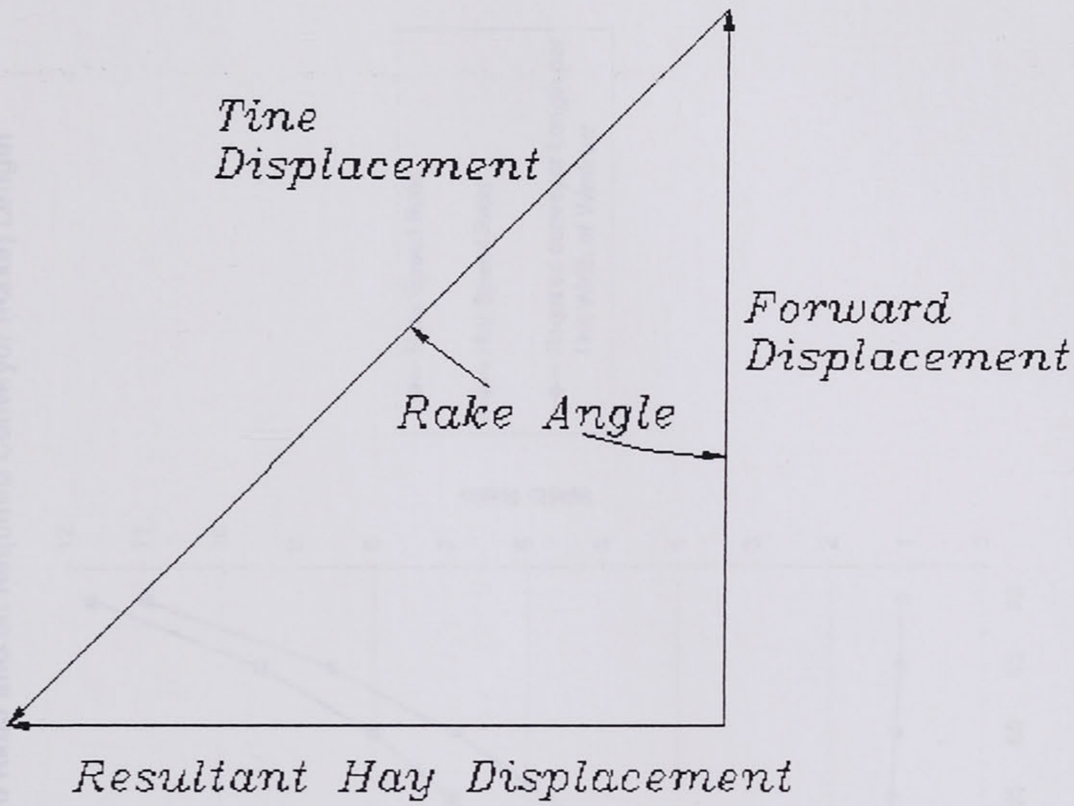


Figure 10 Illustration of Vectors involved in Hay Raking

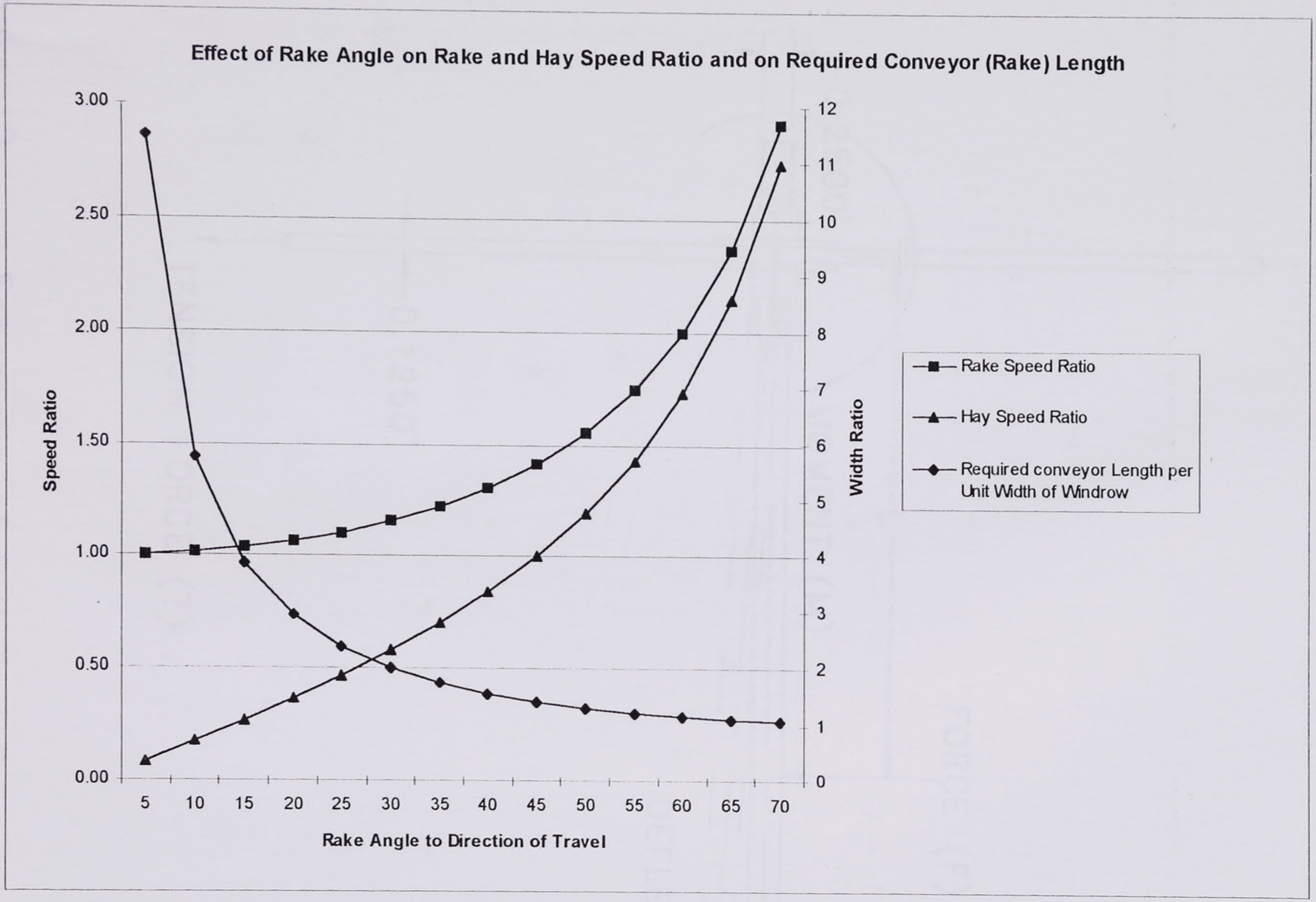


Figure 11. Correlation between Rake Angle and the ratios of Rake Speed and Hay Speed to Forward Speed

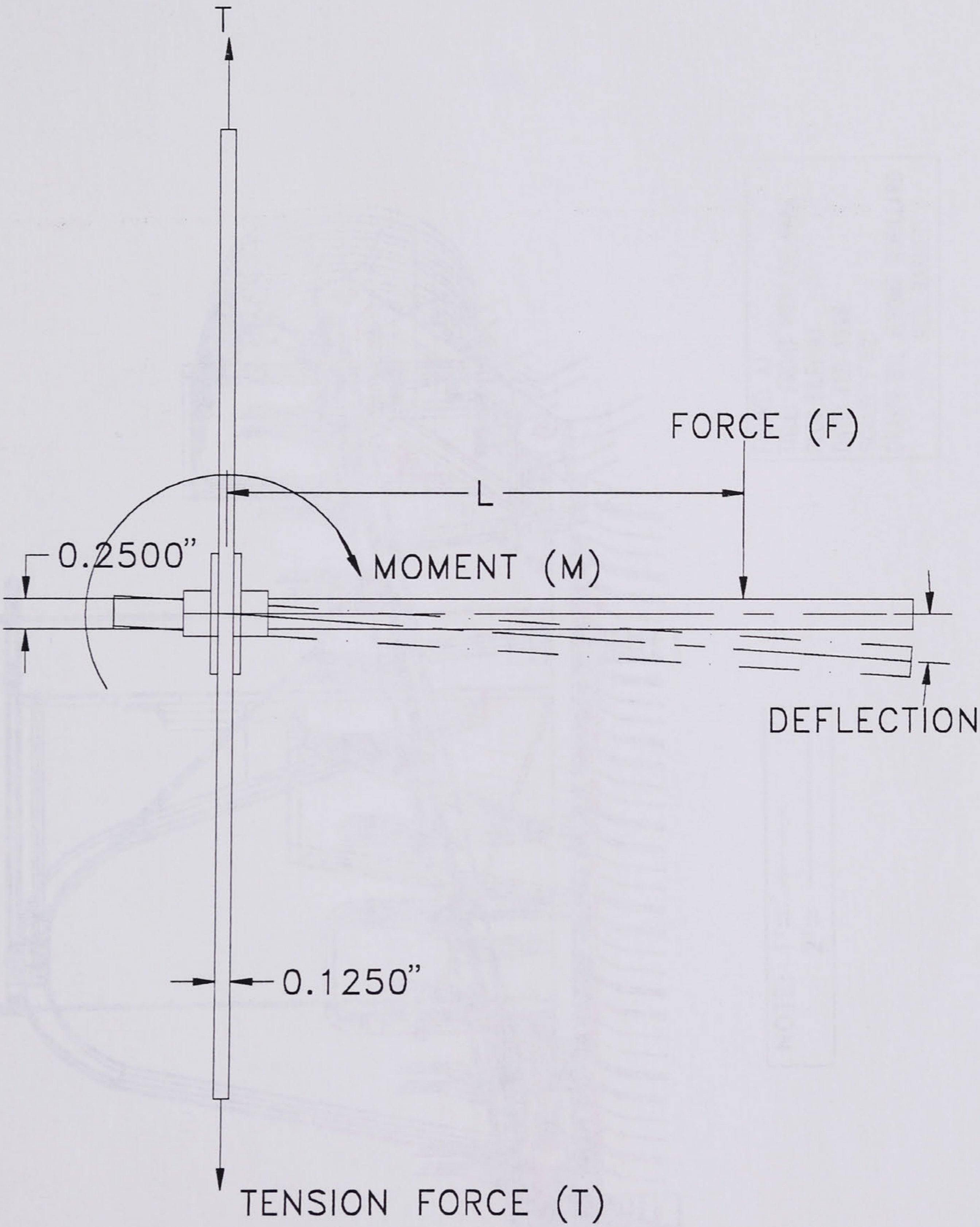
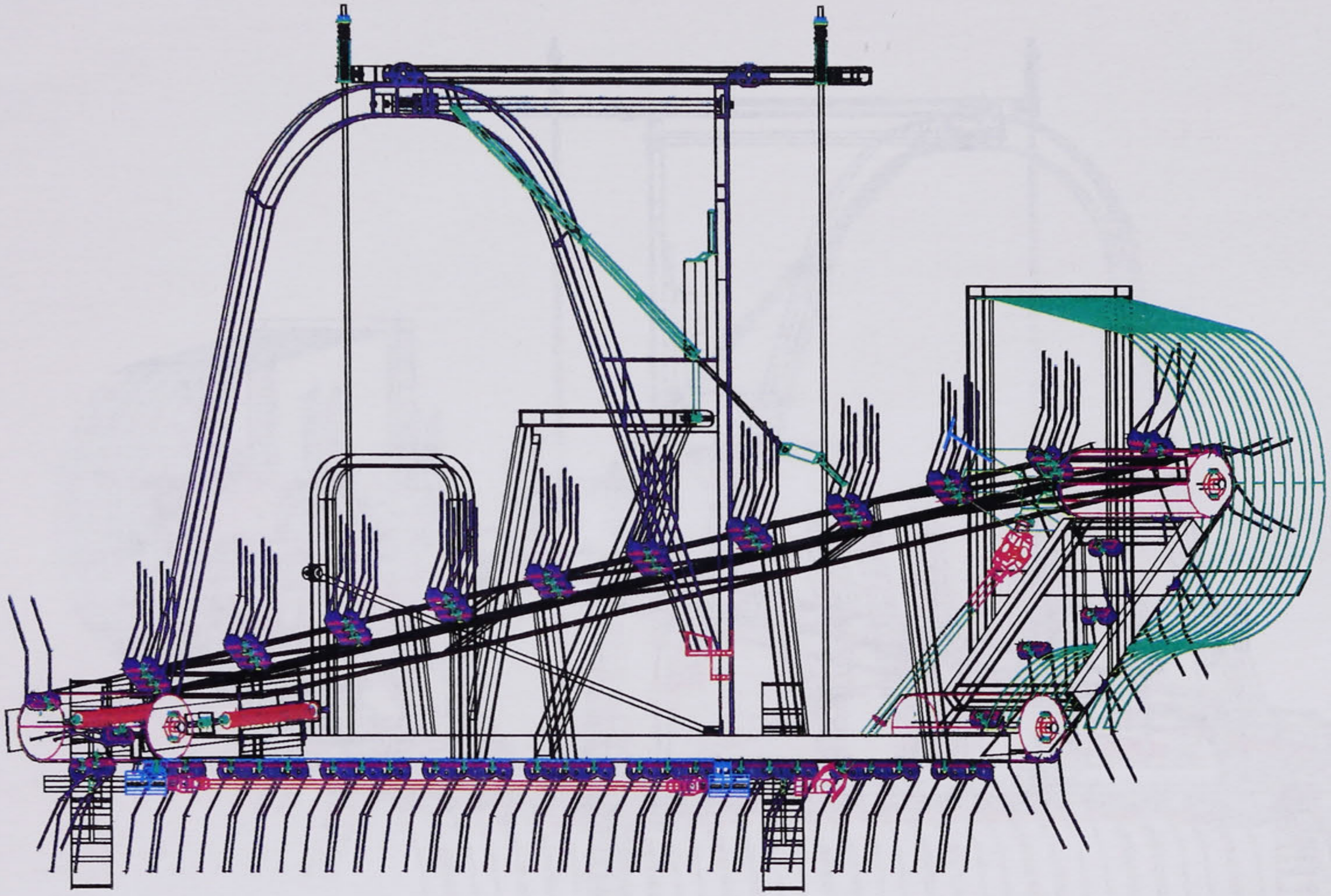
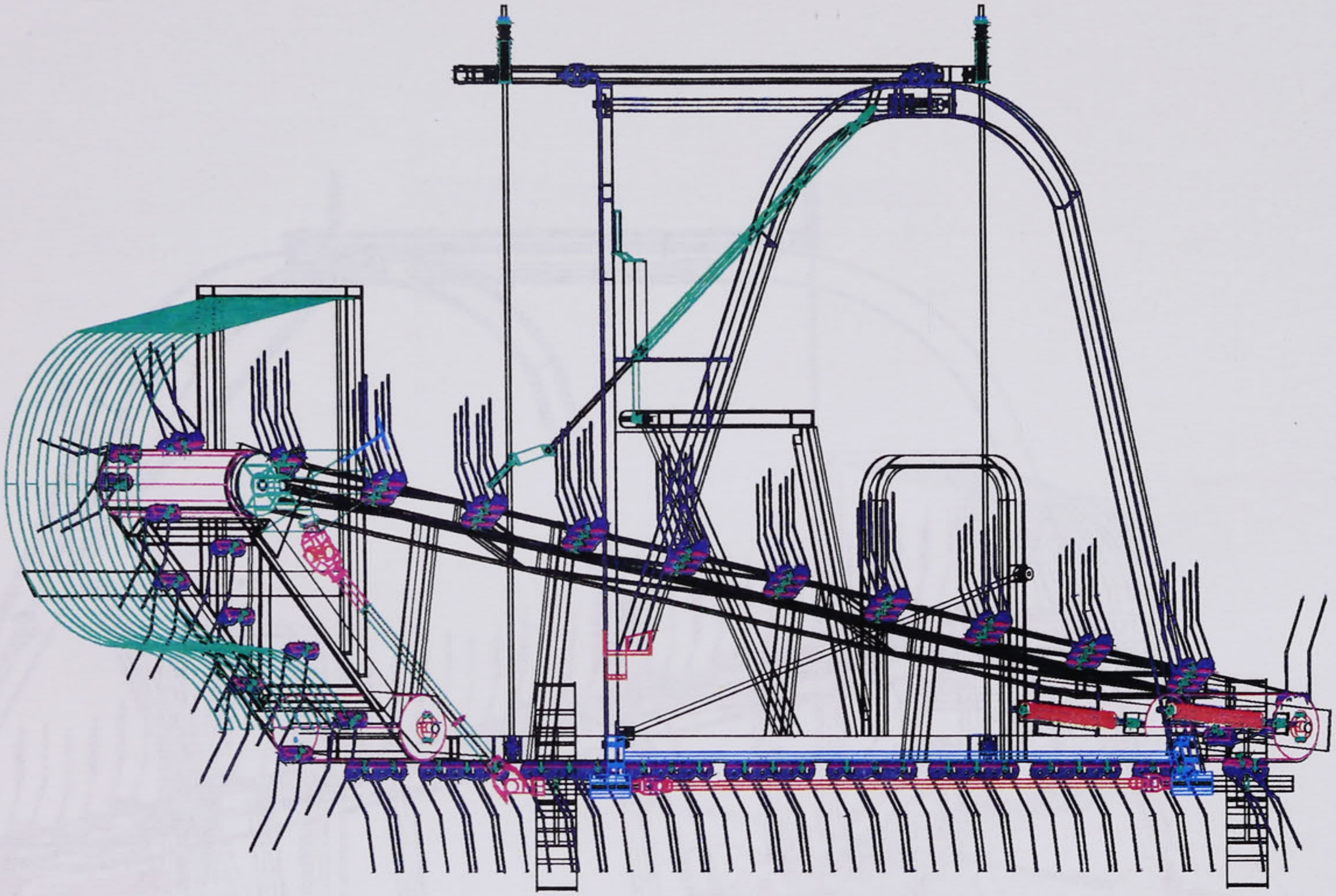


Figure 12 Schematic of Belt Test Setup



NOTE: 1'= ———
2'= —————

FIGURE: A1
TITLE: FRONT VIEW OF RAKE
PLOT FILE: A1
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ———
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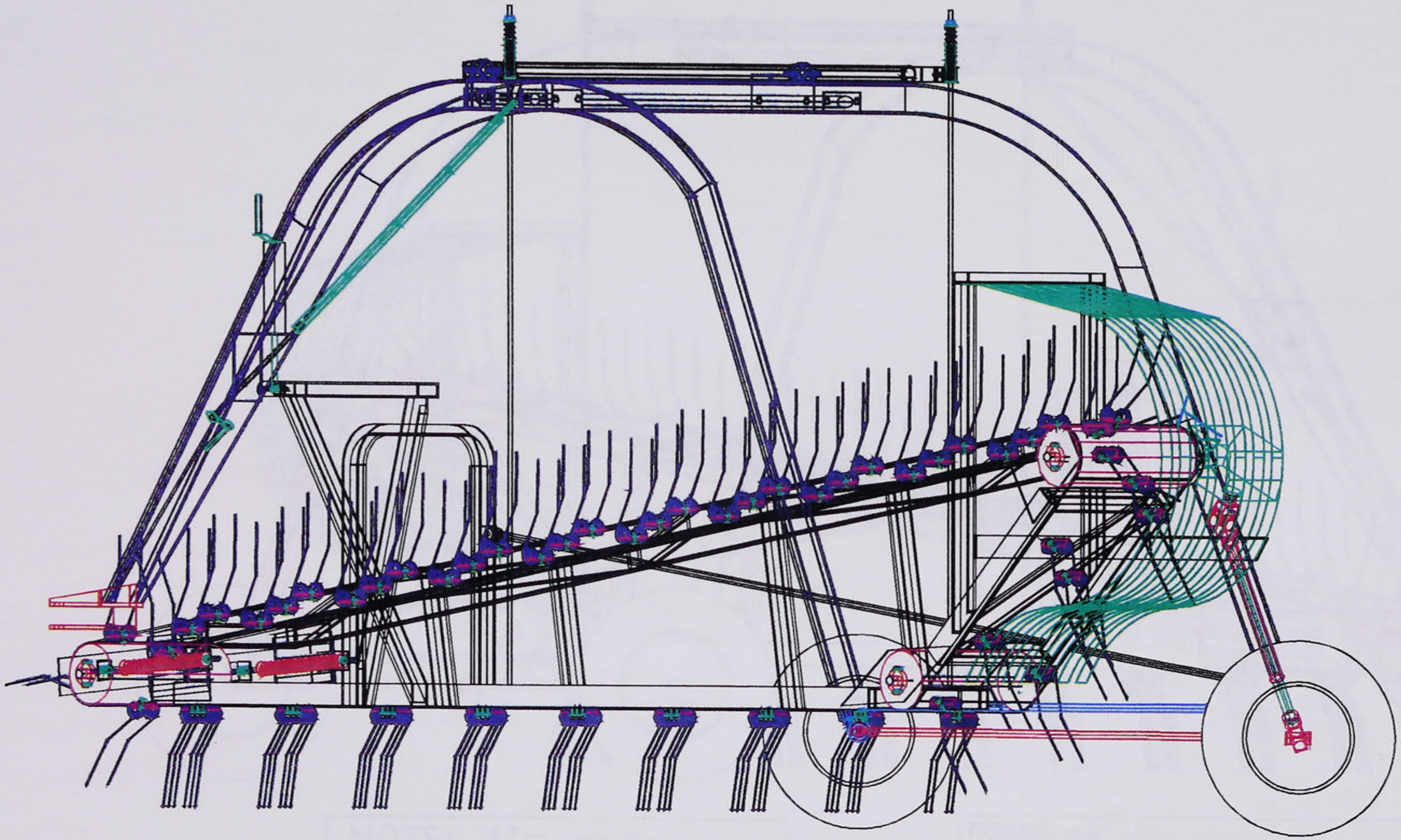
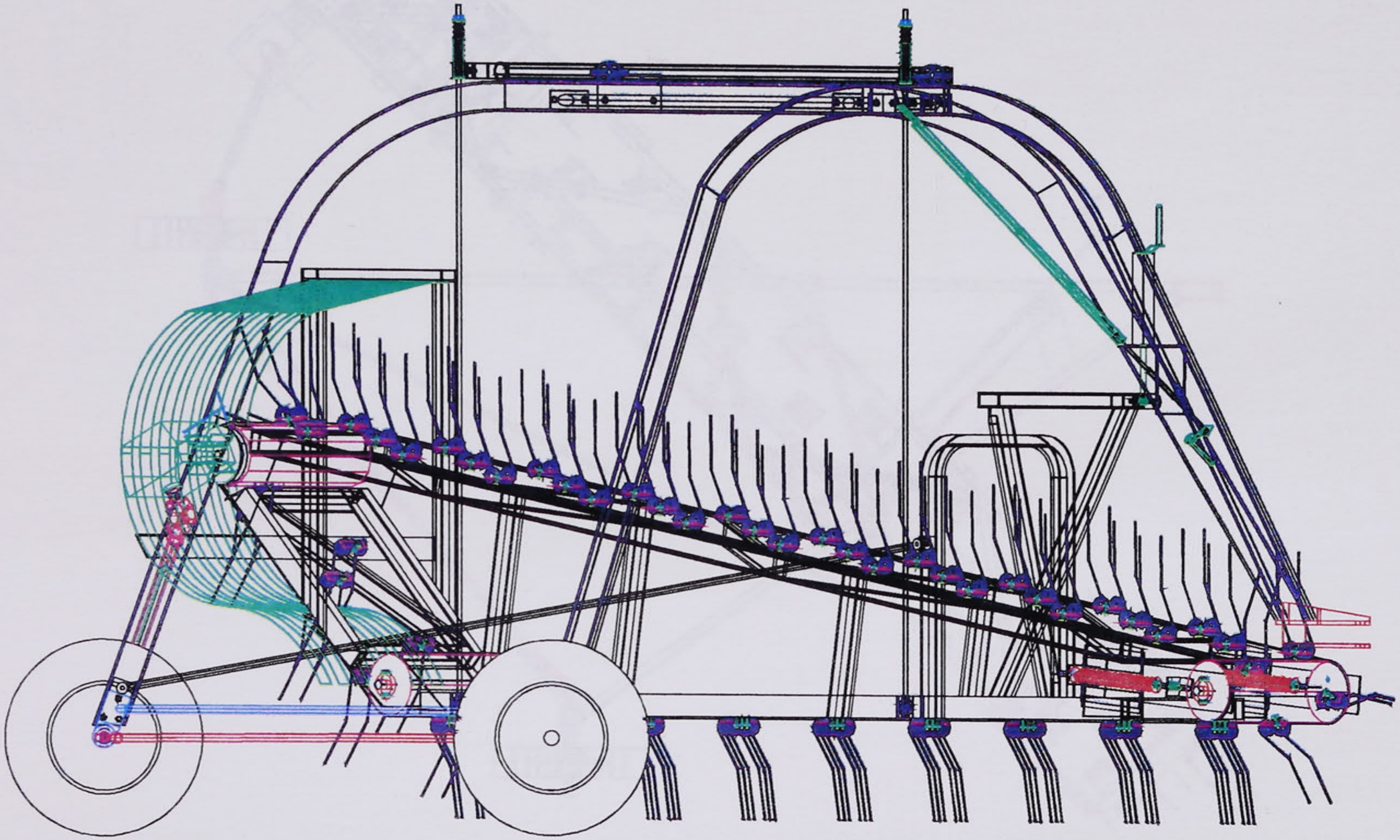
FIGURE: A2
TITLE: REAR VIEW OF RAKE
PLOT FILE: A2
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ———
2'= ——————

FIGURE: A3
TITLE: SIDE VIEW (DELIVERY) OF RAKE
PLOT FILE: A3
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ______
2'= __________

FIGURE: A4
TITLE: SIDE VIEW (PICKUP) OF RAKE
PLOT FILE: A4
DATE: FEB 1996
SCALE: 1"=2'
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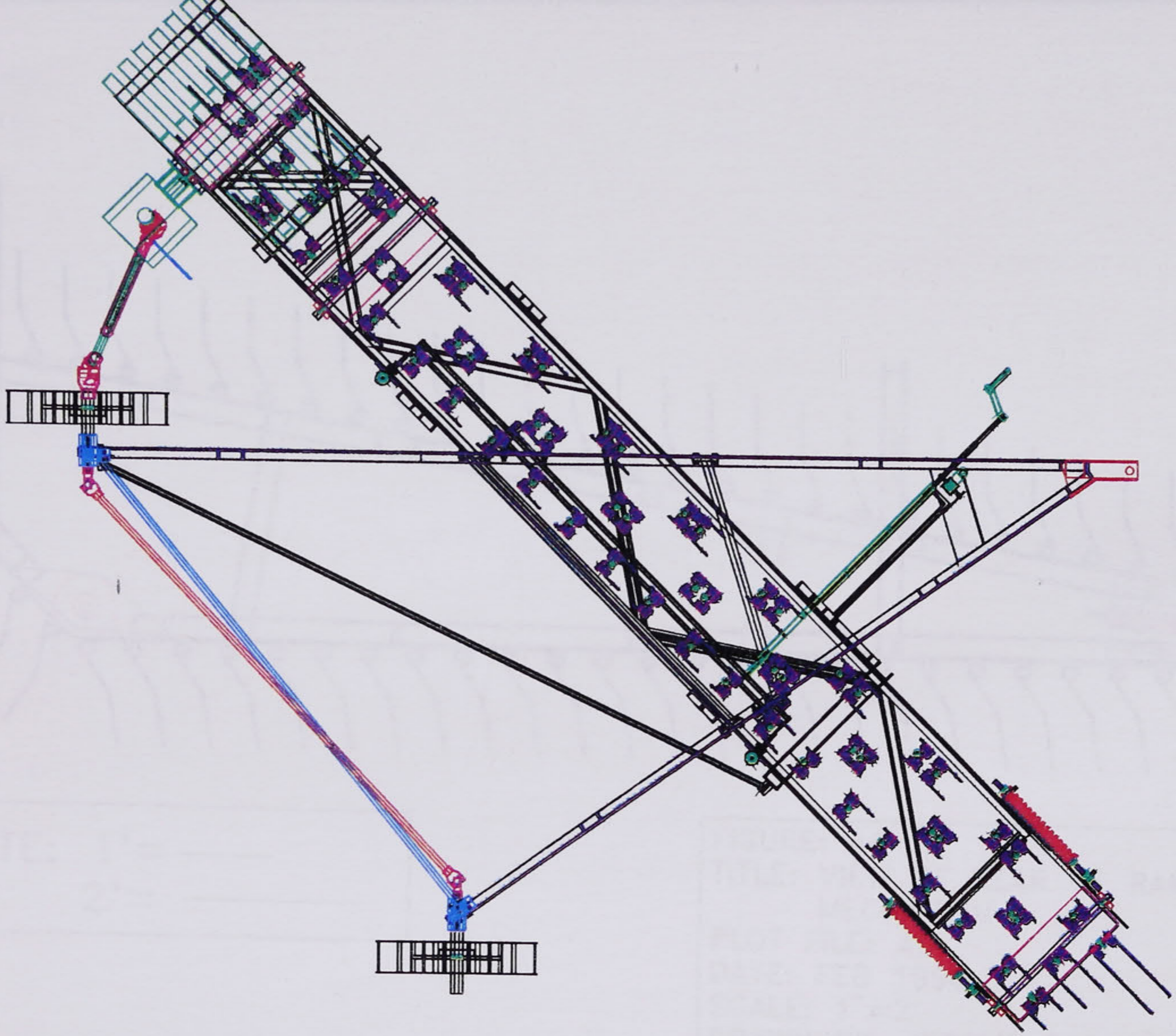
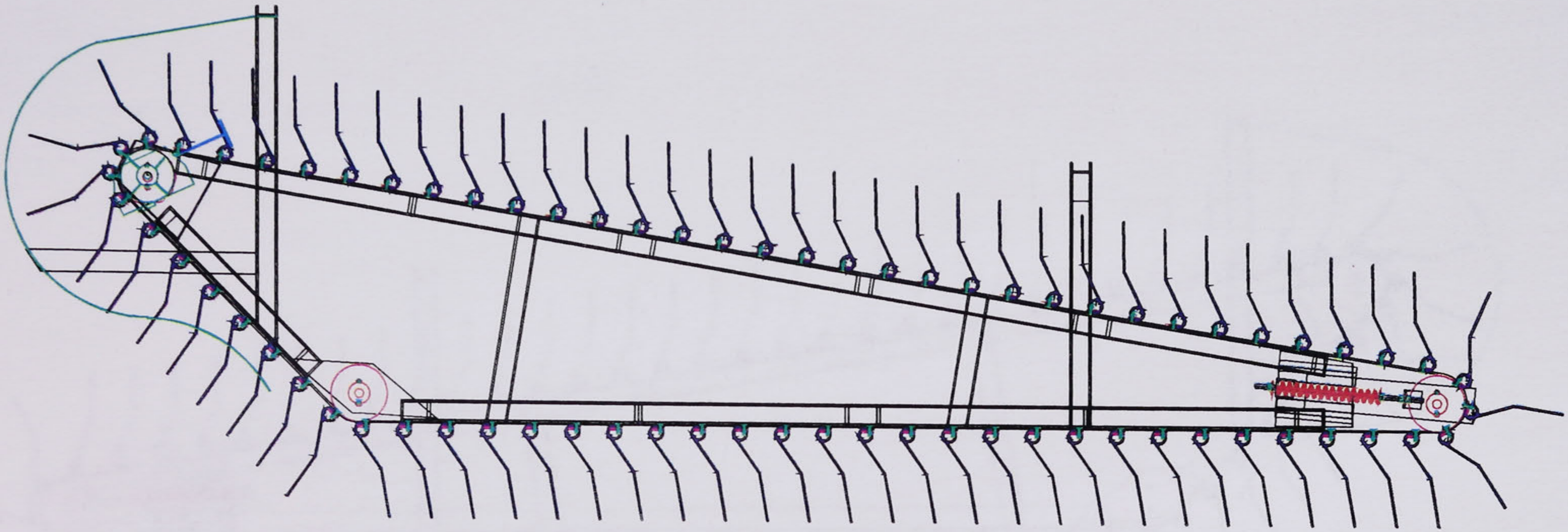


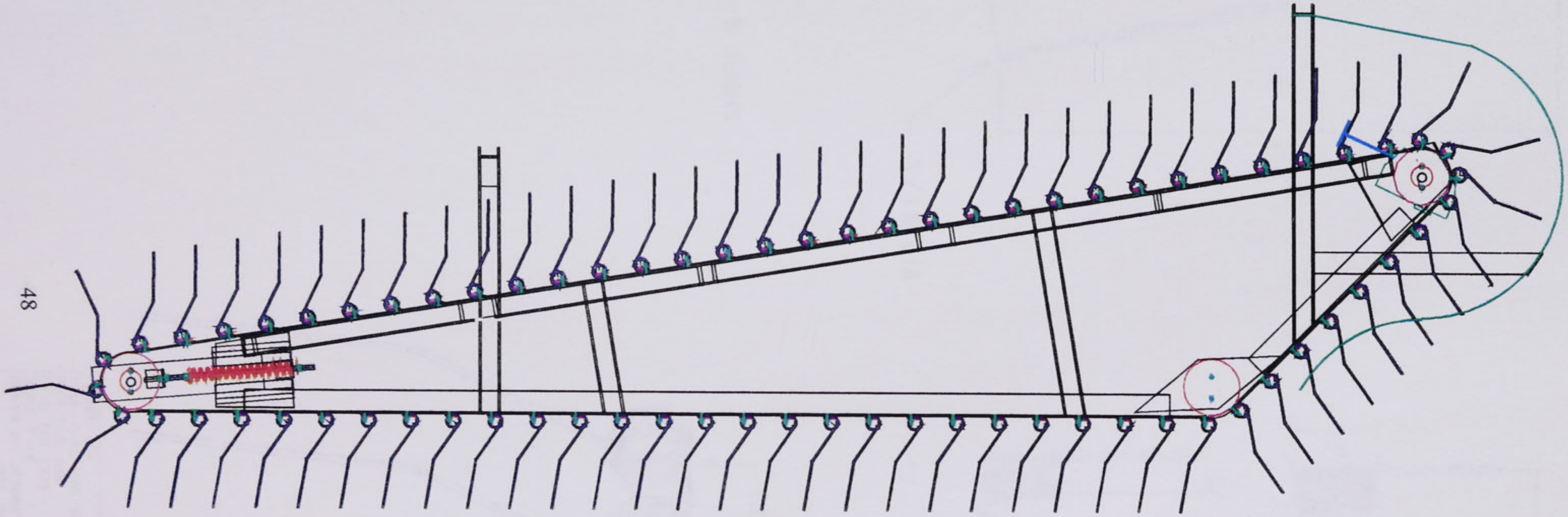
NOTE: 1'= ——
2'= ——

FIGURE: A5
TITLE: TOP VIEW OF RAKE
PLOT FILE: A5
DATE: FEB 1996
SCALE: 1"=2.5'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ——
2'= ————

FIGURE: A6
TITLE: VIEW OF REAR OF RAKING MECHANISM
PLOT FILE: A6
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ______
2'= __________

FIGURE: A7
TITLE: VIEW OF FRONT RAKING MECHANISM
PLOT FILE: A7
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

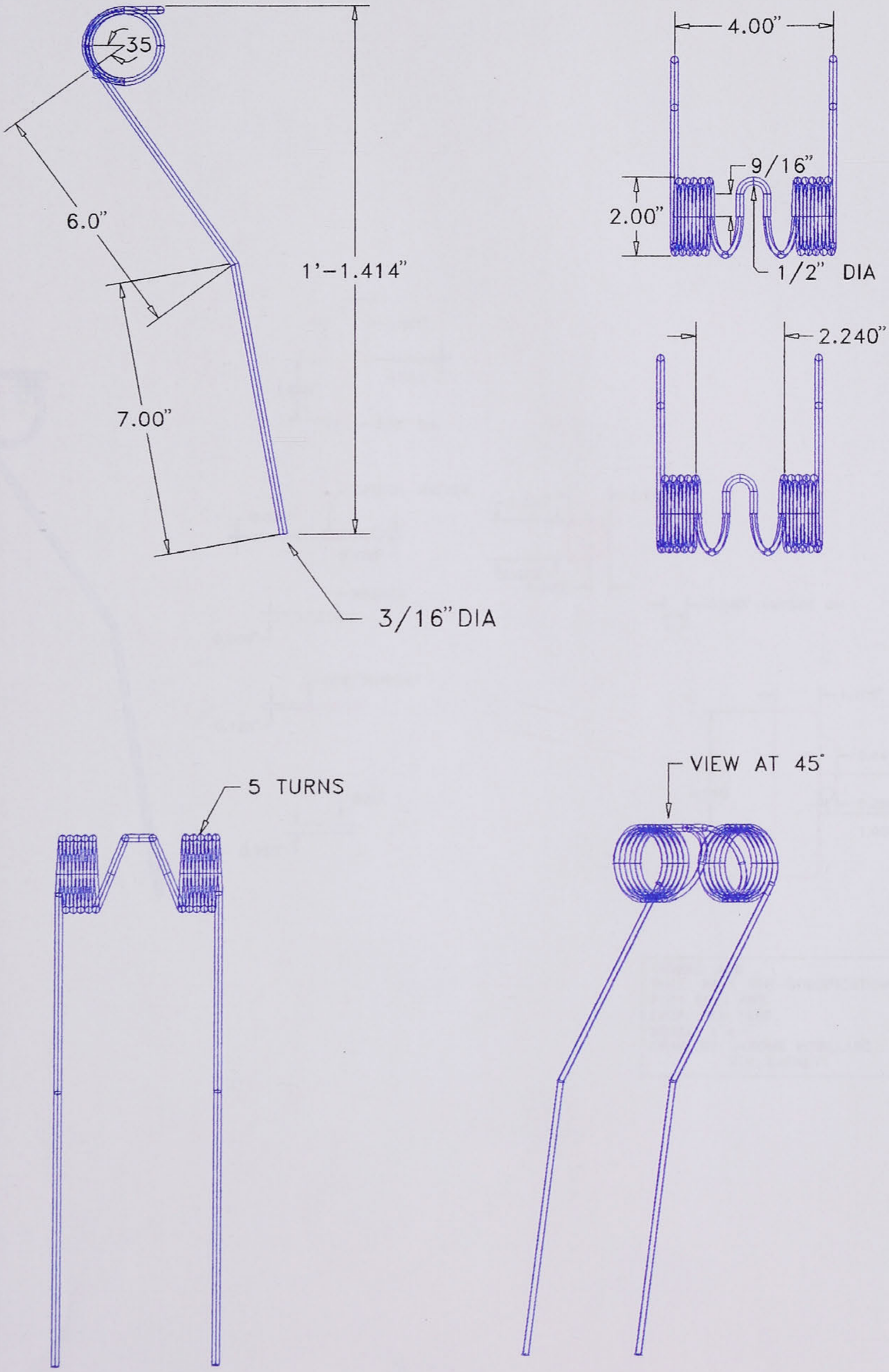


FIGURE: A8A
TITLE: RAKE TINE SPECIFICATIONS (VIEW 1)
PLOT FILE: AA8
DATE: FEB 1996
SCALE 1"=4"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

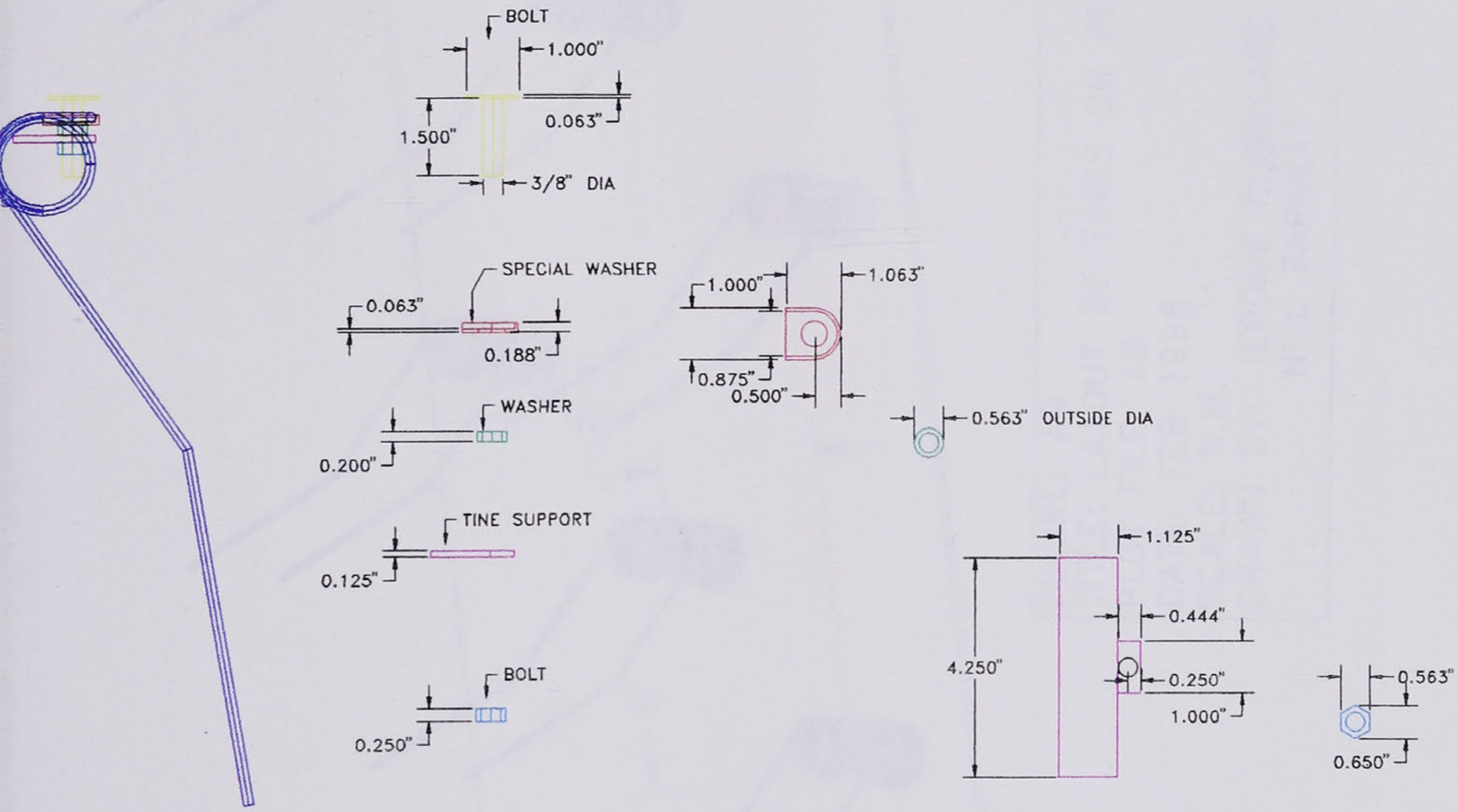


FIGURE: A8B
TITLE: RAKE TINE SPECIFICATIONS (VIEW 2)
PLOT FILE: AB8
DATE: FEB 1996
SCALE: 1" = 4"
DRAW BY: JEROME ROBILLARD
NEIL BARNETT

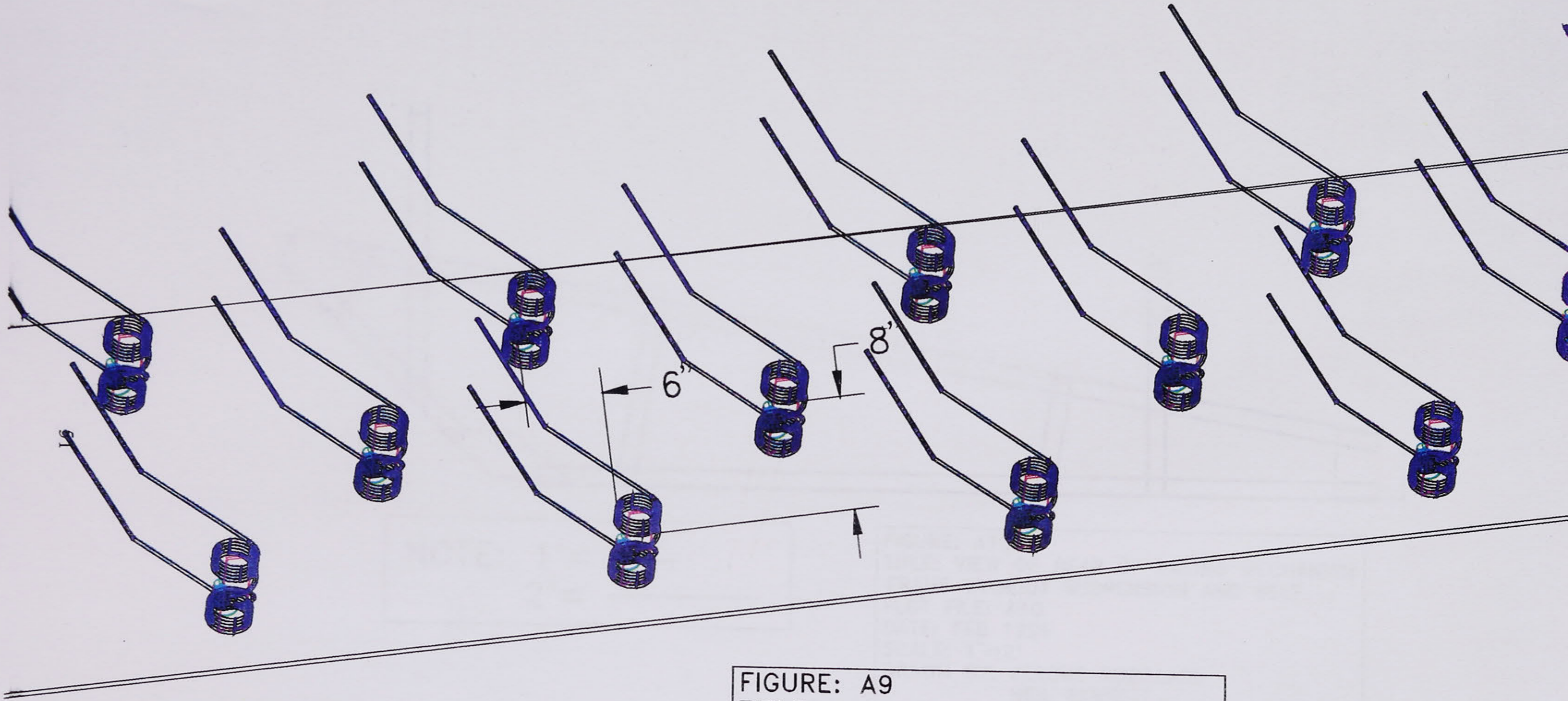
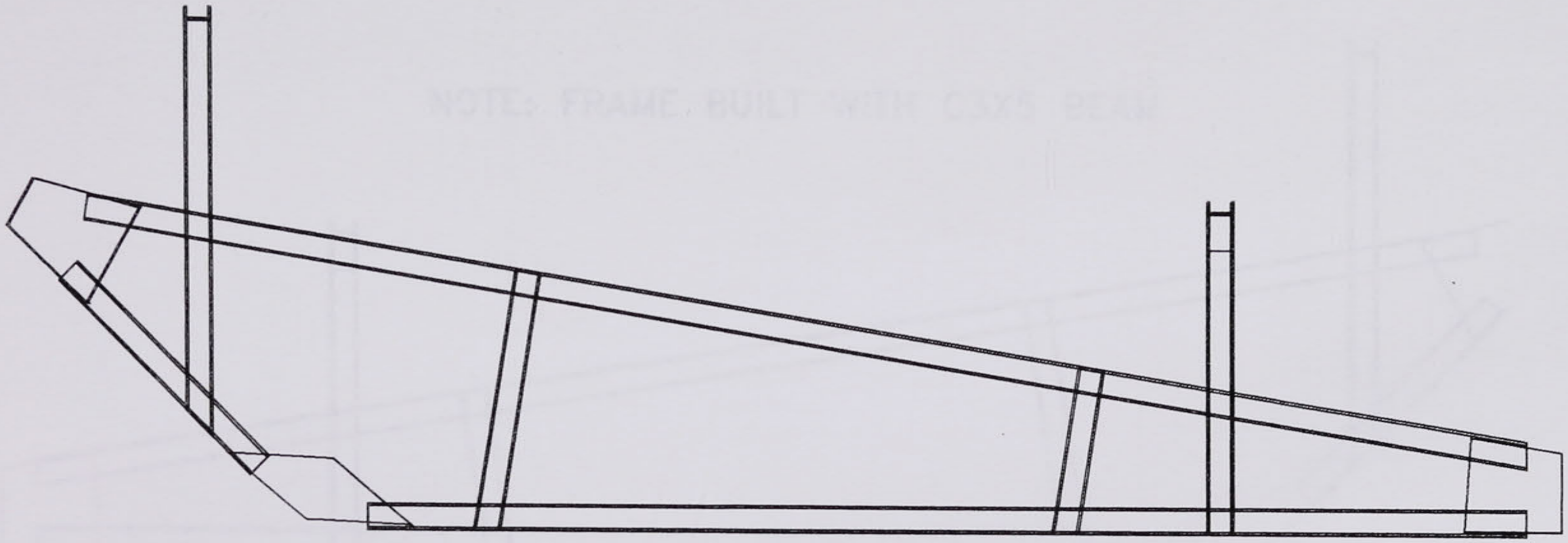


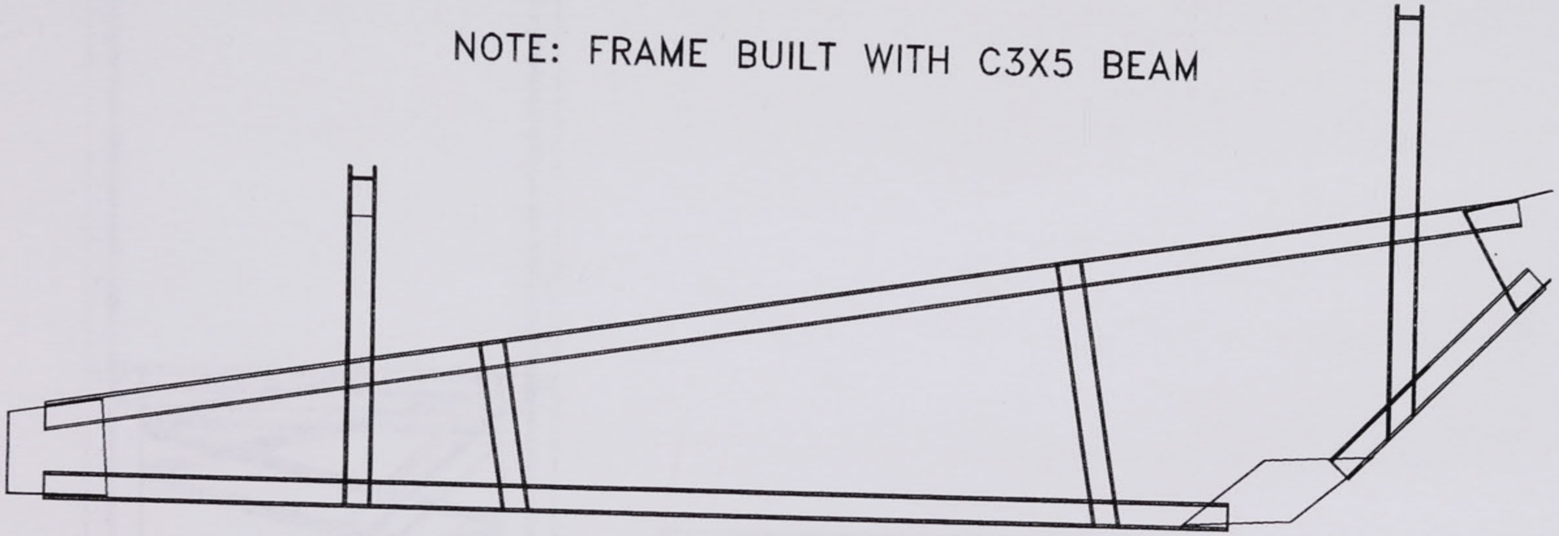
FIGURE: A9
TITLE: LAYOUT OF TINES ON BELT
PLOT FILE: A9
DATE: FEB 1996
SCALE: N.M.
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ——
2'= ————

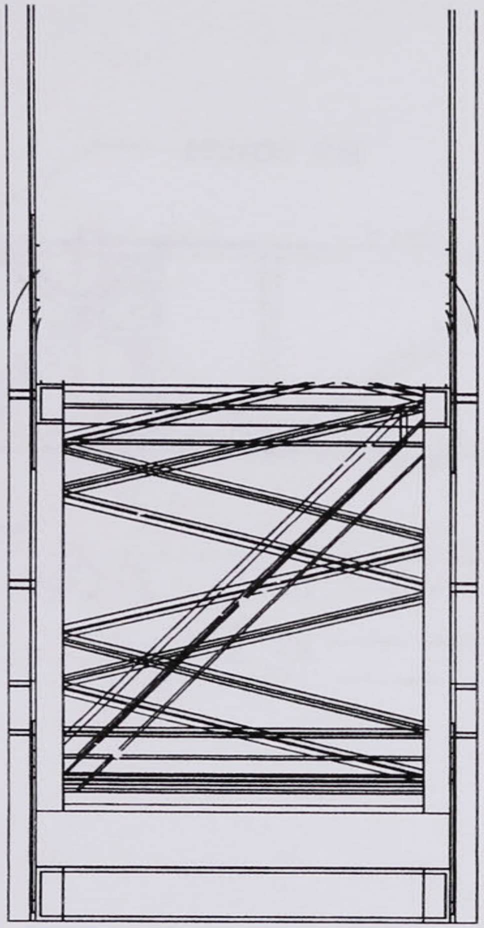
FIGURE: A10
TITLE: VIEW OF REAR OF RAKING MECHANISM
FRAME WITHOUT SUSPENSION AND BELT
PLOT FILE: A10
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

NOTE: FRAME BUILT WITH C3X5 BEAM



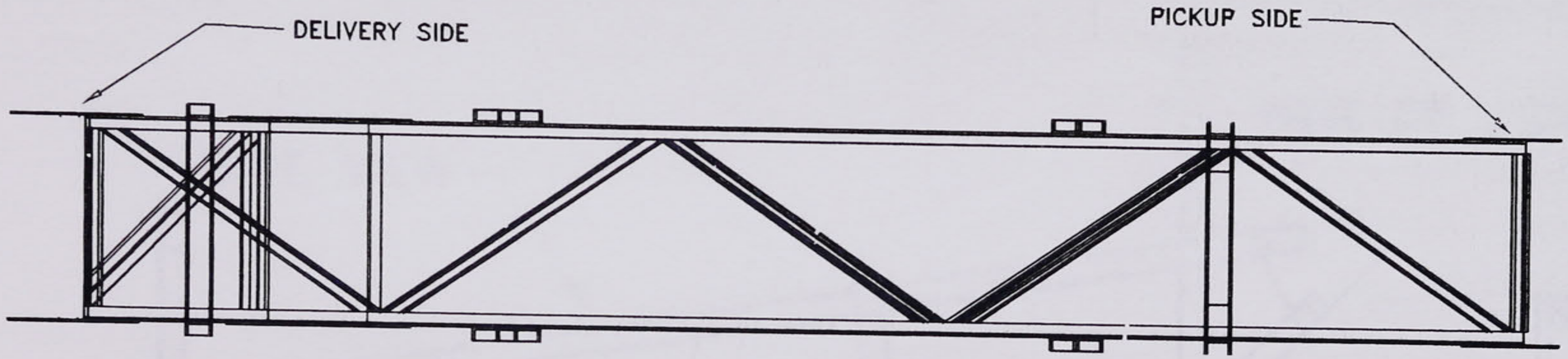
NOTE: 1'= ——
2'= ————

FIGURE: A11
TITLE: VIEW OF FRONT OF RAKING MECHANISM
FRAME WITHOUT SUSPENSION AND BELT
PLOT FILE: A11
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'=
2'=

FIGURE: A12
TITLE:DELIVERY SIDE VIEW OF RAKING MECHANISM FRAME WITHOUT SUSPENSION AND BELT
PLOT FILE: A12
DATE: FEB 1996
SCALE: 1"=1'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ——

2'= ————

FIGURE: A13
TITLE: TOP VIEW OF RAKING MECHANISM WITHOUT SUSPENSION AND BELT
PLOT FILE: A13
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

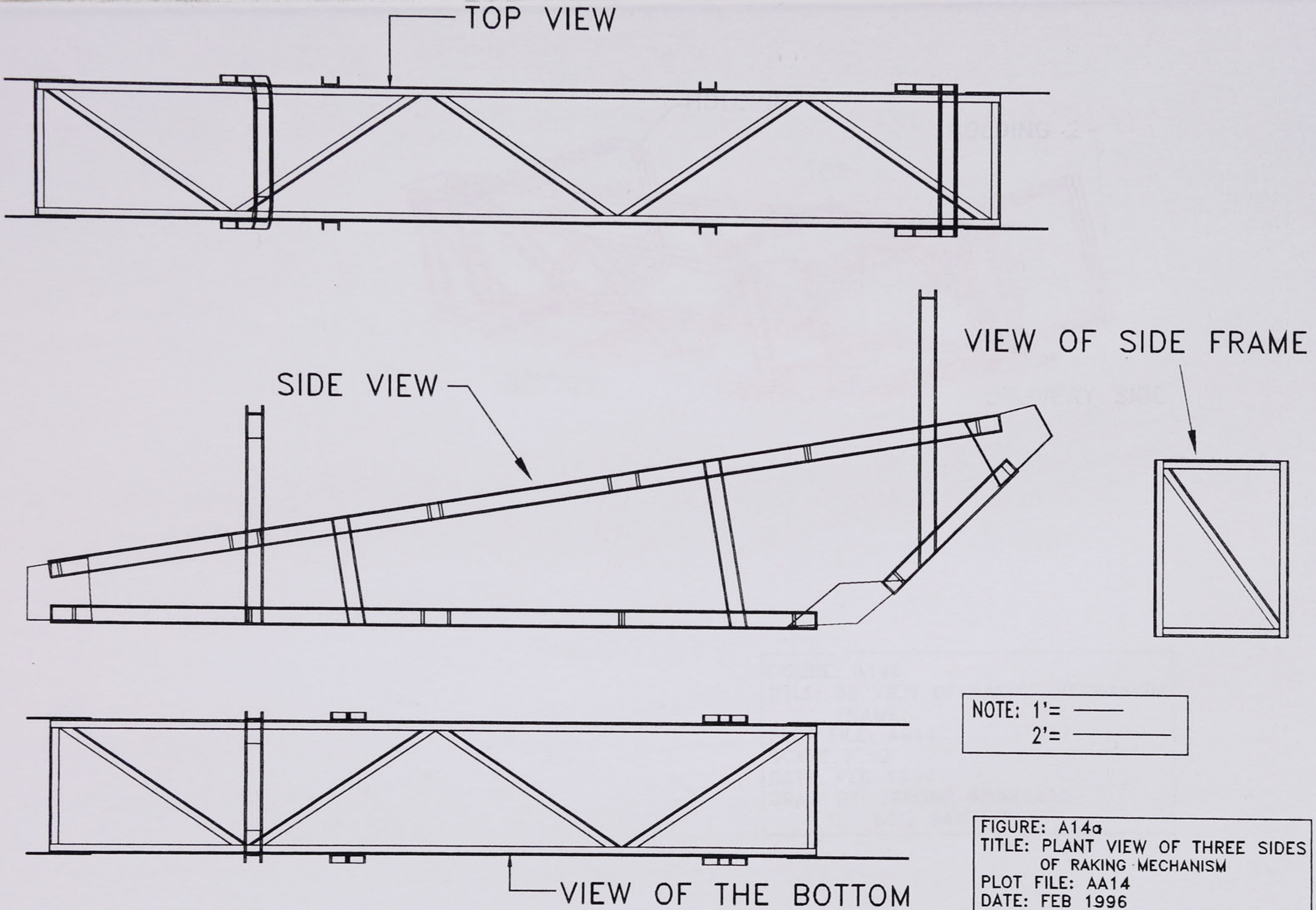


FIGURE: A14a
TITLE: PLANT VIEW OF THREE SIDES OF RAKING MECHANISM
PLOT FILE: AA14
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

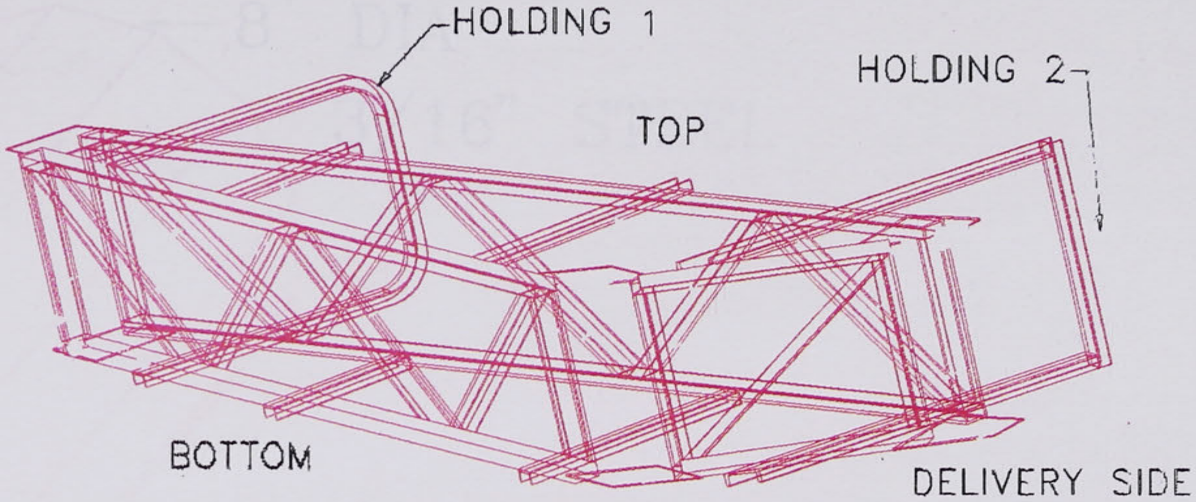


FIGURE: A14B
TITLE: 3D VIEW OF RAKING MECHANISM FRAME
PLOT FILE: AB14
SCALE 1"=2'
DATE: FEB 1996
DRAW BY: JEROME ROBILLARD
NEIL BARNETT

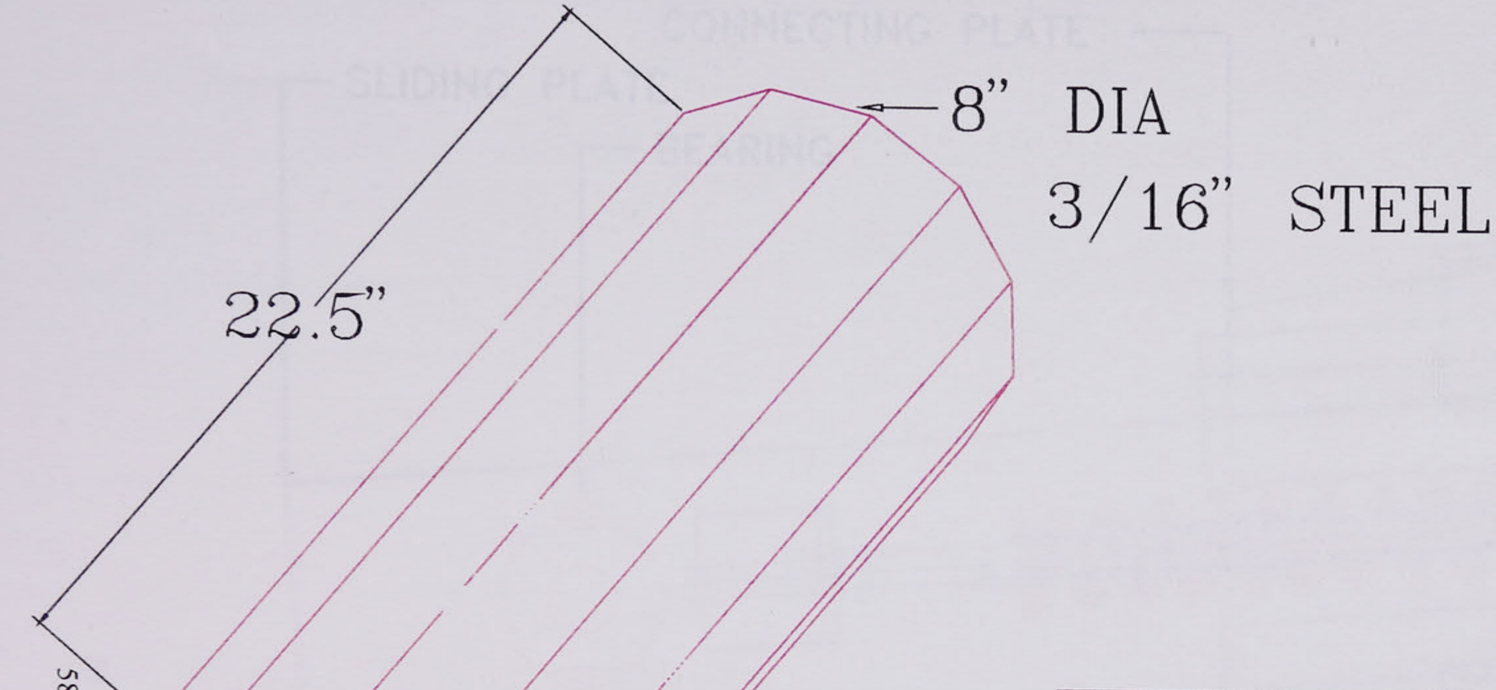
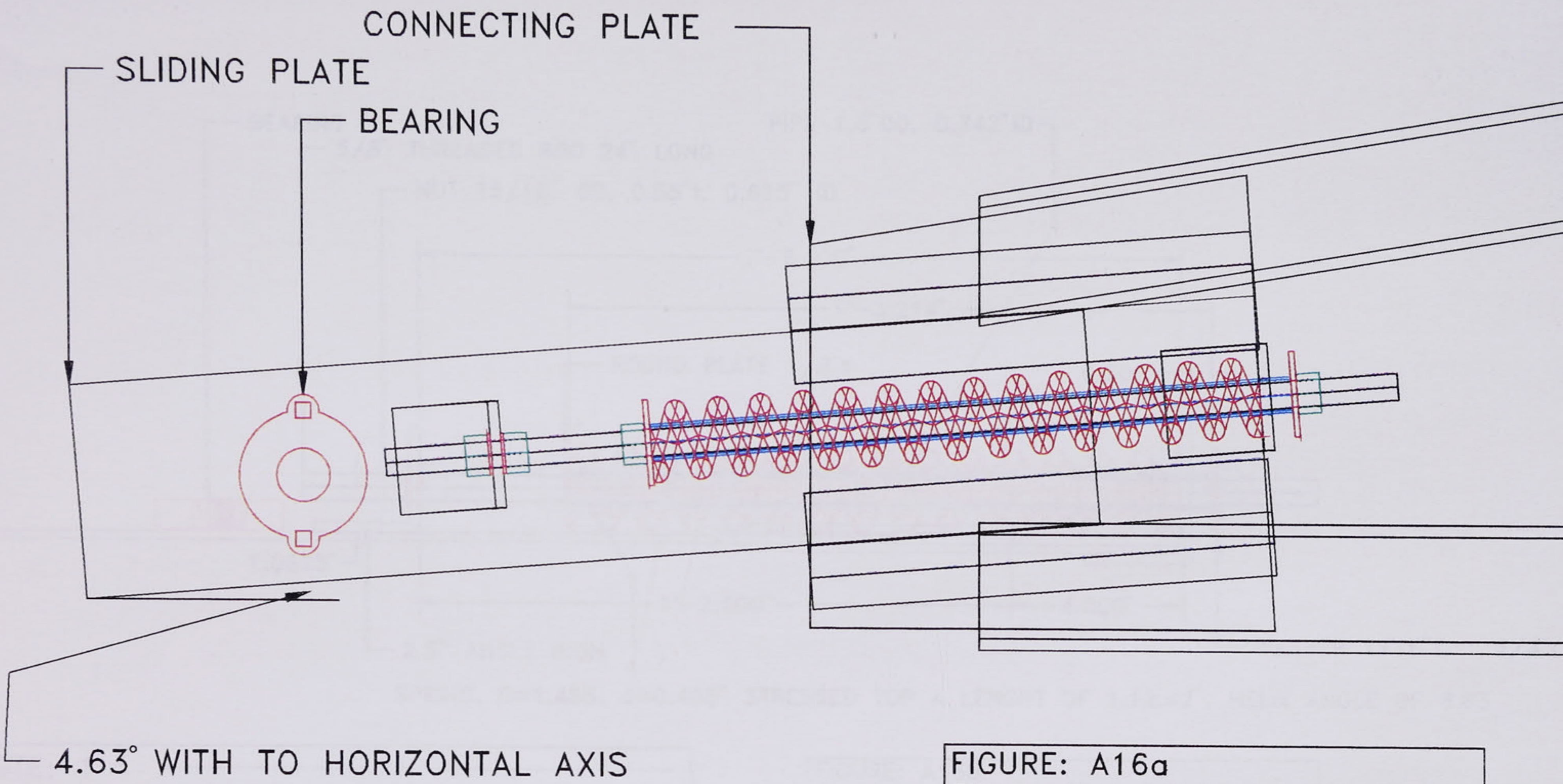
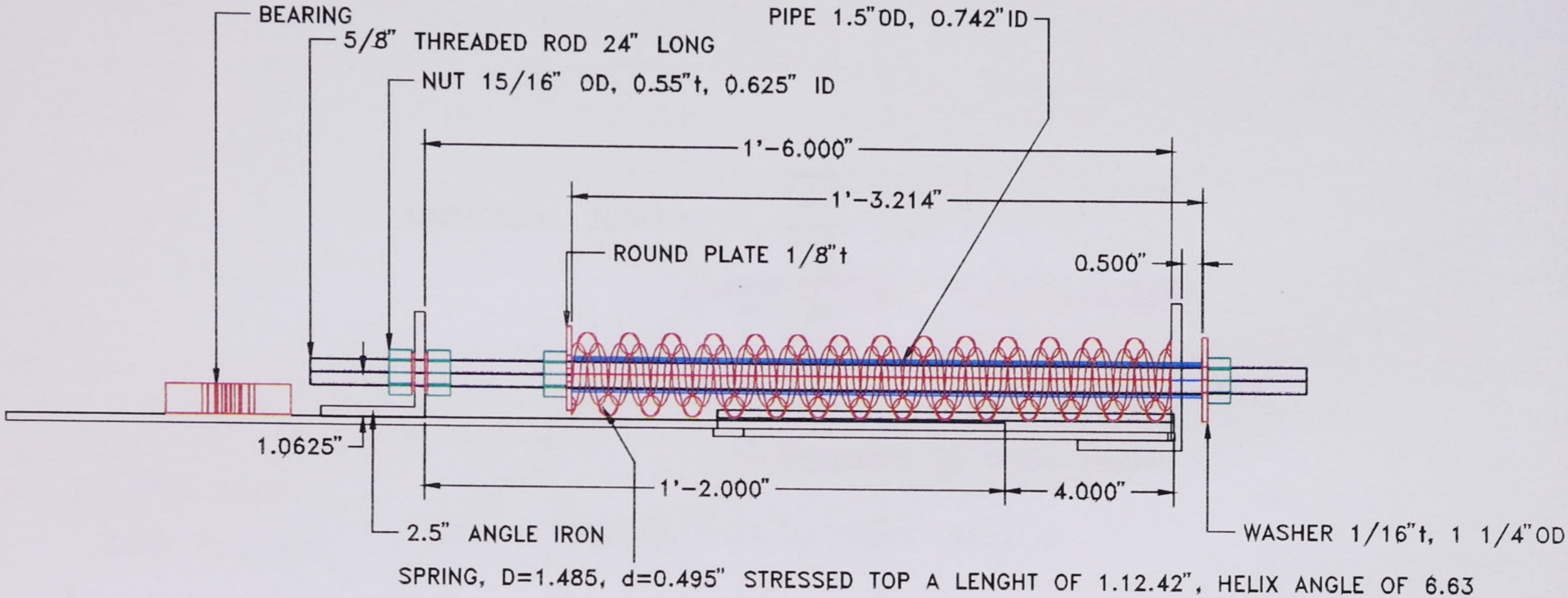


FIGURE: A15
TITLE: ROLLER FOR BELT
PLOT FILE: A15
DATE: FEB 95
SCALE: 1"=4"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ____________
4"= ______

FIGURE: A16a
TITLE: SIDE VIEW OF TIGHTENER
PLOT FILE: AA16
DATE: FEB 1996
SCALE: 1"=4"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ____________
4"= ______

FIGURE: A16b
TITLE: TOP VIEW OF TIGHTENER
PLOT FILE: AB16
DATE: FEB 1996
SCALE: 1"=4"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

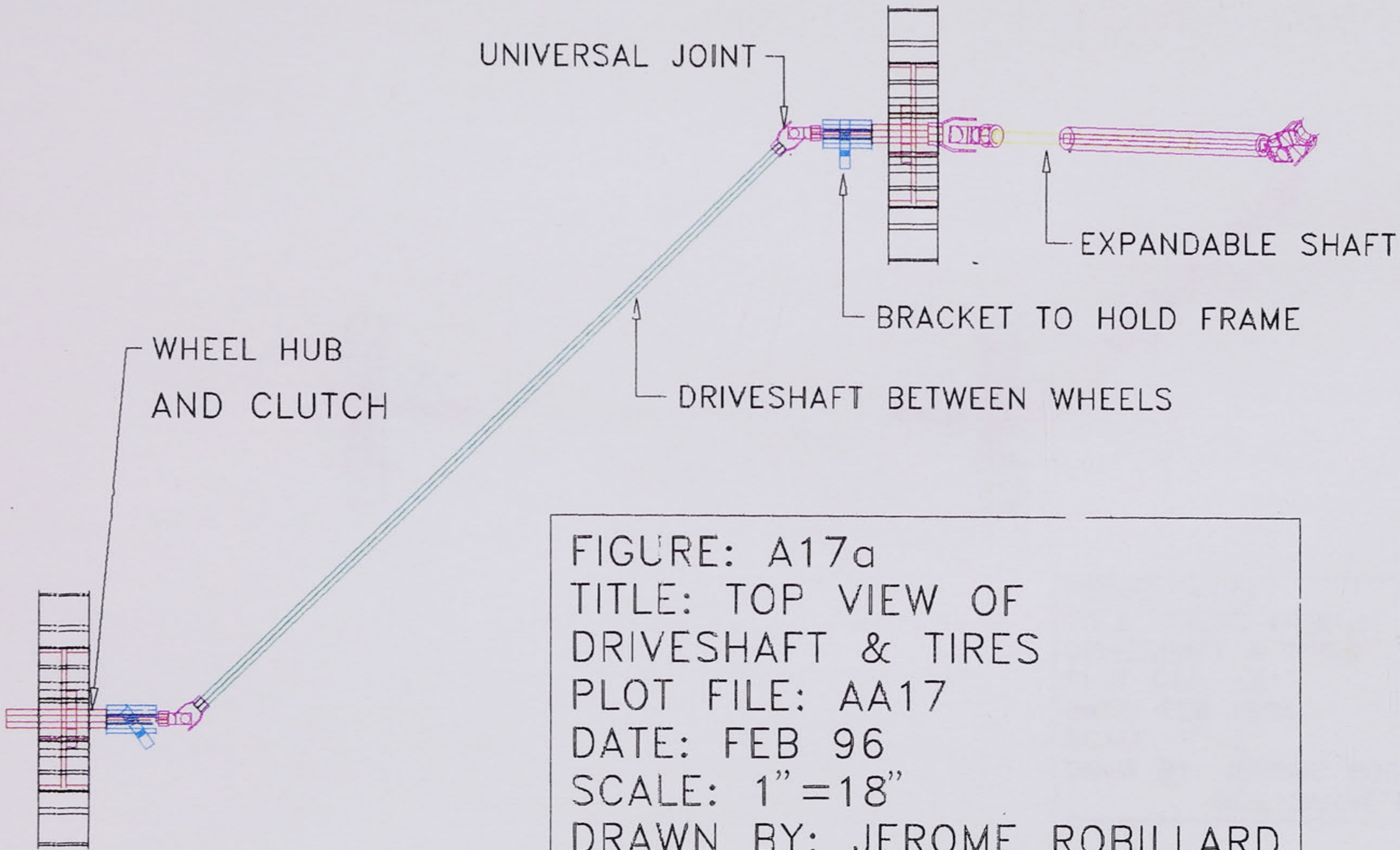


FIGURE: A17a
TITLE: TOP VIEW OF
DRIVESHAFT & TIRES
PLOT FILE: AA17
DATE: FEB 96
SCALE: 1"=18"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

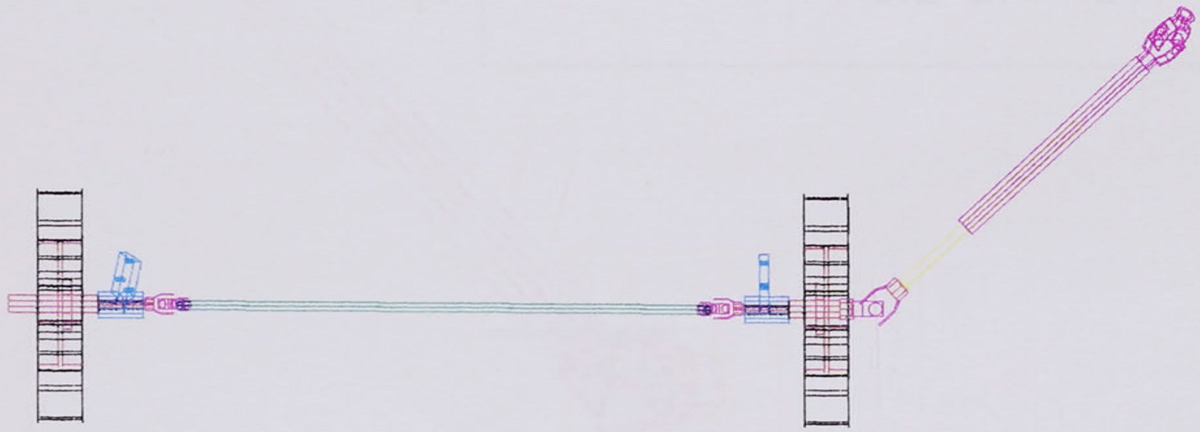


FIGURE: A17B
TITLE: FRONT VIEW OF
DRIVESHAFT & TIRES
PLOT FILE: AB17
DATE: FEB 1996
SCALE: 1"=2'
DRAW BY: JEROME ROBILLARD
NEIL BARNETT

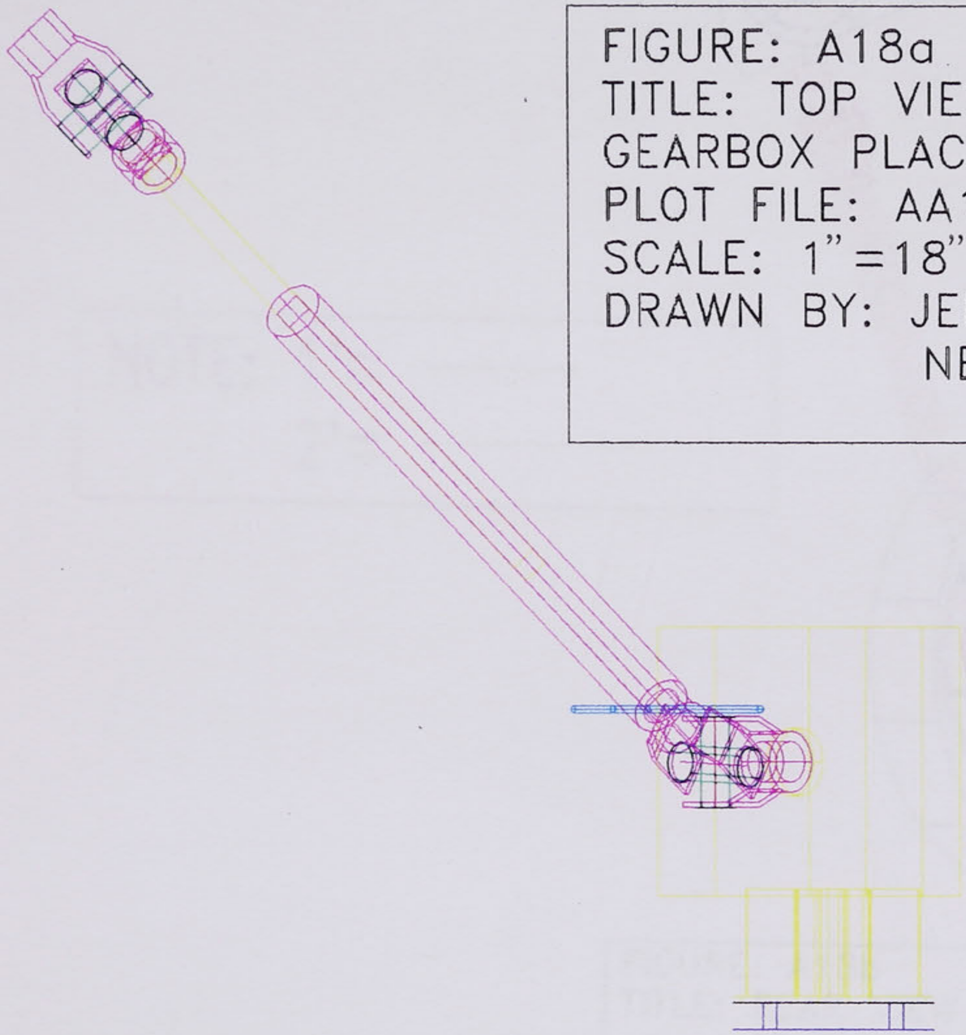
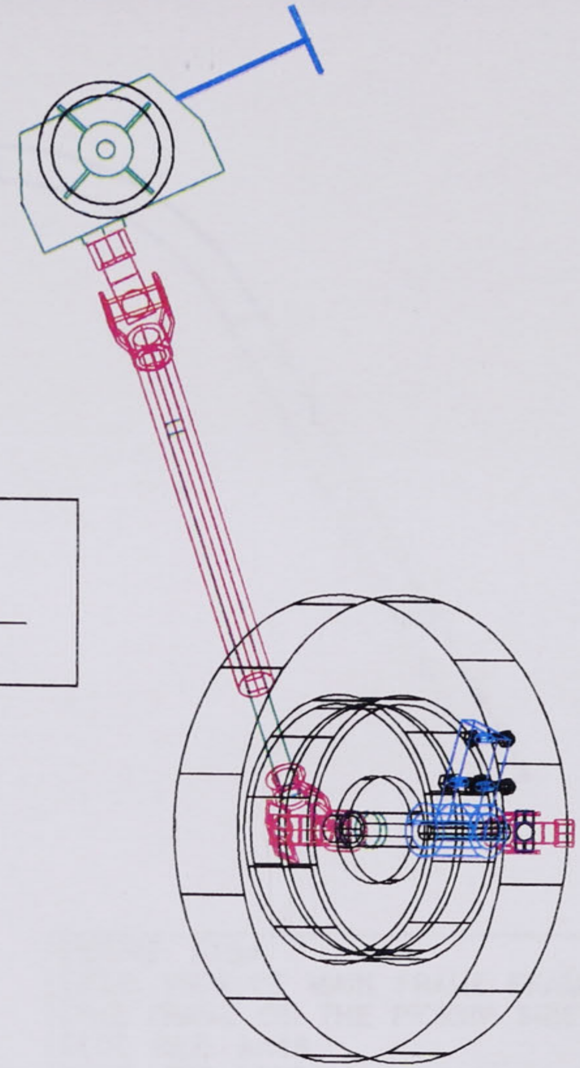


FIGURE: A18a
TITLE: TOP VIEW OF
GEARBOX PLACEMENT
PLOT FILE: AA18
SCALE: 1"=18"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ————

2'= ————————

FIGURE: A18b
TITLE: REAR VIEW OF
GEARBOX PLACEMENT
PLOT FILE: AB18
DATE: FEB 1996
SCALE: 1"=1'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

FIGURE: A19A
TITLE: VIEW OF MAIN FRAME FRAME
(THE FRAME ON THE PICKUP SIDE)
PLOT FILE: AA19
DATE: FEB 1996
SCALE 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

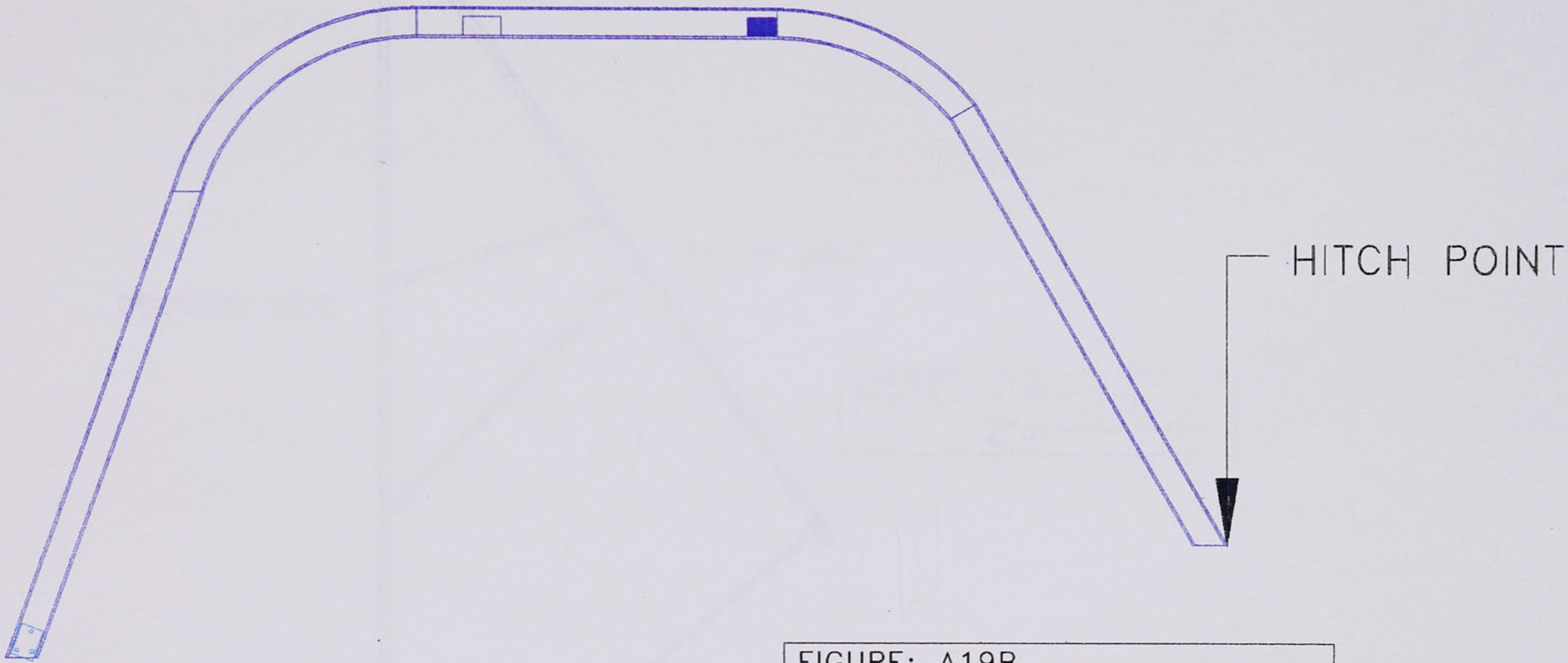


FIGURE: A19B
TITLE: VIEW OF MAIN FRAME
(FRAME ON THE DELIVERY SIDE)
PLOT FILE: AB19
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

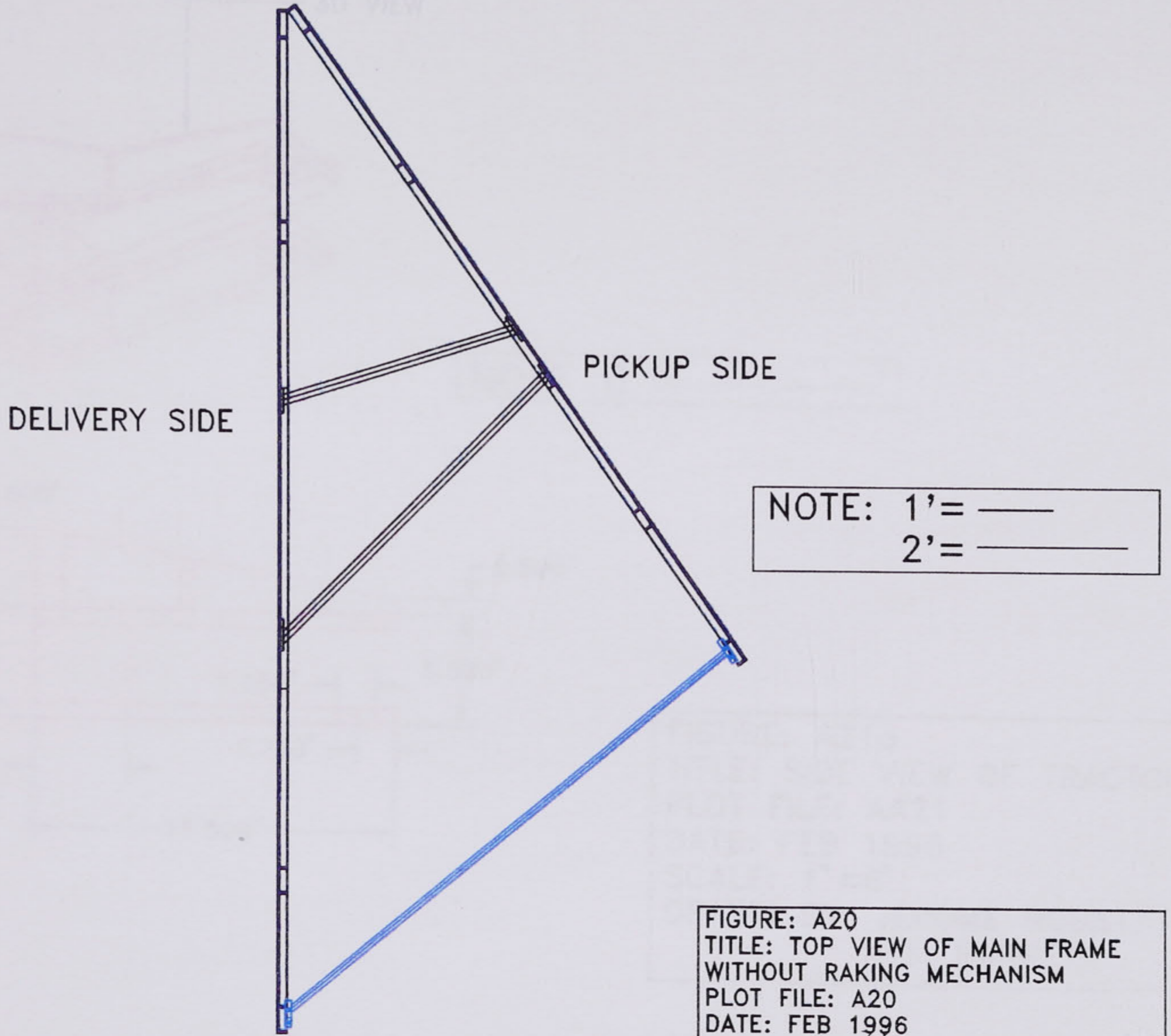
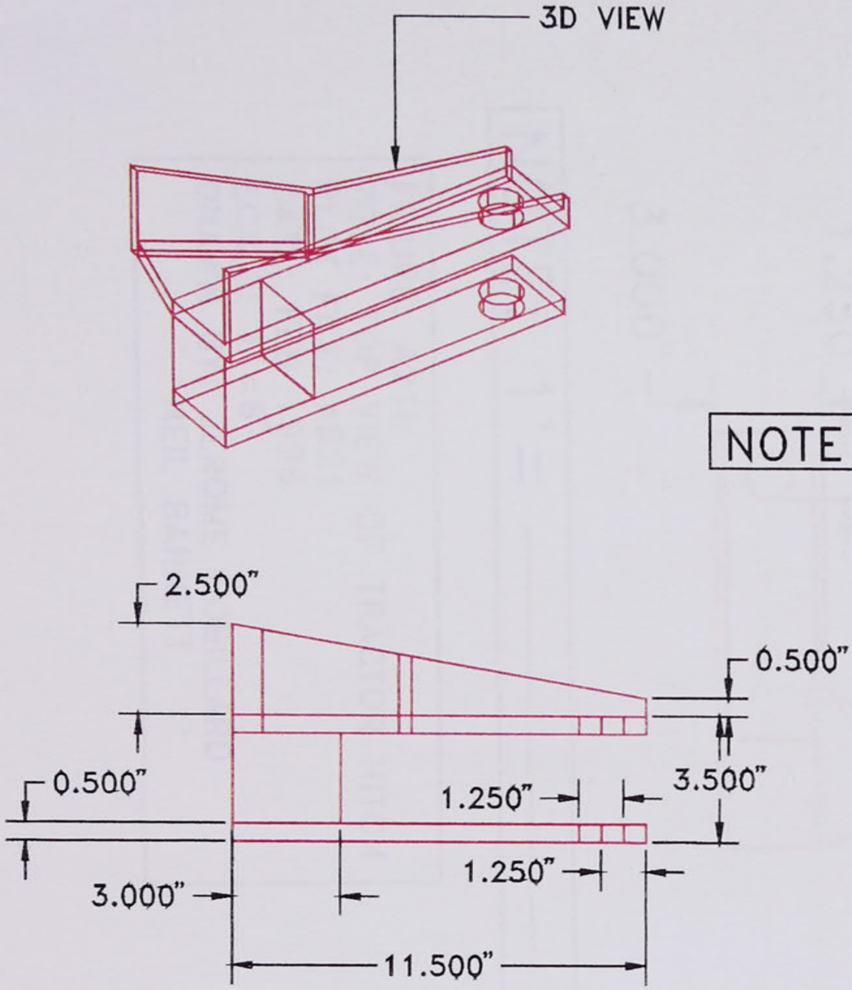
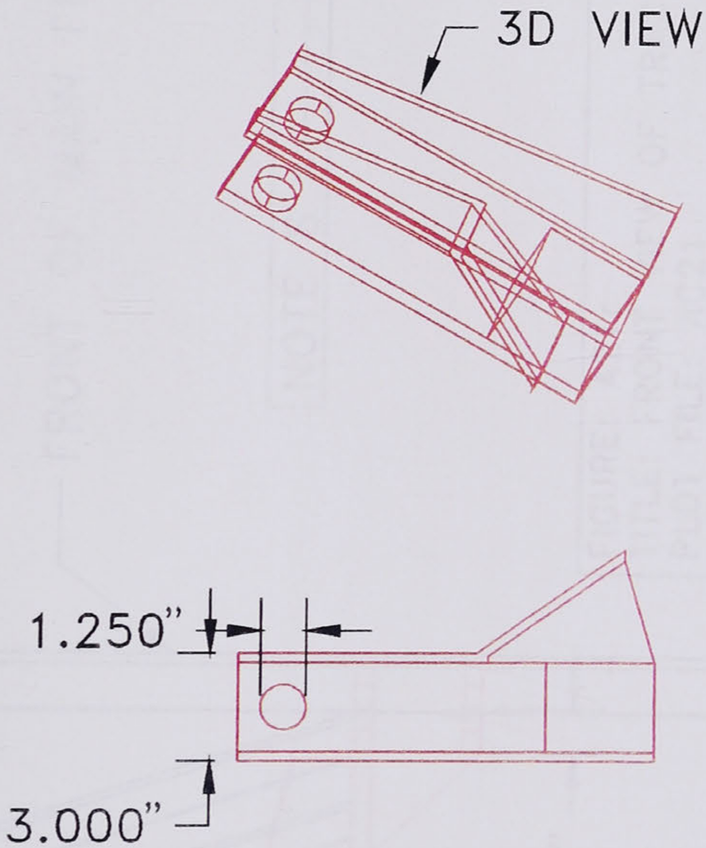


FIGURE: A20
TITLE: TOP VIEW OF MAIN FRAME
WITHOUT RAKING MECHANISM
PLOT FILE: A20
DATE: FEB 1996
SCALE: 1"=2.5'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE 6" = ________

FIGURE: A21a
TITLE: SIDE VIEW OF TRACTOR HITCH
PLOT FILE: AA21
DATE: FEB 1996
SCALE: 1"=6"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ______________

FIGURE: A21b
TITLE: TOP VIEW OF TRACTOR HITCH
PLOT FILE: AB21
DATE: FEB 1996
SCALE: 1"=6"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

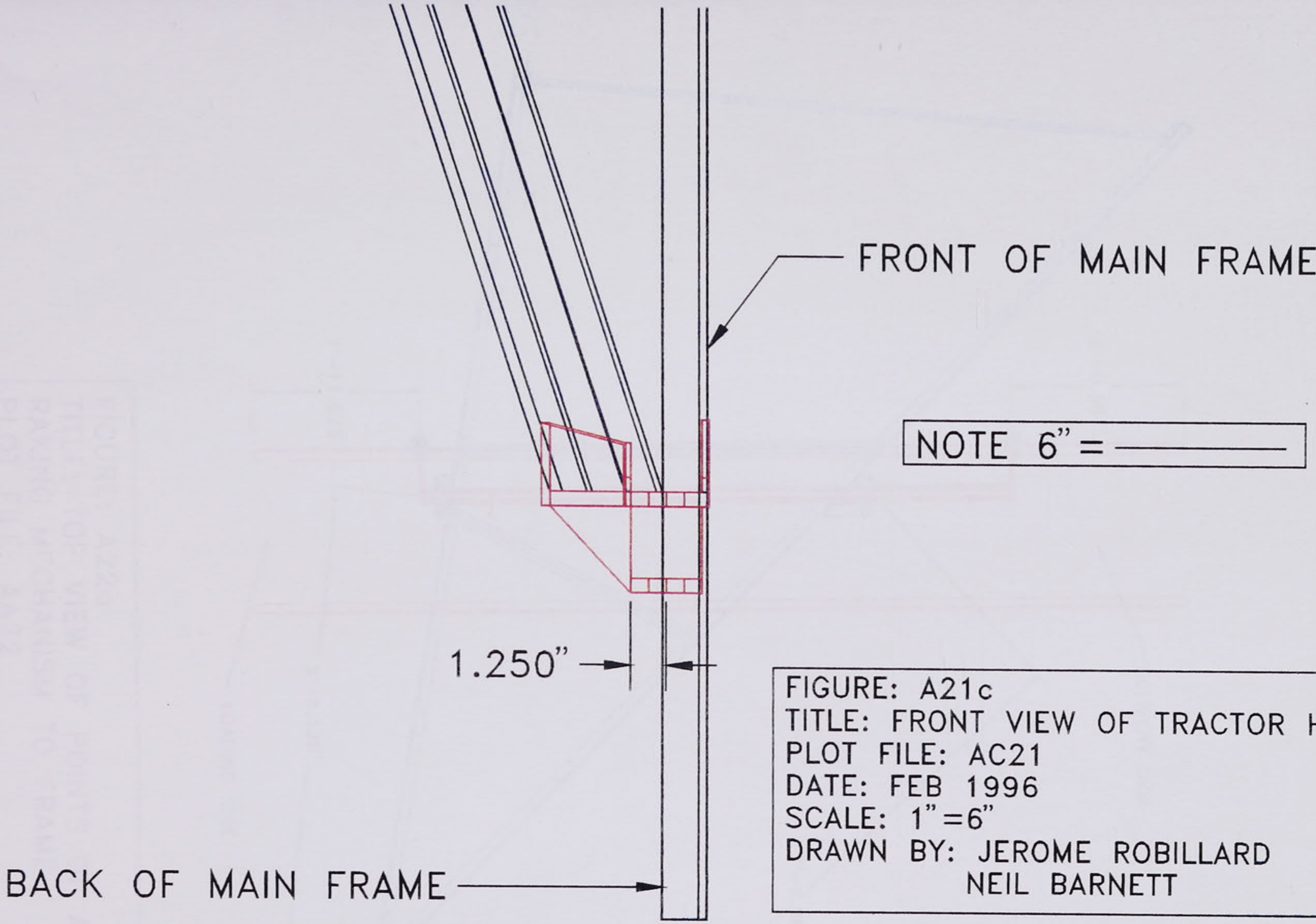


FIGURE: A21c
TITLE: FRONT VIEW OF TRACTOR HITCH
PLOT FILE: AC21
DATE: FEB 1996
SCALE: 1"=6"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

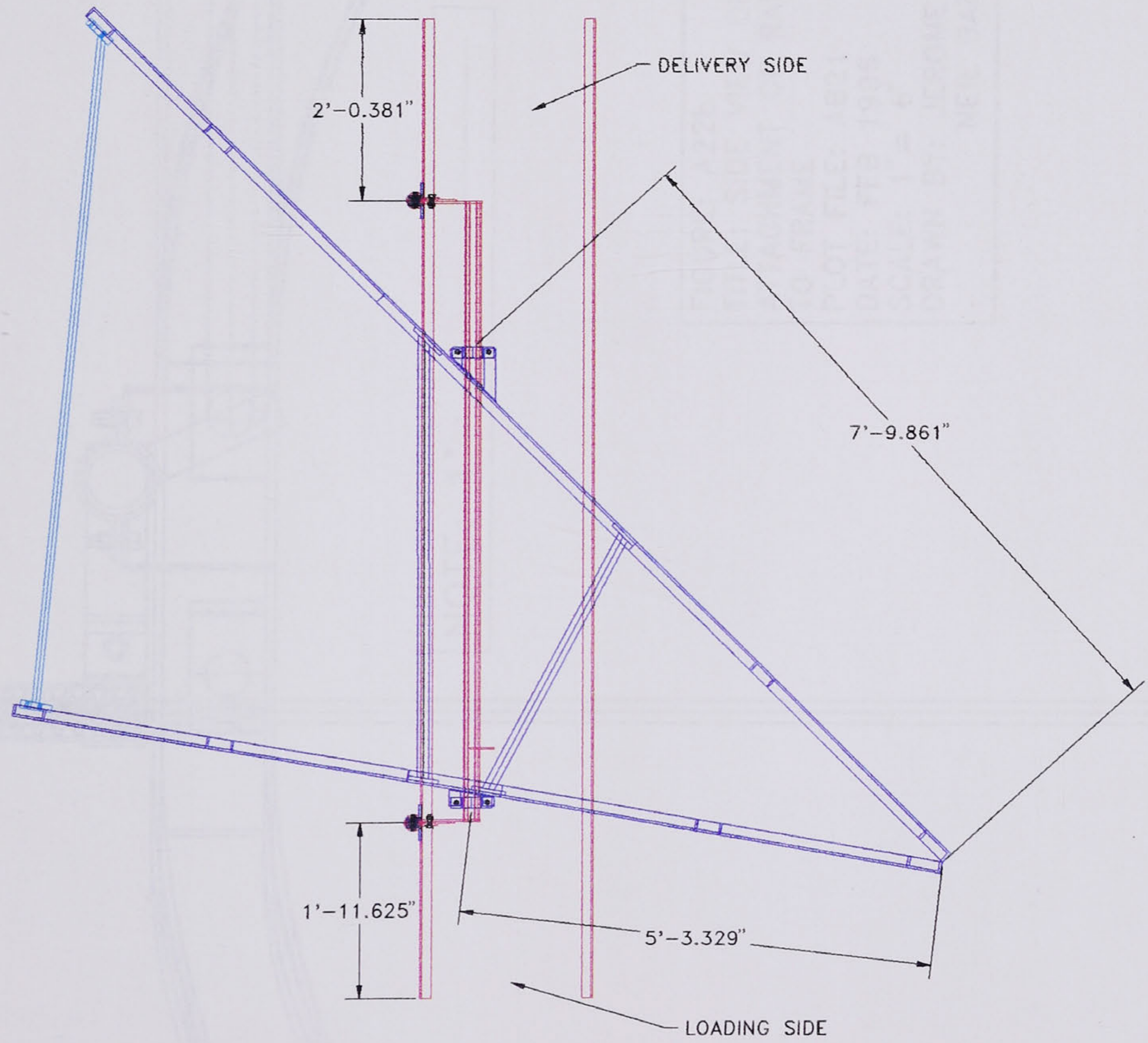


FIGURE: A22a
TITLE: TOP VIEW OF POINTS OF ATTACHMENT OF RAKING MECHANISM TO FRAME
PLOT FILE: AA22
DATE: FEB 1996
SCALE: 1"=2'
DRAW BY: JEROME ROBILLARD
NEIL BARNETT

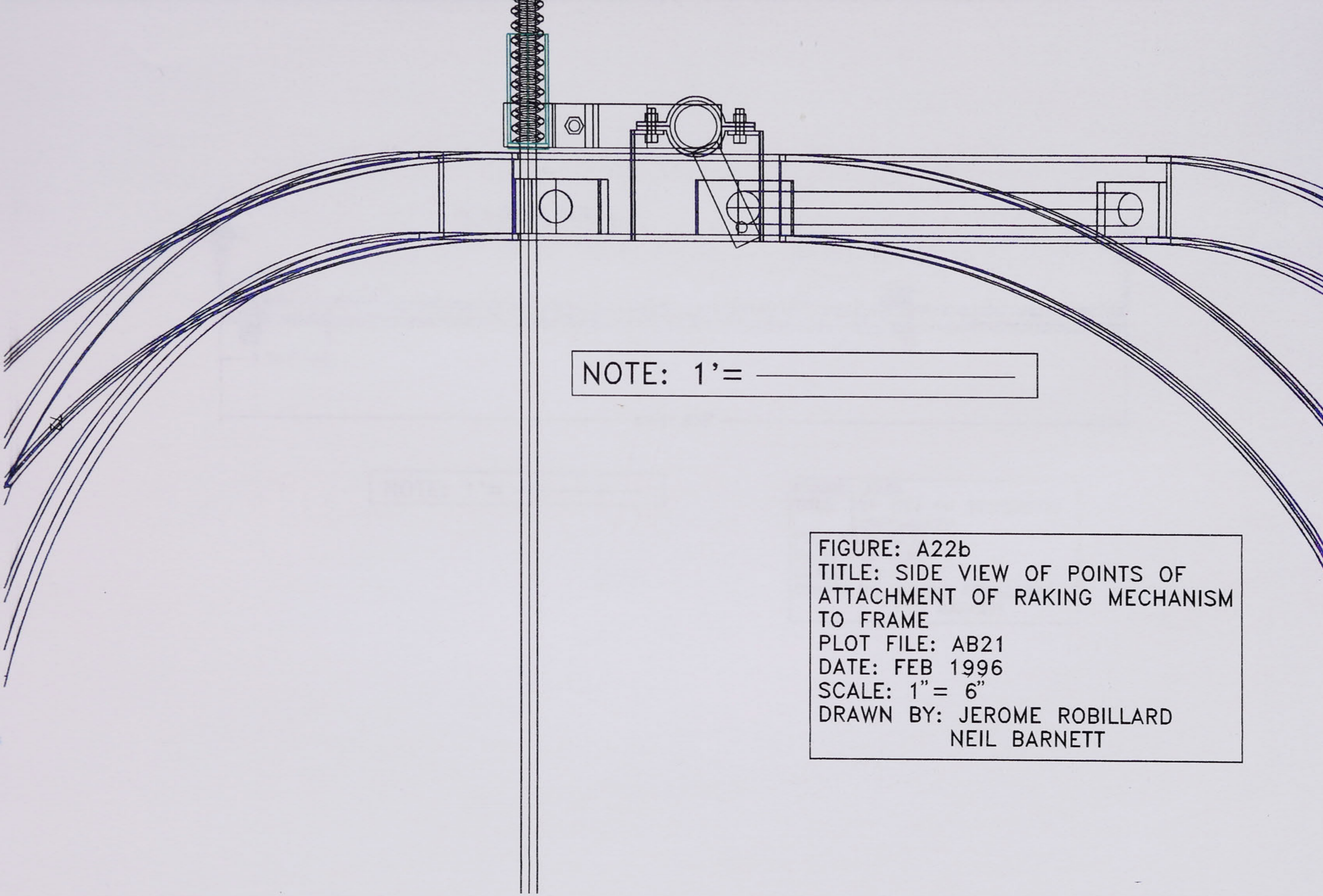
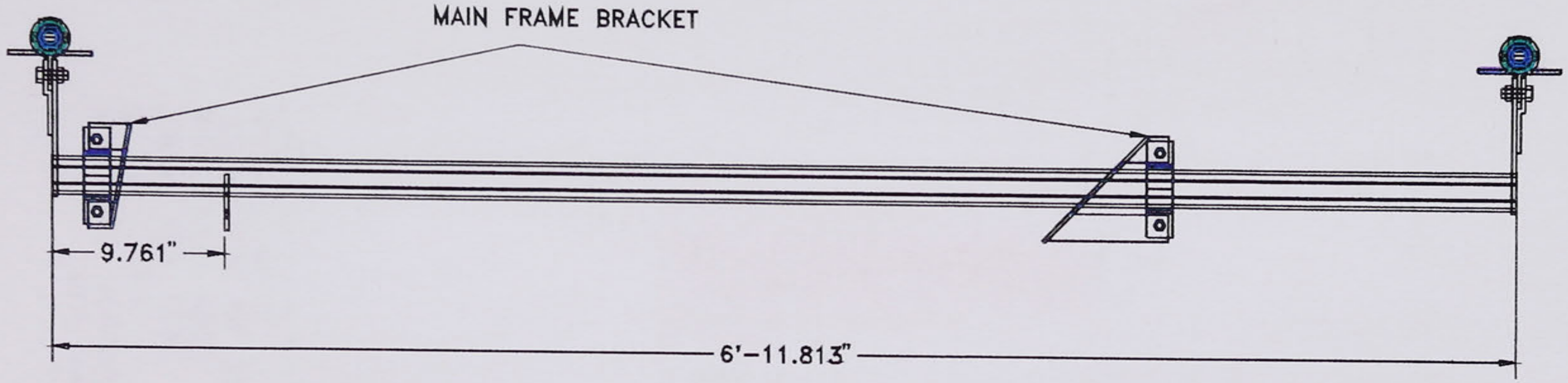


FIGURE: A22b
TITLE: SIDE VIEW OF POINTS OF ATTACHMENT OF RAKING MECHANISM TO FRAME
PLOT FILE: AB21
DATE: FEB 1996
SCALE: 1" = 6"
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ____________

FIGURE: A23a
TITLE: TOP VIEW OF SUSPENSION COMPONENTS
PLOT FILE: AA23
DATE: FEB 1996
SCALE: 1"=1'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

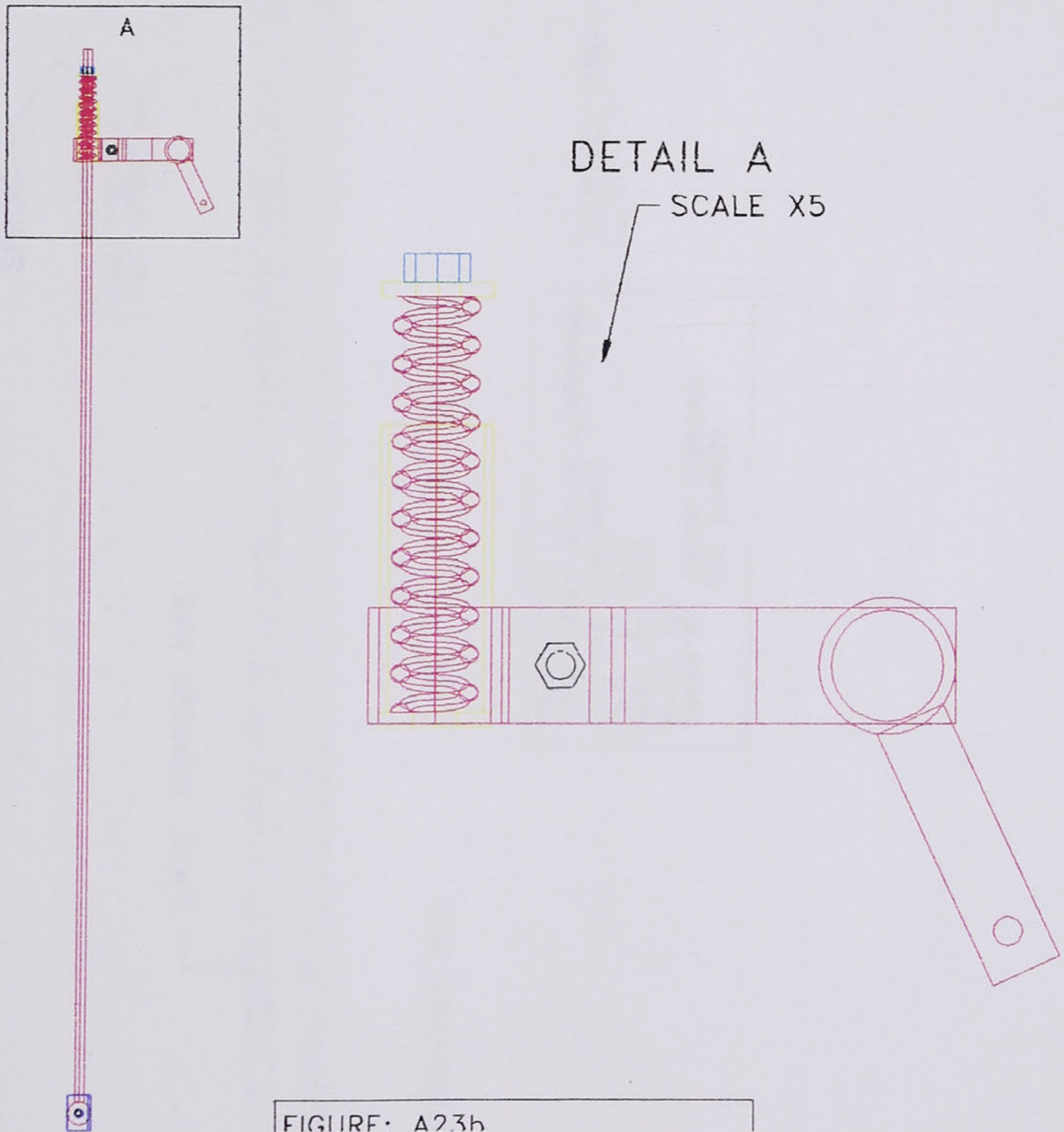


FIGURE: A23b
TITLE: SIDE OF SUSPENSION
PLOT FILE: AB23
DATE: FEB 1996
SCALE: 1"=1.5'
DRAW BY: JEROME ROBILLARD
NEIL BARNETT

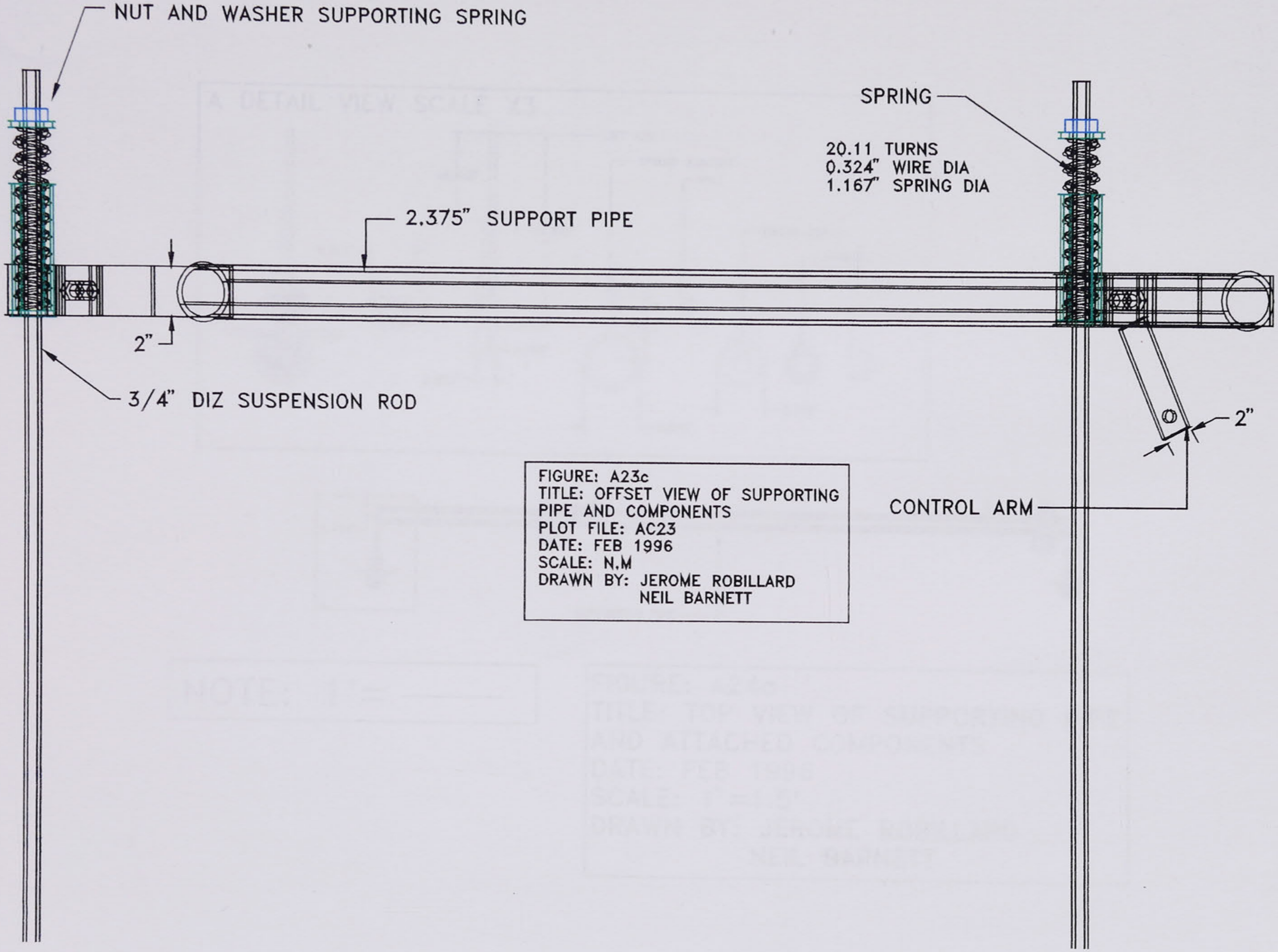
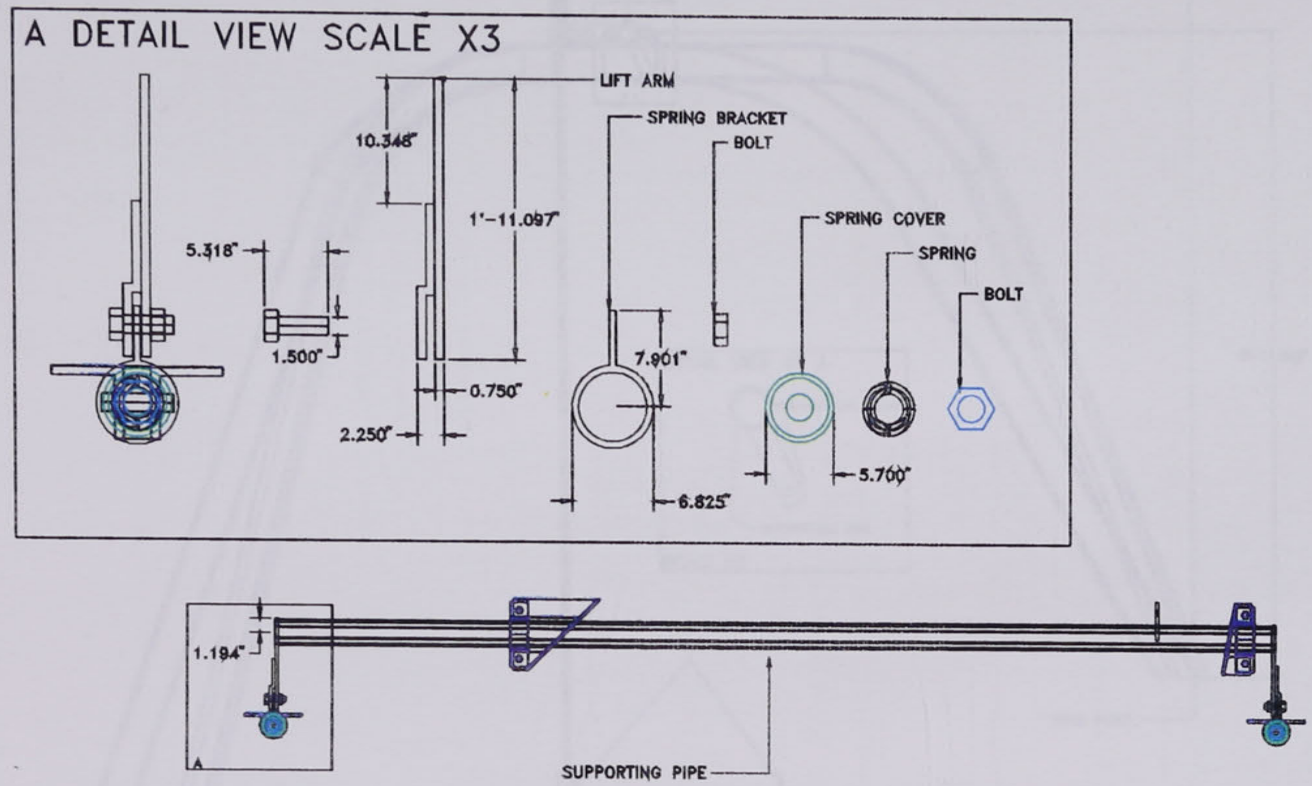
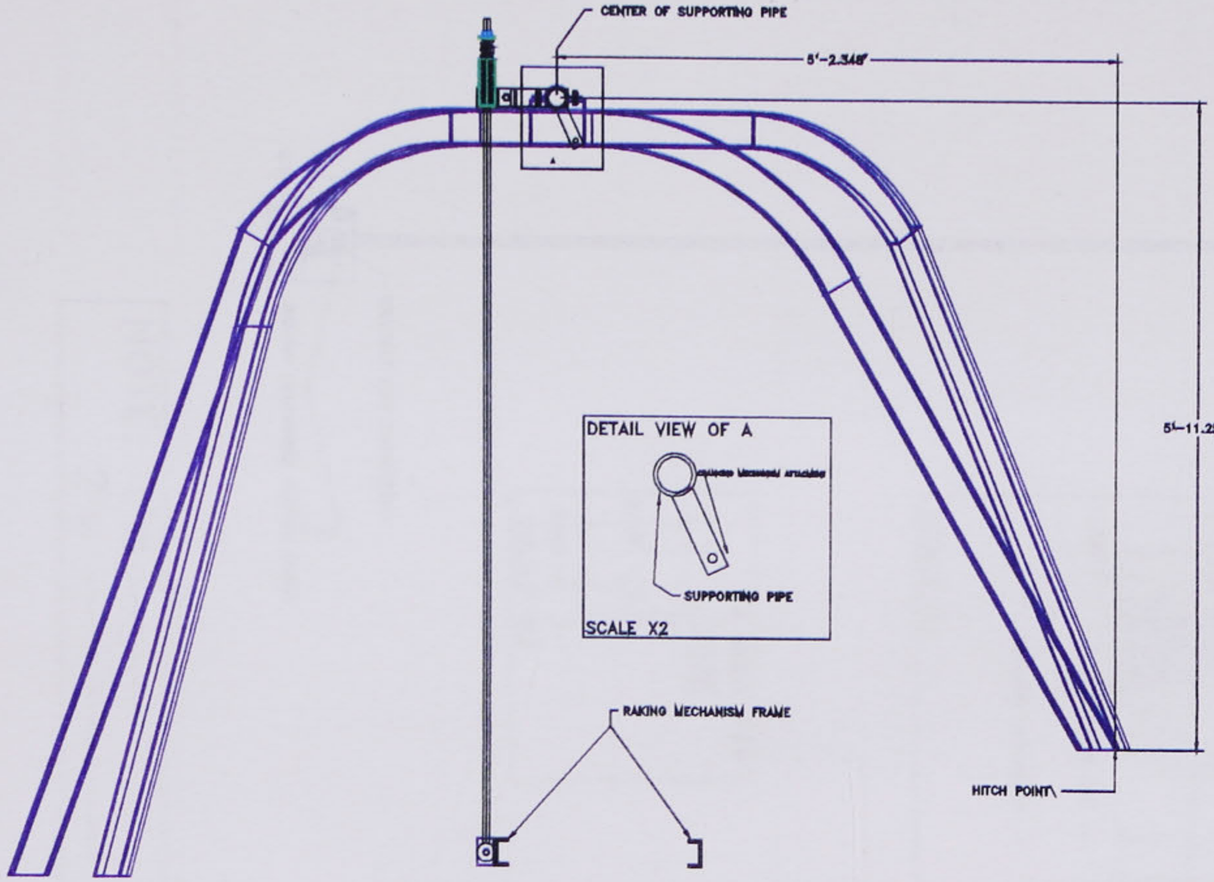


FIGURE: A23c
TITLE: OFFSET VIEW OF SUPPORTING PIPE AND COMPONENTS
PLOT FILE: AC23
DATE: FEB 1996
SCALE: N.M
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



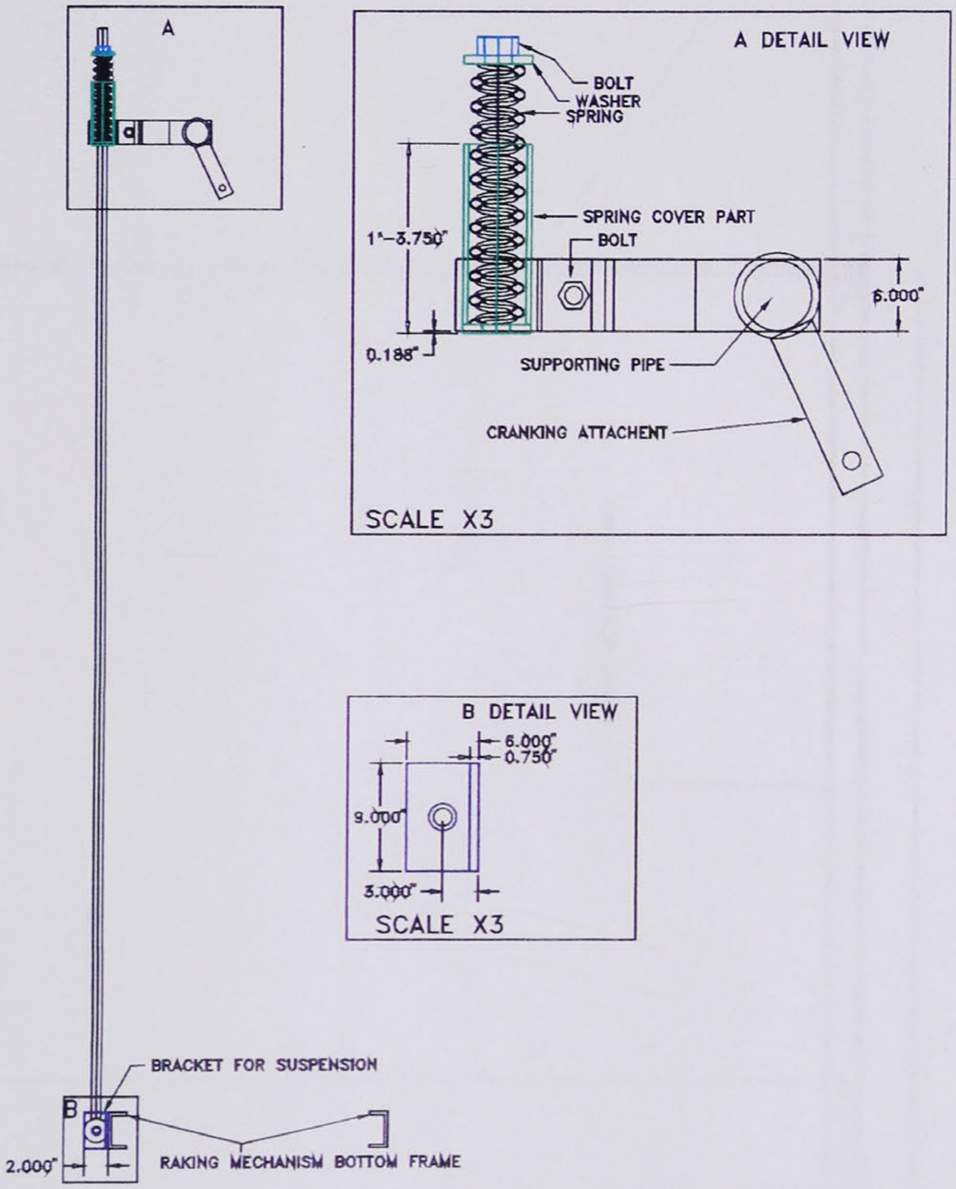
NOTE: 1'= ———

FIGURE: A24a
TITLE: TOP VIEW OF SUPPORTING PIPE AND ATTACHED COMPONENTS
DATE: FEB 1996
SCALE: 1"=1.5'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



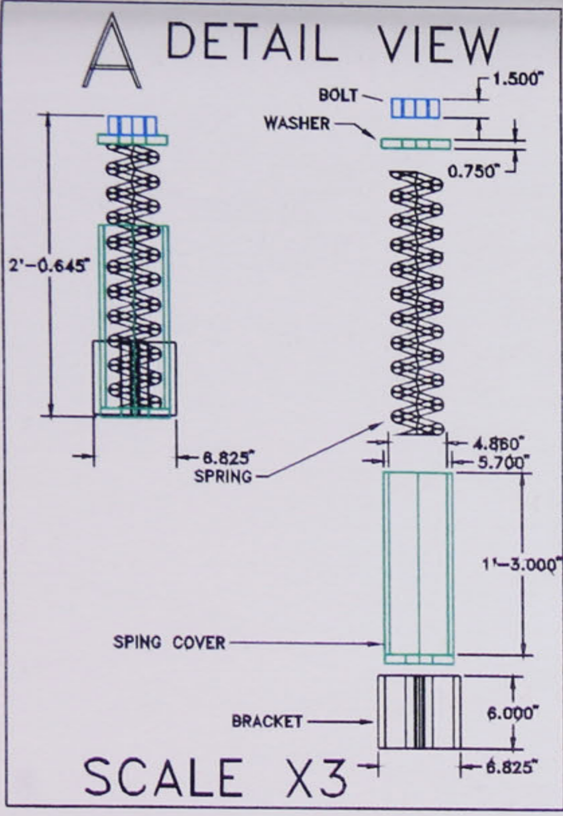
NOTE: 1'= ______
2'= __________

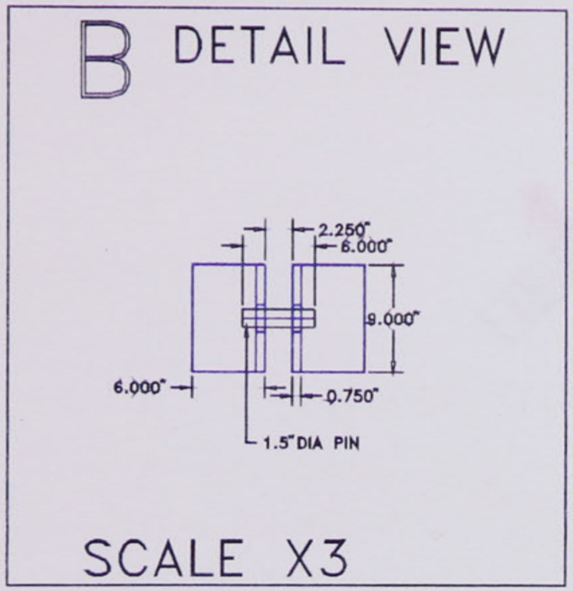
FIGURE: A24b
TITLE: SIDE VIEW OF SUPPORTING PIPE AND ATTACHED COMPONENTS
PLOT FILE: AB24
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

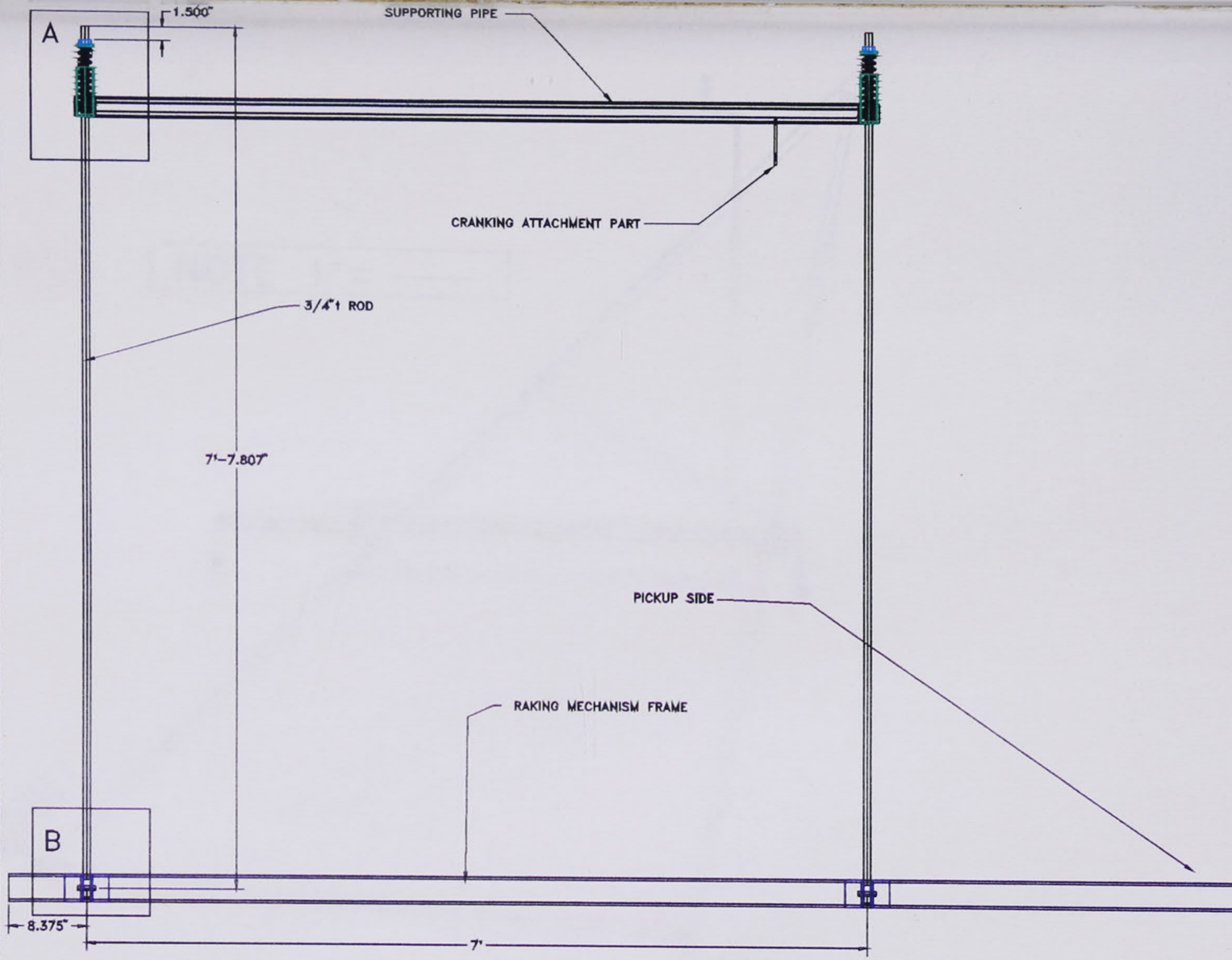


NOTE: 1'= ______
2'= ______

FIGURE: A25a
TITLE: SIDE VIEW OF REAR SUSPENSION ROD AND BOTTOM
PLOT FILE: AA25
DATE: FEB 1996
SCALE: 1"=1.5'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT







NOTE: 1'= ________

2'= ____________

FIGURE: A25b
TITLE: REAR VIEW OF REAR SUSPENSION ROD AND BOTTOM BRACKET
PLOT FILE: AB25
DATE: FEB 1996
SCALE: 1"=1.5'
DRAW BY: JEROME ROBILLARD
NEIL BARNETT

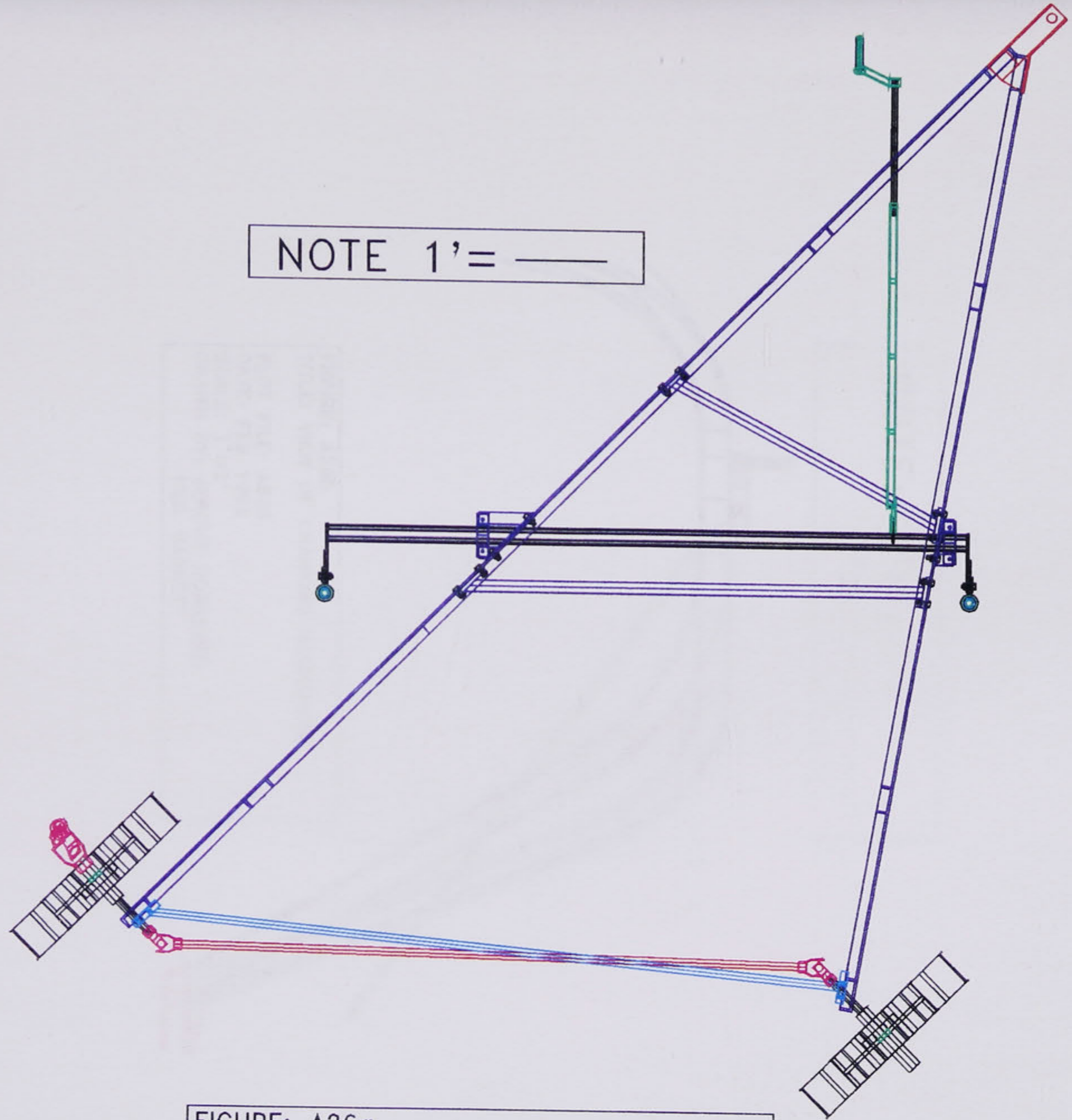


FIGURE: A26a
TITLE: TOP VIEW OF CRANKING MECHANISM
PLOT FILE: AA26
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

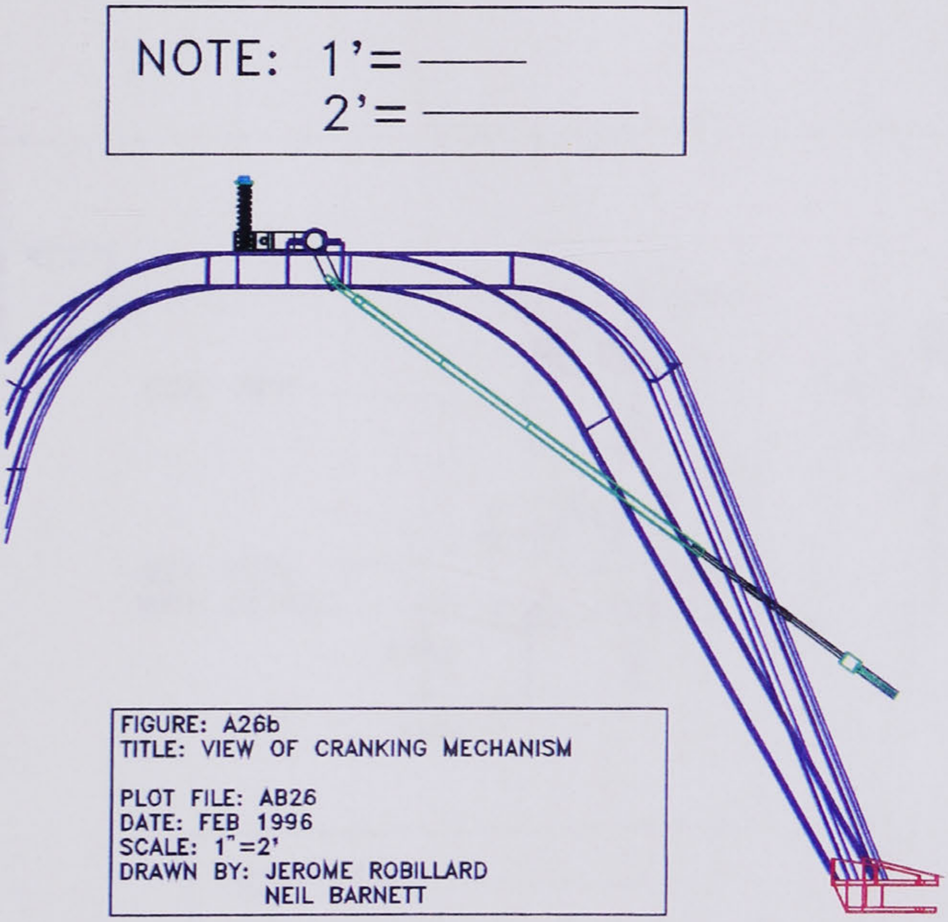
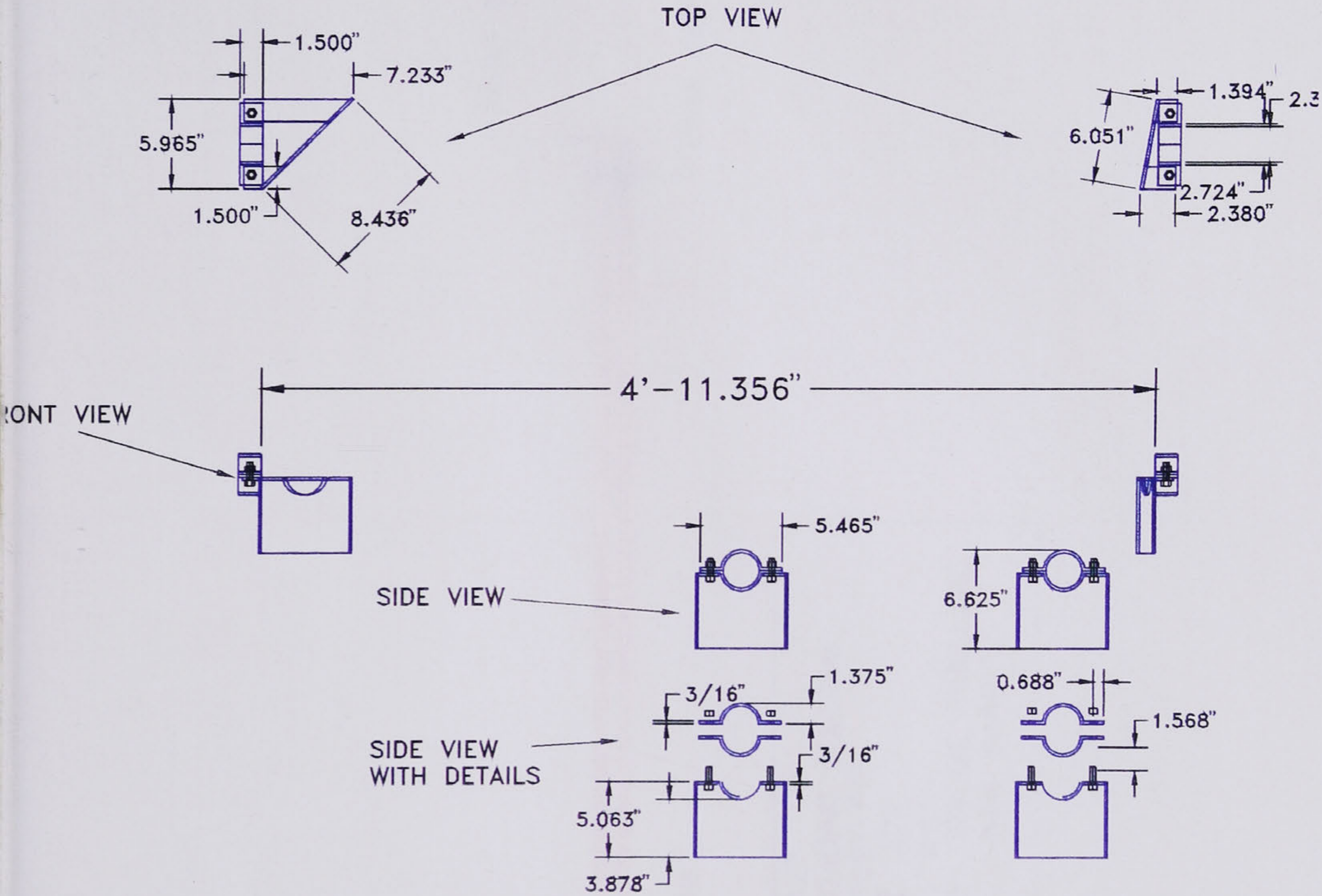


FIGURE: A26b
TITLE: VIEW OF CRANKING MECHANISM

PLOT FILE: AB26
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ______________

2'= ______________

FIGURE: A26c
TITLE: BRACKET OF CRANKING MECHANISM
PLOT FILE: AC26
DATE: FEB 1996
SCALE: 1"=1'
DRAW BY: JEROME ROBILLARD
NEIL BARNETT

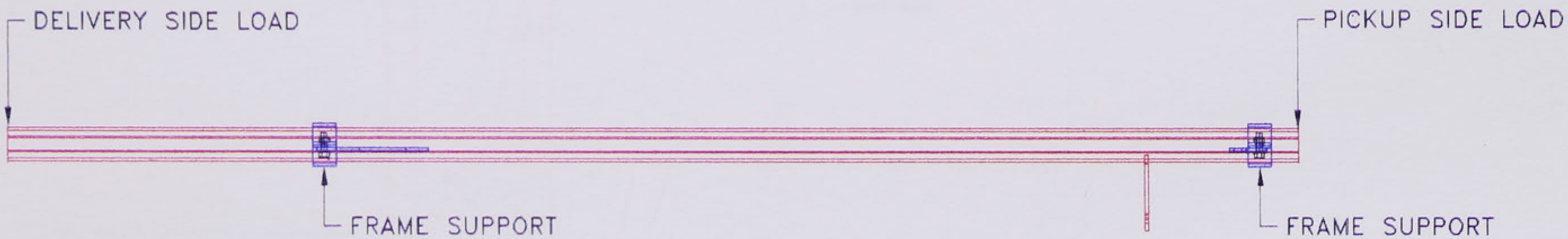
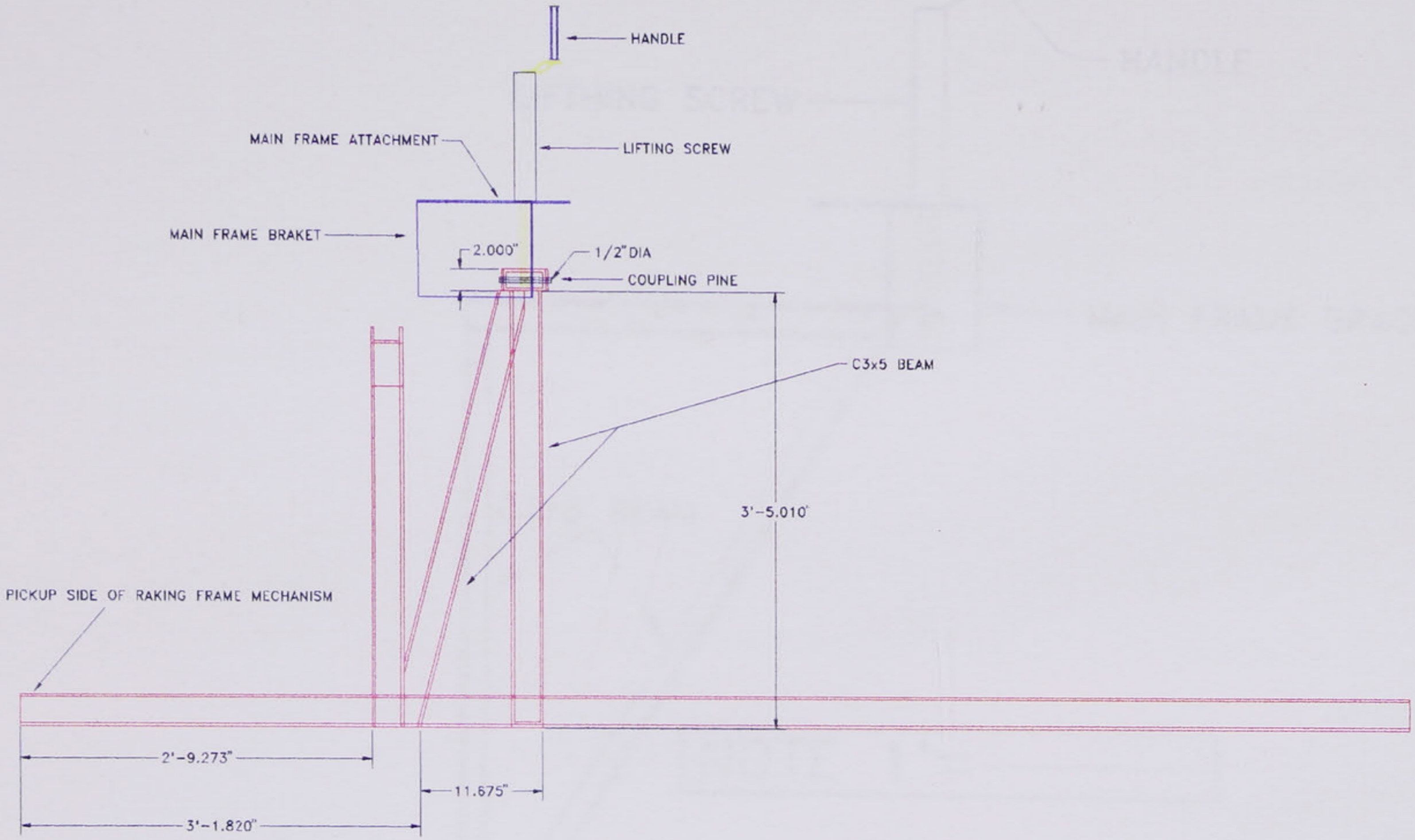


FIGURE: A27
TITLE: LOADING DIAGRAM
FOR SUPPORT PIPE
PLOT FILE: A27
SCALE: 1"=1'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ________
2'= ____________

FIGURE: A28a
TITLE: FRONT VIEW OF SUSPENSION ATTACHMENT TO RAKING MECHANISM
PLOT FILE: AA28
DATE: FEB 1996
SCALE: 1"=1.5'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

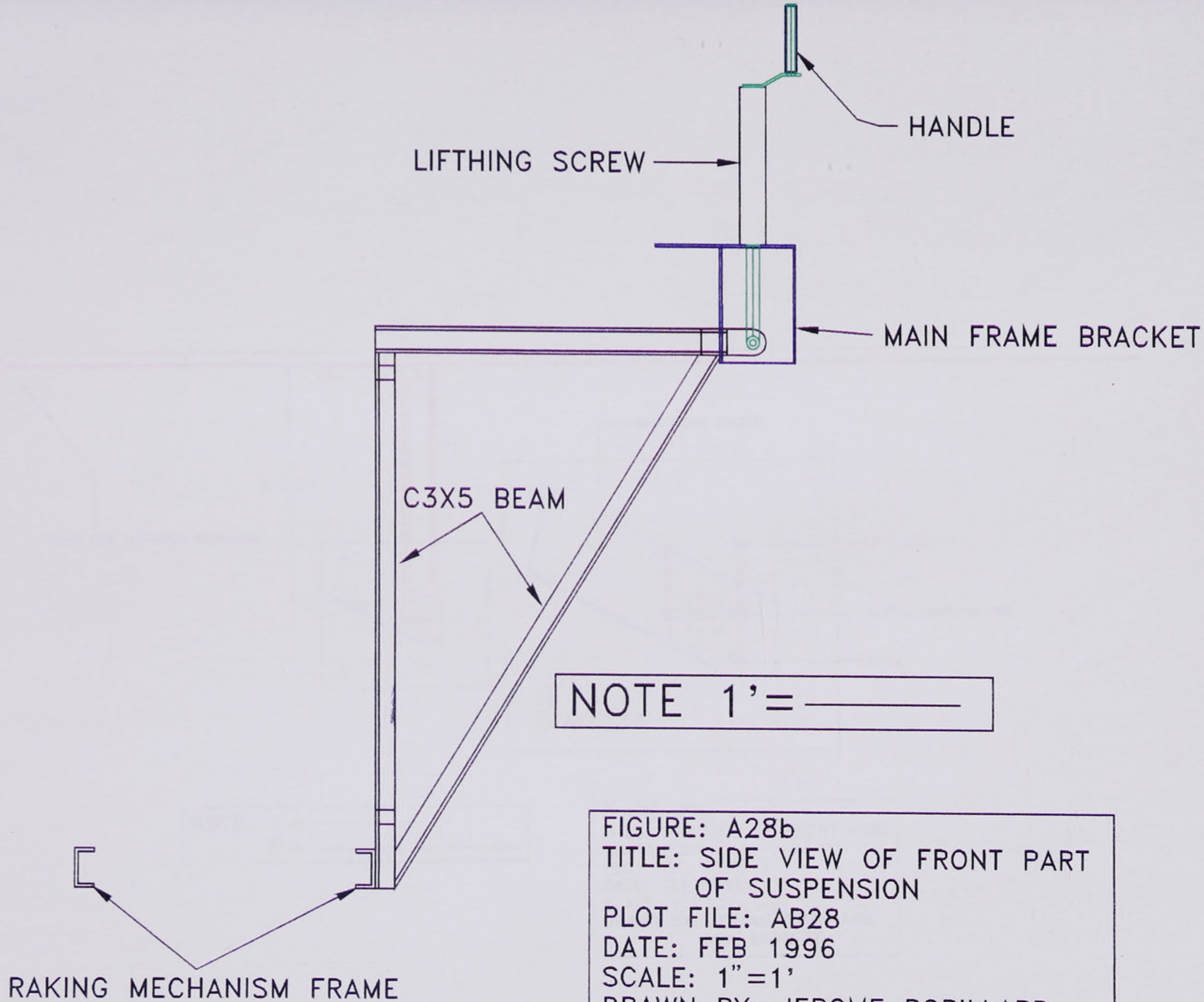
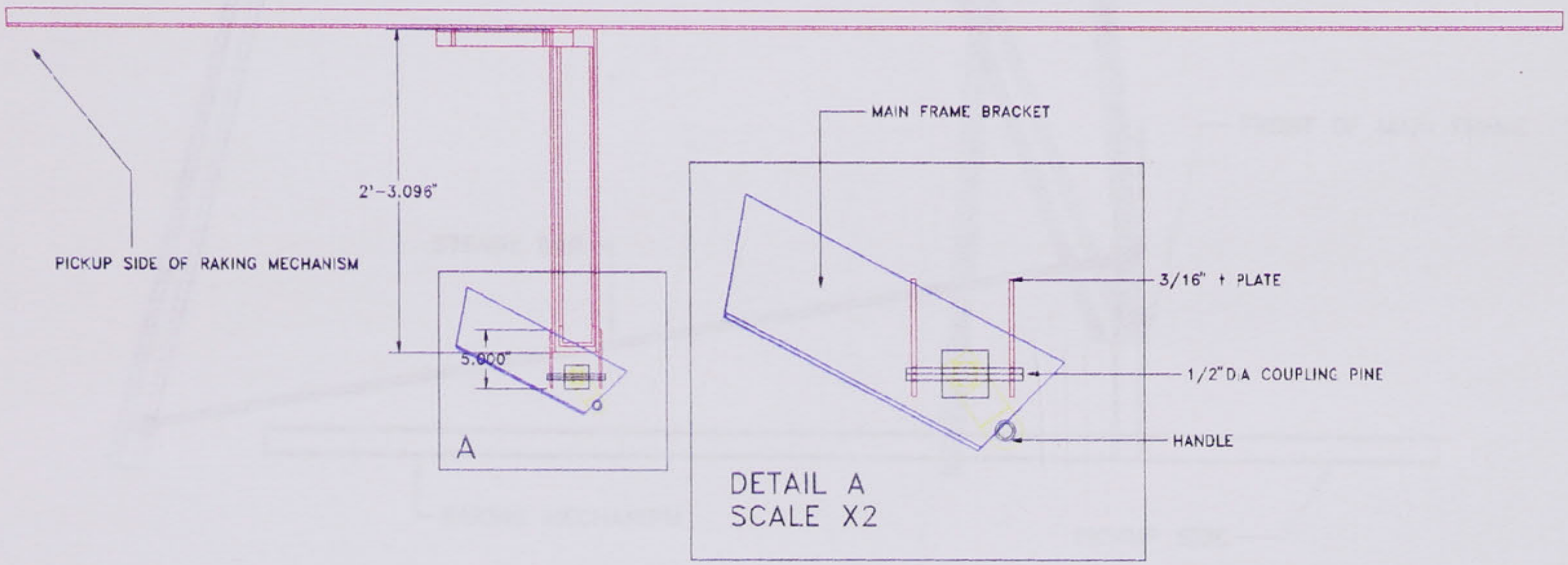
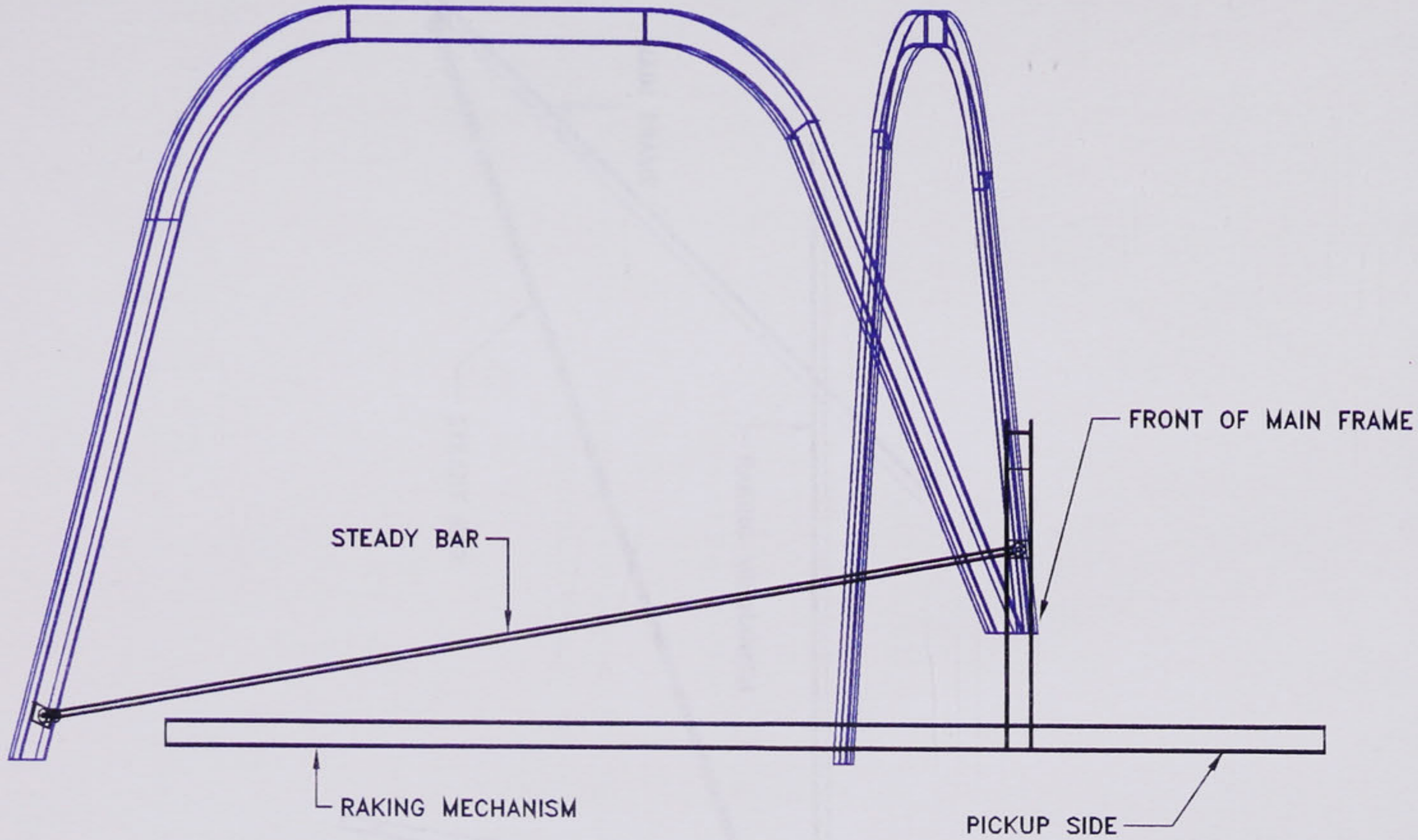


FIGURE: A28b
TITLE: SIDE VIEW OF FRONT PART OF SUSPENSION
PLOT FILE: AB28
DATE: FEB 1996
SCALE: 1"=1'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



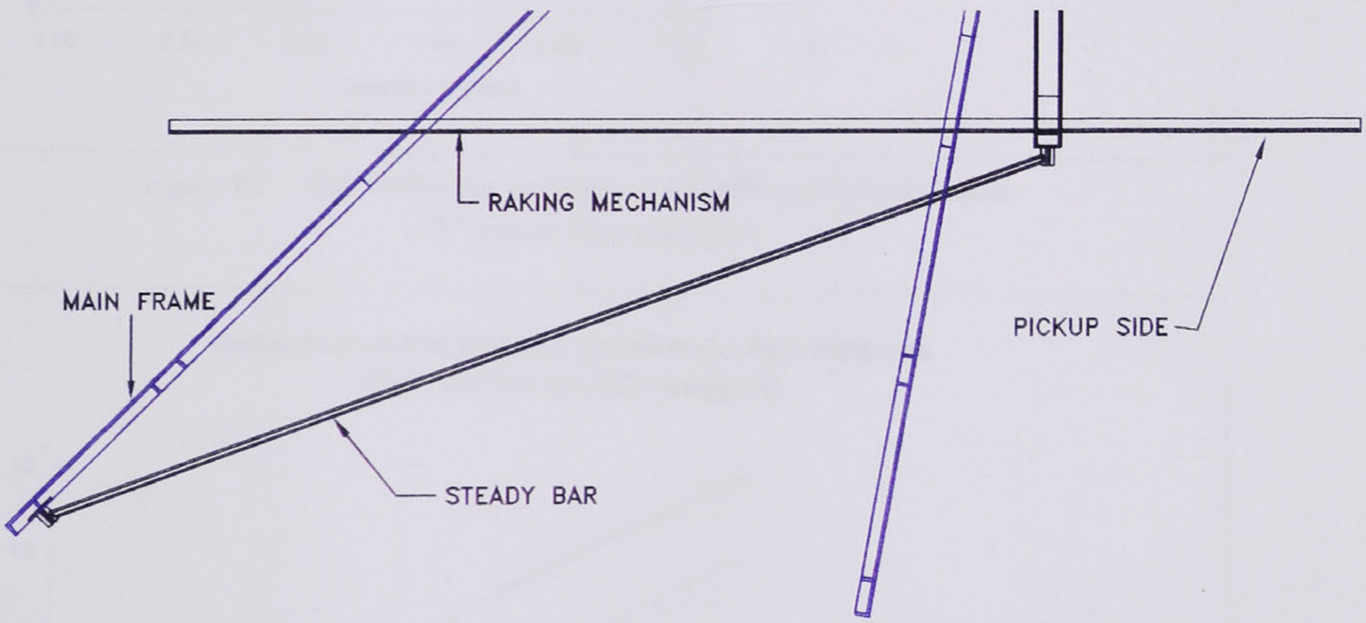
NOTE: 1'= ________
2'= ________

FIGURE: A28c
TITLE: TOP VIEW OF FRONT PART SUSPENSION
PLOT FILE: AC28
DATE: FEB 1996
SCALE: 1"=1.5'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ———
2'= ——————

FIGURE: A29a
TITLE:REAR VIEW OF STEADY BAR
PLOT FILE: AA29
DATE: FEB 1996
SCALE: 1"=2'
DRAW BY: JEROME ROBILLARD
NEIL BARNETT



NOTE: 1'= ———
2'= ————————

FIGURE: A29b
TITLE: TOP VIEW OF STEADY BAR AT REAR OF MAIN FRAME
PLOT FILE: AC29
DATE: FEB 1996
SCALE: 1"=2'
DRAWN BY: JEROME ROBILLARD
NEIL BARNETT

Appendix B - Data

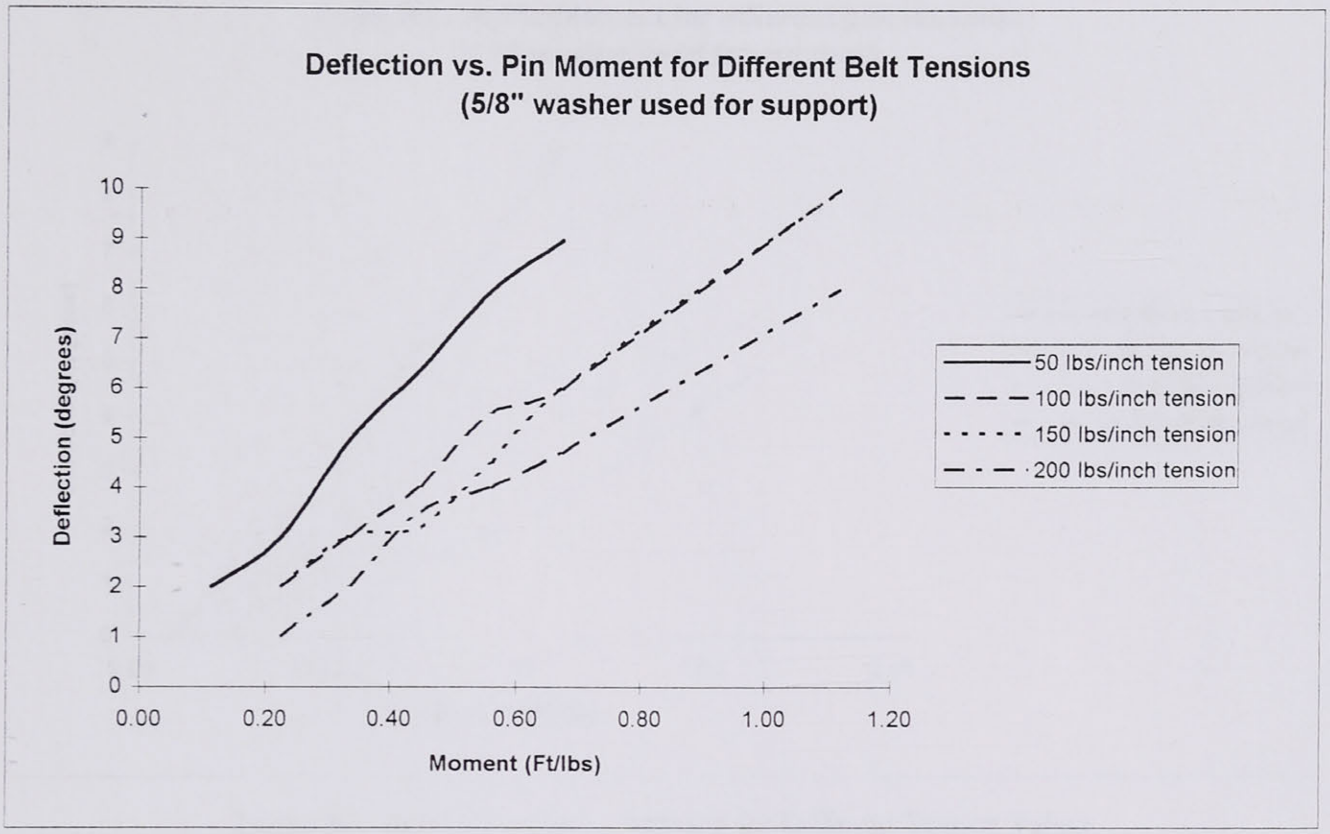


Figure B1 Belt Deflection vs. Moment for Different Tension Values
(5/8" washer supporting pin)

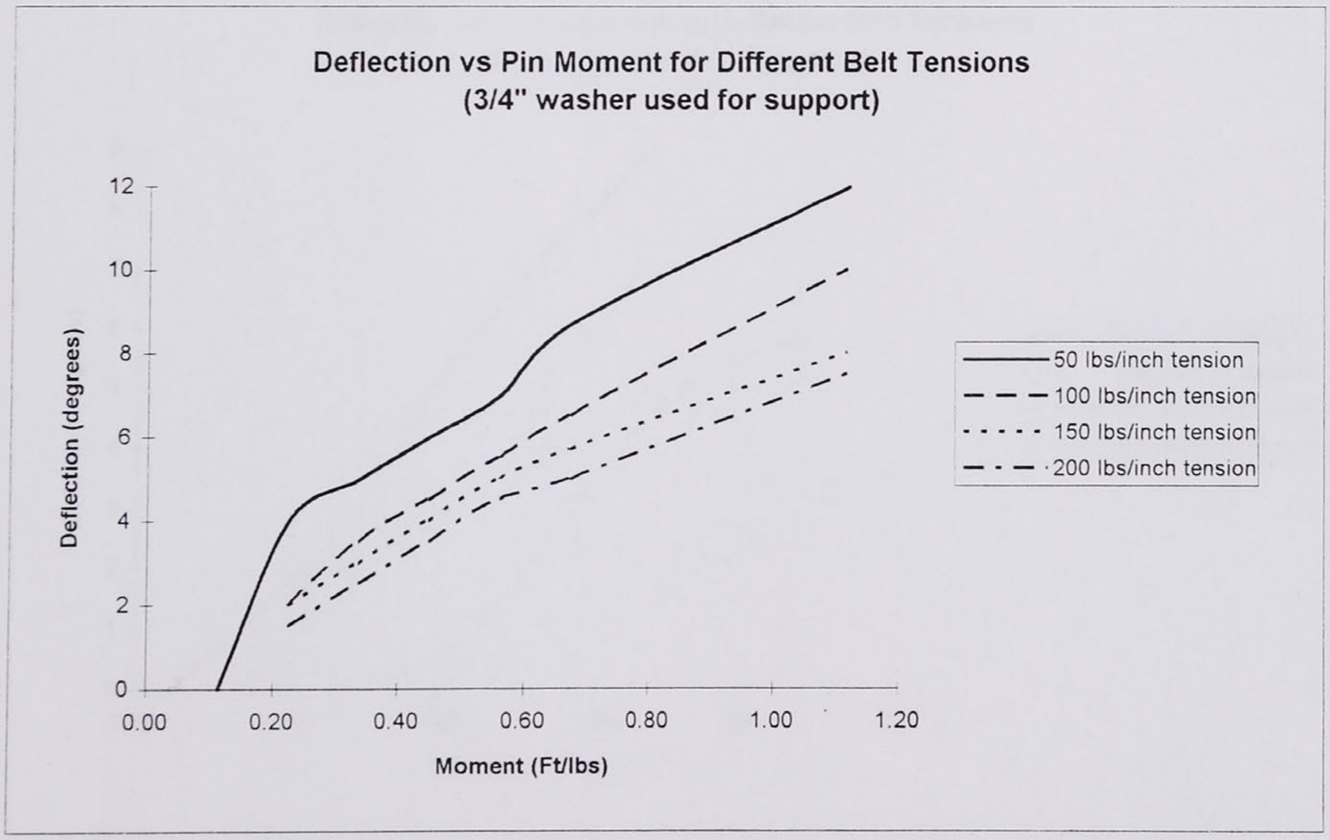


Figure B2 Belt Deflection vs. Moment for Different Tension Values
(3/4" washer supporting pin)

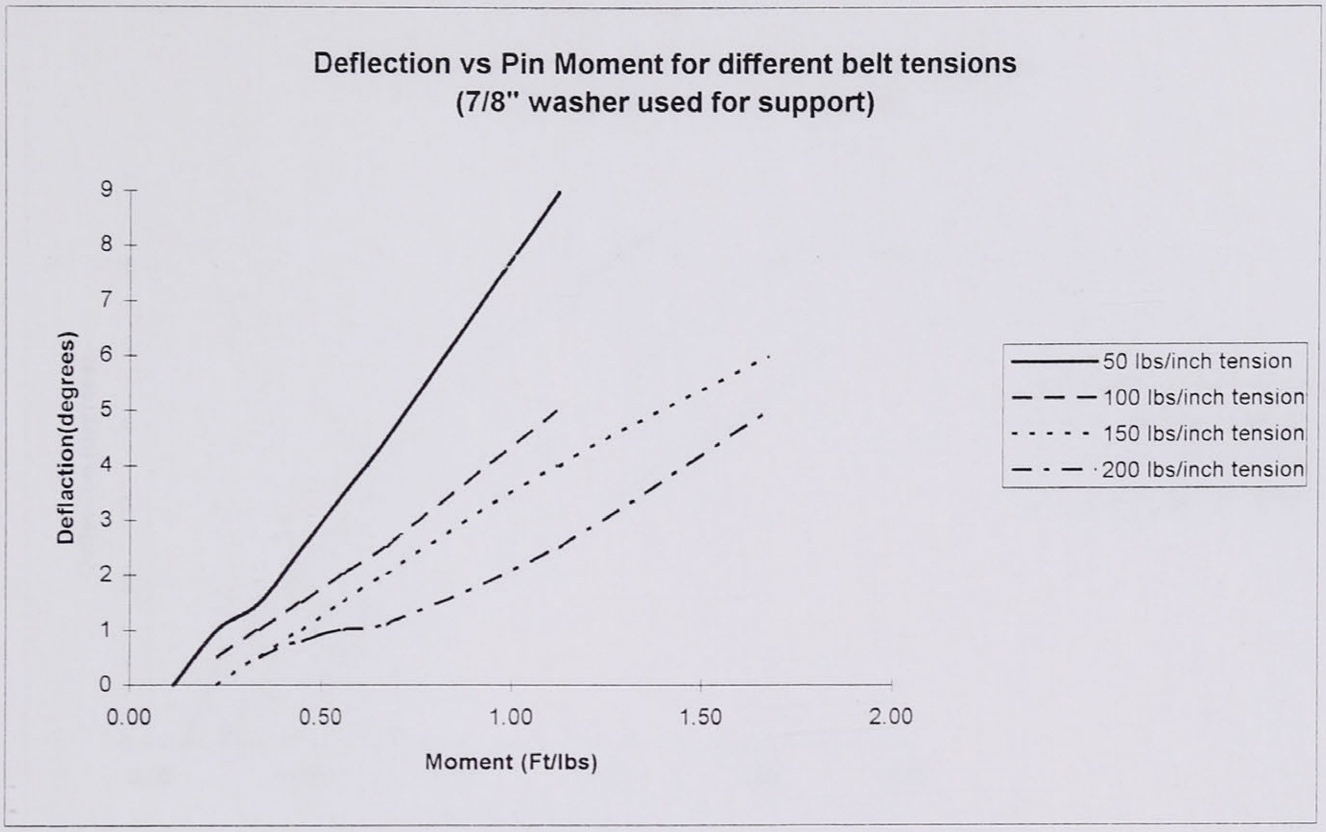


Figure B3 Belt Deflection vs. Moment for Different Tension Values
(7/8" washer supporting pin)

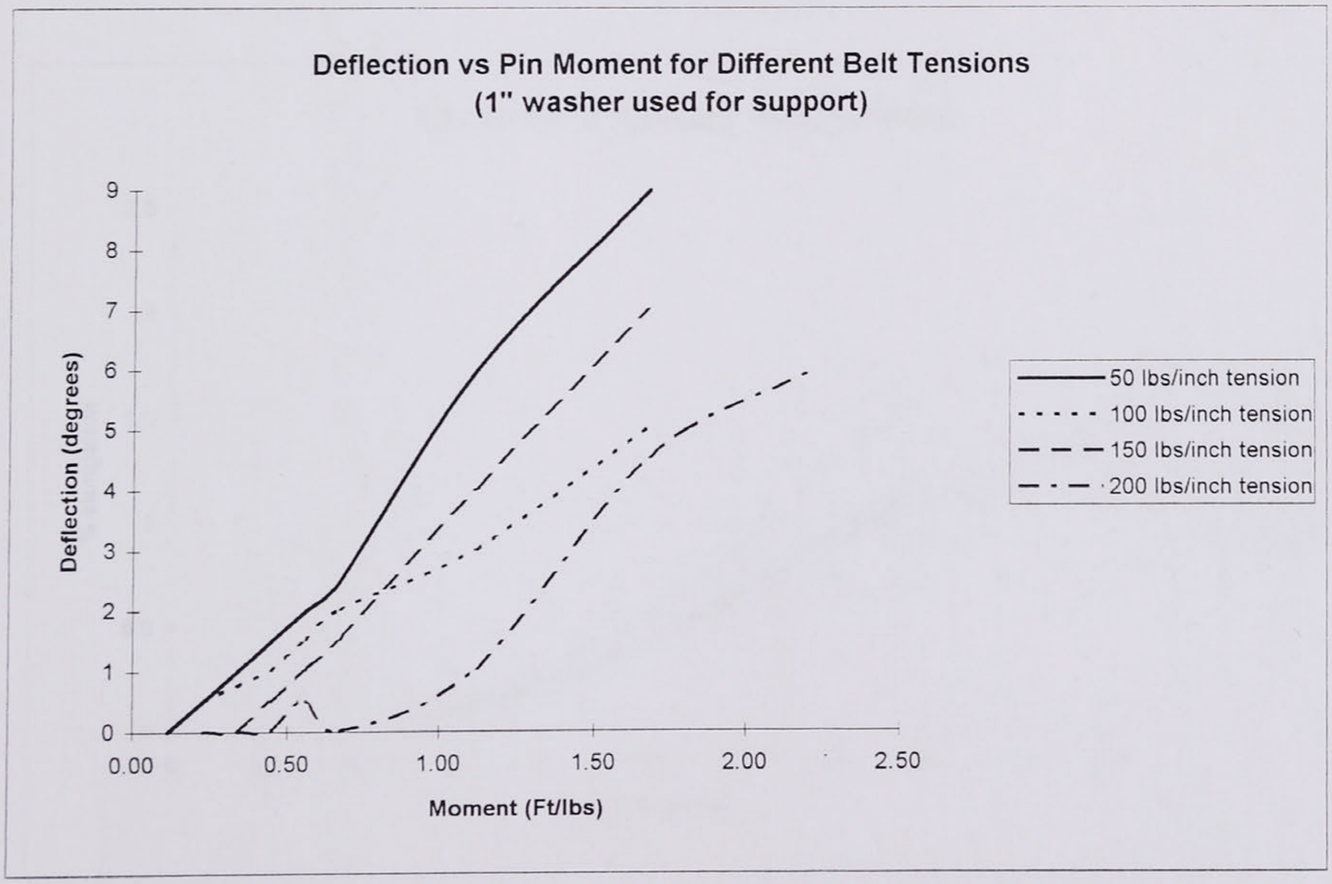


Figure B4 Belt Deflection vs. Moment for Different Tension Values
(1" washer supporting pin)

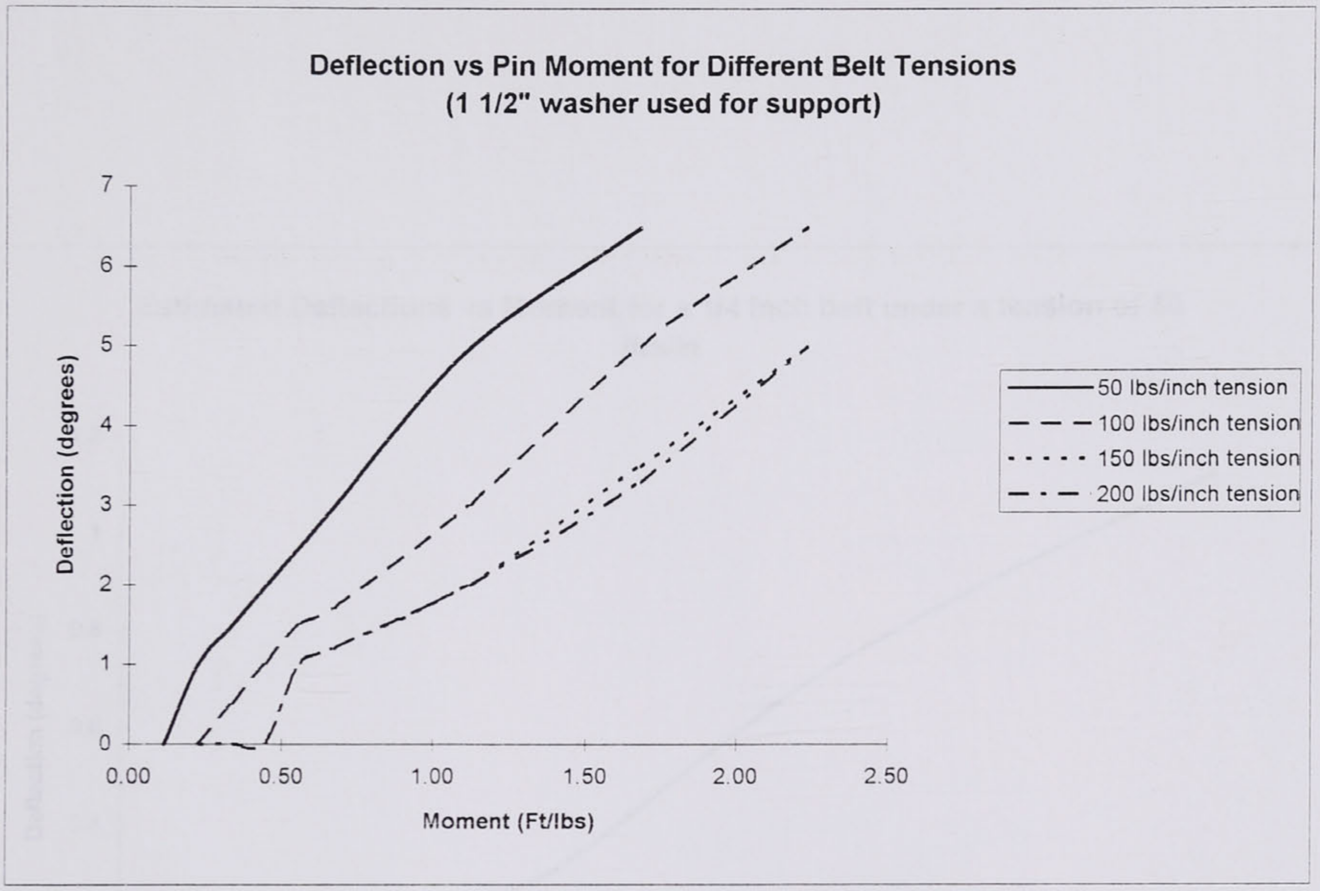


Figure B5 Belt Deflection vs. Moment for Different Tension Values
(1 1/2" washer supporting pin)

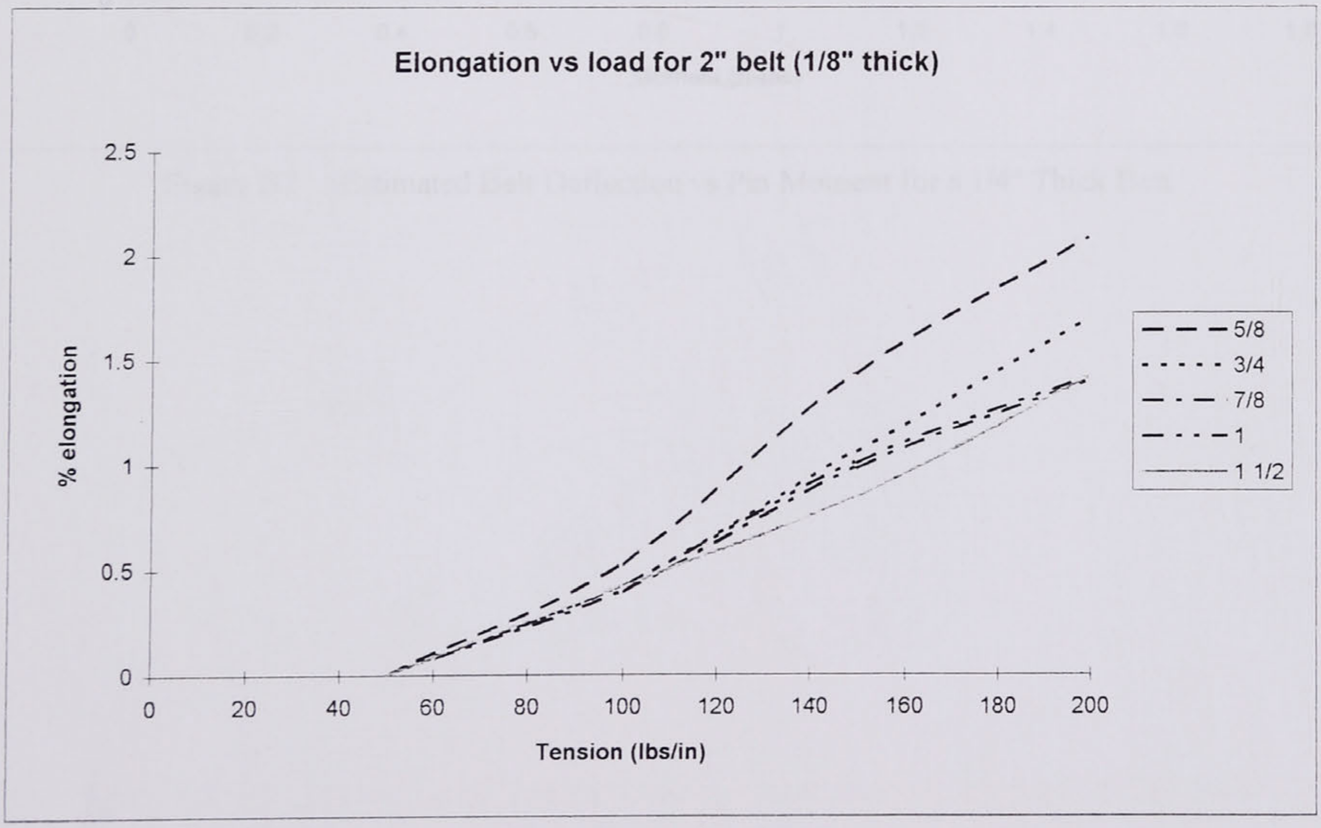


Figure B6 Elongation vs. Load for 1/8" Reinforced Rubber Belt
(2" wide specimen, 1/4 inch hole in center with different size of washers used to support pin)

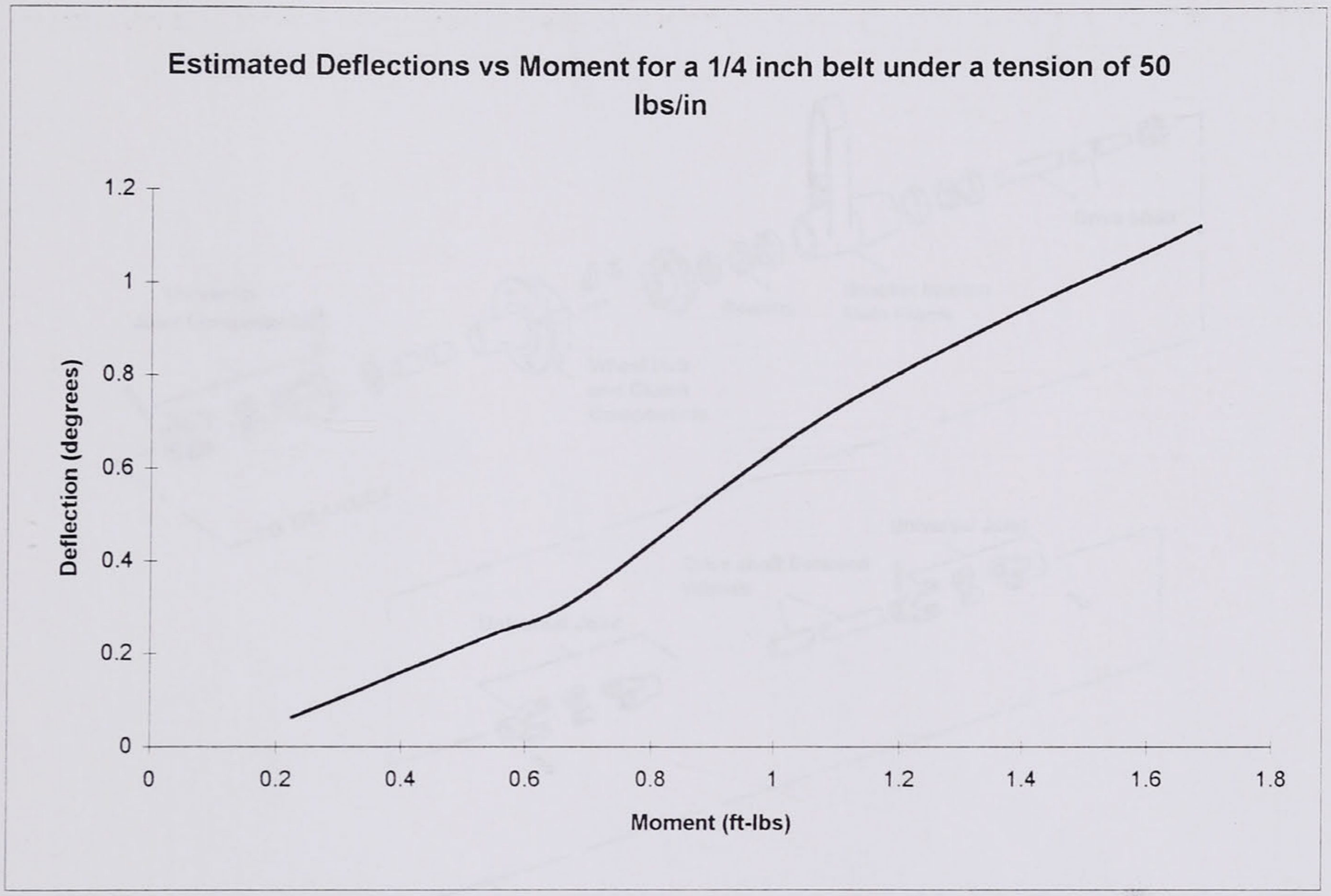


Figure B7 Estimated Belt Deflection vs Pin Moment for a 1/4" Thick Belt

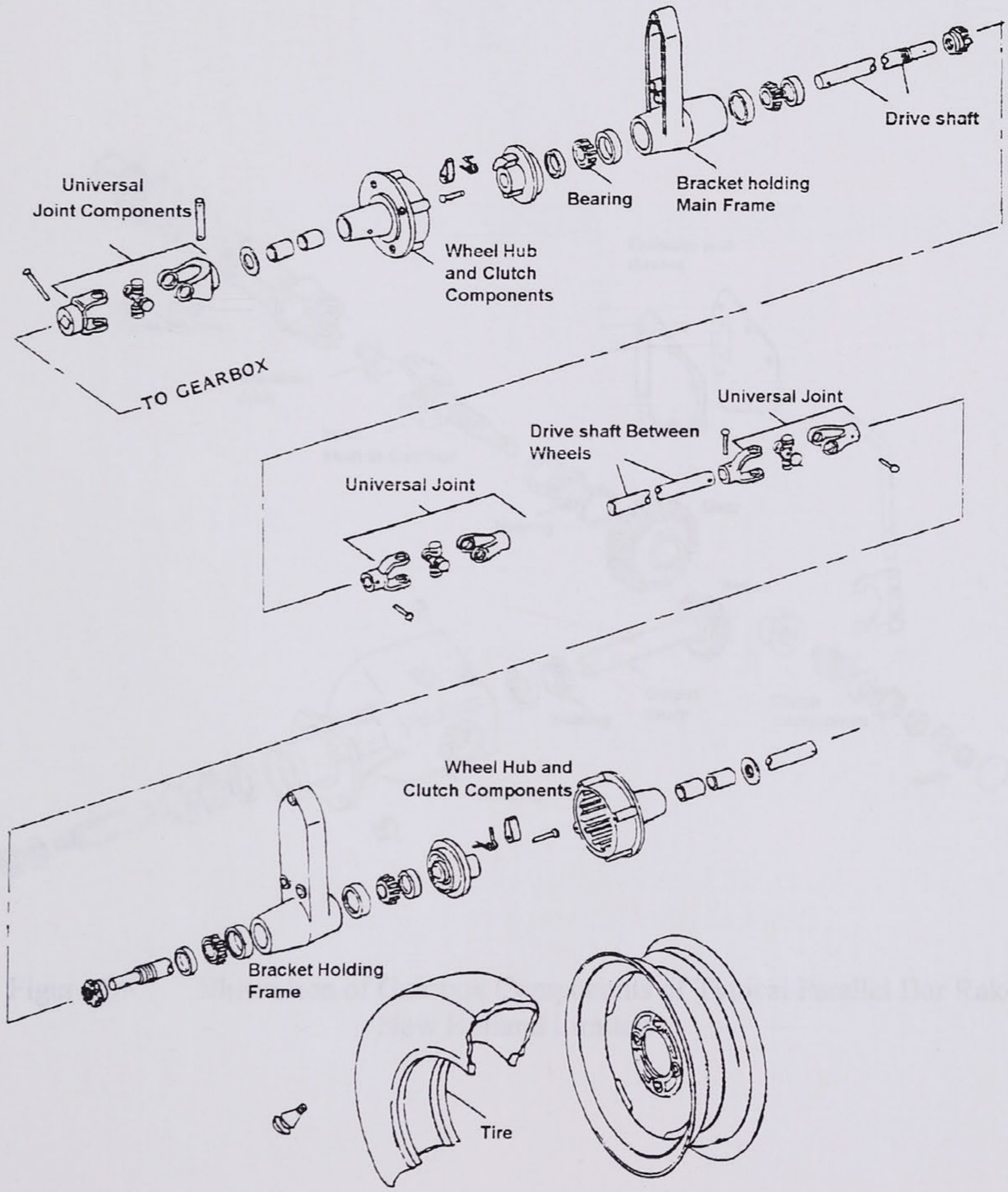


Figure B8 Illustration of Typical Driving Components for a Parallel Bar Rake
New Holland Limited

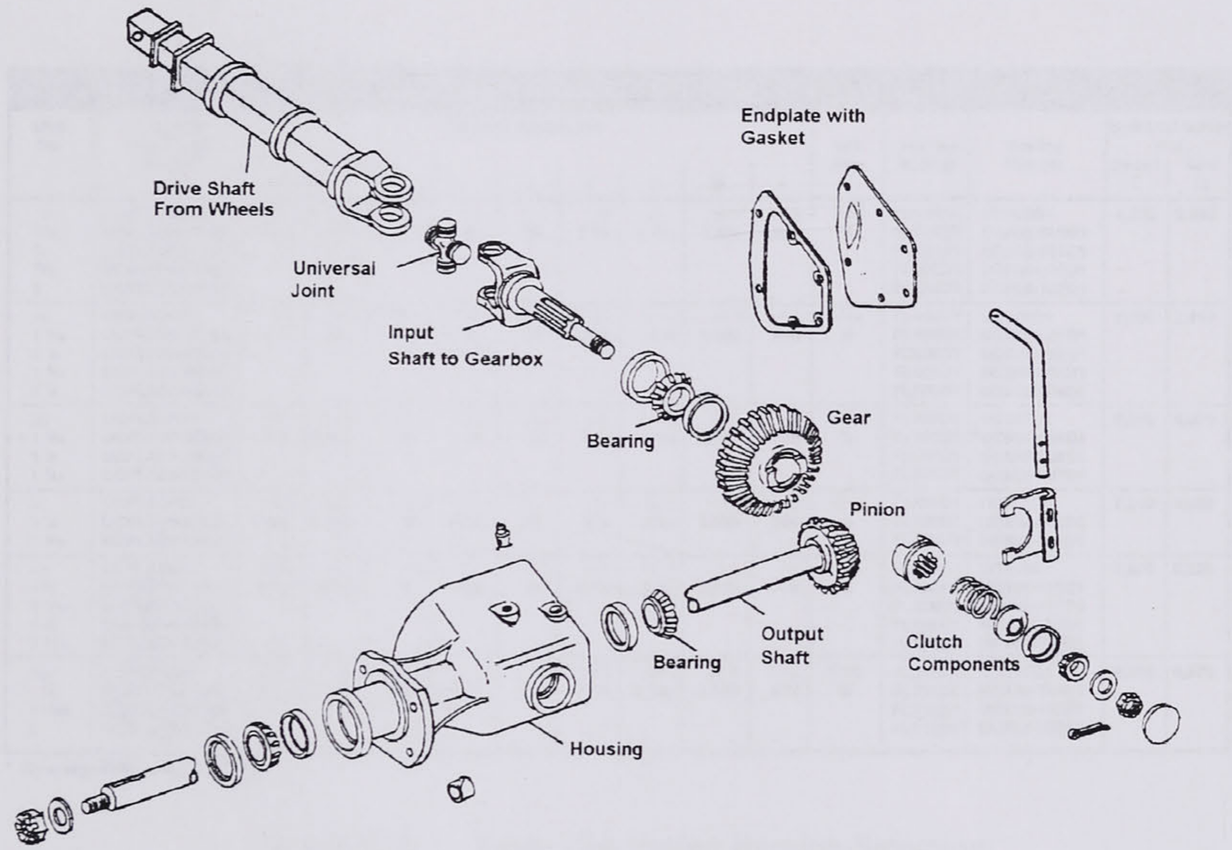


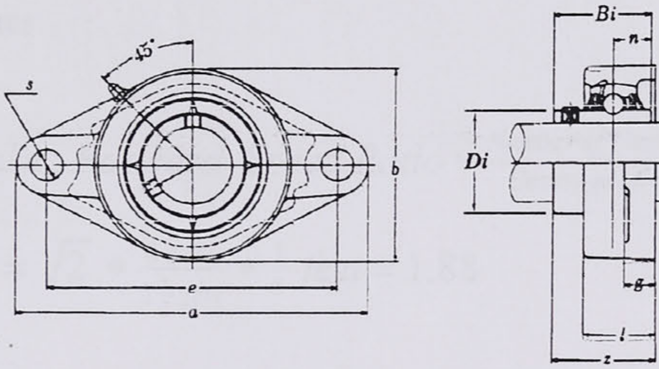
Figure B9 Illustration of Gearbox Components of Typical Parallel Bar Rake
New Holland Limited

NTN

SERIES UCFLX

SET SCREW TYPE

Units: Millimeters
INCHES



FLANGED UNITS - CAST HOUSING (TWO BOLT)

Shaft Dia.	Complete Flanged Unit Number	Nominal Dimensions a	e	g	l	s	b	z	Bi	n	Bolt Size	Housing Number	Bearing Number	Basic Load Ratings (lbs) Dynamic C	Static Co
25	UCFLX05D1	141	117	13	30	12	83	40.2	38.1	15.9	M10	FLX05D1	UCX05D1	4,390	2,540
13/16	UCFLX05-013D1	5 9/16	4 19/32	1/2	1 3/16	15/32	3 9/32	1 19/32	1.500	.626	3/8	FLX05D1	UCX05-013D1		
7/8	UCFLX05-014D1											FLX05D1	UCX05-014D1		
15/16	UCFLX05-015D1											FLX05D1	UCX05-015D1		
1	UCFLX05-100D1											FLX05D1	UCX05-100D1		
30	UCFLX06D1	156	130	15	34	16	95	44.4	42.9	17.5	M14	FLX06D1	UCX06D1	5,780	3,440
1 1/16	UCFLX06-101D1	6 1/8	5 1/8	19/32	1 11/32	5/8	3 3/4	1 3/4	1.689	.689	1/2	FLX06D1	UCX06-101D1		
1 1/8	UCFLX06-102D1											FLX06D1	UCX06-102D1		
1 3/16	UCFLX06-103D1											FLX06D1	UCX06-103D1		
1 1/4	UCFLX06-104D1											FLX06D1	UCX06-104D1		
35	UCFLX07D1	171	144	16	38	16	105	51.2	49.2	19	M14	FLX07D1	UCX07D1	6,550	4,000
* 1 5/16	UCFLX07-105D1	6 3/4	5 21/32	5/8	1 1/2	5/8	4 1/8	2 1/32	1.937	.748	1/2	*FLX07D1	*UCX07-105D1		
1 3/8	UCFLX07-106D1											FLX07D1	UCX07-106D1		
1 7/16	UCFLX07-107D1											FLX07D1	UCX07-107D1		
40	UCFLX08D1	179	148	16	40	16	111	52.2	49.2	19	M14	FLX08D1	UCX08D1	7,310	4,590
1 1/2	UCFLX08-108D1	7 1/16	5 27/32	5/8	1 9/16	5/8	4 3/8	2 1/16	1.937	.748	1/2	FLX08D1	UCX08-108D1		
1 9/16	UCFLX08-109D1											FLX08D1	UCX08-109D1		
45	UCFLX09D1	189	157	16	40	16	116	55.6	51.6	19	M14	FLX09D1	UCX09D1	7,870	5,220
1 5/8	UCFLX09-110D1	7 7/16	6 3/16	5/8	1 9/16	5/8	4 9/16	2 3/16	2.031	.748	1/2	FLX09D1	UCX09-110D1		
1 11/16	UCFLX09-111D1											FLX09D1	UCX09-111D1		
1 3/4	UCFLX09-112D1											FLX09D1	UCX09-112D1		
1 13/16	UCFLX09-113D1											FLX09D1	UCX09-113D1		
50	UCFLX10D1	216	184	18	44	19	133	59.4	55.6	22.2	M16	FLX10D1	UCX10D1	9,780	6,570
1 7/8	UCFLX10-114D1	8 1/2	7 1/4	23/32	1 23/32	3/4	5 1/4	2 11/32	2.189	.874	5/8	FLX10D1	UCX10-114D1		
1 15/16	UCFLX10-115D1											FLX10D1	UCX10-115D1		
2	UCFLX10-200D1											FLX10D1	UCX10-200D1		

* Bearing Selected For Design

Figure B10 Table Illustrating Bearing Selection
NTN Bearing Corporation of Canada Limited

Appendix C - Component Calculations

I) Coverage by Rake Tines

$$Coverage\ Factor = Rake : Forward\ \ Speed\ Ratio * \frac{Number\ of\ Tines\ per\ foot\ of\ belt}{Coverage\ of\ each\ tine\ (feet)}$$

$$= \sqrt{2} * \frac{6\ tines}{1\frac{1}{2}\ feet} * \frac{1}{3}\ feet = 1.88$$

II) Belt Tensions

(Equations and Coefficients taken from Juvinall and Marshek, 1991, p.712)

$$\frac{P_1}{P_2} = e^{f\phi} \qquad P_i = \frac{(P_1+P_2)}{2}$$

P_1 = Tight Side Tension
P_2 = Slack Side Tension
P_i = Initial Tension
f = Coefficient of Friction Between Rubber and Steel
ϕ = Angle of Contact between Belt and Roller

$$\frac{P_1}{P_2} = e^{0.25*(180-35.74)*\frac{\pi}{180}} = 1.87$$

$$P_i = \frac{(2.87P_2)}{2}$$

With **P_i= 75lbs/in (F_i=1650 lbs)**
P_2 = 52.2 lbs/in (F_1=1150 lbs)
P_1 = 97.7 lbs/in (F_2=2150 lbs)

III) Power Transmission

$$hp = \frac{(F_2 - F_1) * V_m}{33000} = \frac{(F_2 - F_1) * V_s}{550}$$

hp = Power in Horsepower
V_m = Belt Speed in Feet per Minute
V_s = Speed in Feet per Second

$$hp = \frac{1000 * V_m}{33000} = \frac{1000 * V_s}{550}$$

IV) Power Required

$$hp = \frac{F * V_m}{33000} = \frac{F * V_s}{550}$$

F = Force on Tines During Raking
(Estimate that 1/2 of tines (60) are under a load at any instant with a load of 1 lb)

$$hp = 60\ tines * 1\frac{lb}{tine} * \frac{V_m}{33000} = 60\ tines * 1\frac{lb}{tine} * \frac{V_s}{550}$$

Belt Velocity (Ft/s)	Forward Velocity (km/hr)	hp available	hp required
4	3.04	7.27	0.44
8	6.07	14.5	0.87
12	9.11	21.82	1.31
16	12.14	29.09	1.75

V) Roller Shaft Revolutions

$$Rev = Life(yrs) * \frac{acres}{year} * \frac{43560\ ft^2/\ acre}{machine\ width*Circumference\ of\ tire} * Gearbox\ Ratio$$

Estimated Machine Life Machine Life =	15 Years
Machines Use =	600 acres
Machine Width =	9.5 ft
Tire Circumference (5.00-15)=	6.8 ft (diameter = 2.167 ft)

$$GearboxRatio = \frac{Tire\ Diameter}{Roller\ Diameter} * \frac{Rake\ Speed}{Forward\ Speed} = \frac{26\ in}{8\ in} * \sqrt{2} = 4.6$$

$$Rev = 15\ yrs * 500\frac{ac}{yr} * \frac{43560\ ft^2/ac}{9.5\ ft * 6.8\ ft} * 4.6 = 23.26 \times 10^6\ rev \approx \mathbf{23.3} \times 10^6\ rev$$

VI) Bearing Load Calculations

(Method described by Juvinall and Marshek, 1991, p. 518-549)

$$C_{req} = F_e K_a (\frac{L}{KL_R})^{0.3}$$

F_i = Tension Force of Belt
C_{req} = Bearing Rated Capacity
F_e = Equivalent Load (equivalent to radial load because of minimal thrust load)
K_a = Application Factor (Taken as 1.5 from Table 14.3, Juvinall and Marshek, 1991)
L = Life Corresponding to Rated Capacity (determined by manufacturer)
K_r = life adjustment reliability factor (90% reliability gives K_r of 1)
L_R = Life Corresponding to Load, F_e

$F_e(max)$ = $2*F_i/2$ bearings
= 1650 lbs

$$C_{req} = 1650\ lbs * 1.5(\frac{23.3{\times}10^6\ rev}{10^6\ rev*1})^{0.3} = 6365\ \text{lbs}$$

VII) Spring Size Calculation for Tightener

(Method described by Juvinall and Marshek, 1991, p. 427-444)

d = wire diameter
D = Spring diameter (Total Diameter-d)
F_{solid} = Force Required to Compress spring To Solid Height
k = Spring Constant
τ_{solid} = Design Shear Stress
K_s = Stress Correction Factor
C = D/d
G = Modulus of Rigidity (11600 ksi)
N = Number of active turns
N_t = Number of total turns
L = Spring Length

Design Parameters

k =1000 lbs/in
Constant force on spring is 1650 lbs
Desire spring to have 1 additional inch of working deflection past constant force
Clash allowance of 10% of maximum working deflection

$$=0.1*\frac{2650\ lbs}{1000\ lbs/in} = 0.265\ in$$

$$F_{solid} = 2650\ lbs + 1000\ lbs/in * .265\ in = 2915\ lbs$$

$$\text{(a) } \tau_{solid} = \frac{8F_{solid}D}{\pi d^3}K_s \qquad \text{(b) } \tau_{solid} = \frac{8*F_{solid}}{\pi d^2}CK_s$$

Using (a):

Assume D = 1.5 in, d=0.5 in for value of τ_{solid}
From Figure 12.4, estimate K_s=1.12
From Figure 12.7, Tensile strength (Su) of 165 ksi. $\tau_{solid} = 0.65*S_u = 103.12$ ksi (preset)

obtain: d = 0.495 in

Now use result in Eq (b) to recalculate D:

τ_{solid} for a wire of 0.495 in diameter is very close to that for a wire of 0.5 in
From (b), CK_s=3.4
Use in figure 12.4 to get C = D/d = 3.0
D = 1.485 in.

$$k = \frac{d^4 G}{8D^3 N} \qquad N_t = N(\text{Ground Ends}) \qquad L = N_t d + \frac{F_{solid}}{k}$$

N_t = 26.58 turns
L = 16.17 in
When under normal load (1650 lbs), Spring length is **14.42** inches

VIII) Nut thread strength calculation for belt tightener

(Method described by Juvinall and Marshek, 1991, p. 357.)

F_{max} = Maximum force that nut can support
S_y = Yield strength (1030 rolled steel used)
d = Major diameter of thread
t = Thickness of nut

d = 5/8", S_y of 1030 rolled steel = 50 ksi, t = 0.55"

$$F_{\max} \approx \pi d * (0.75t) S_y$$

$\boldsymbol{F_{max}}$ **= 40 497 lbs**

$$SF = \frac{40497\ lbs}{2500\ lbs} = 16.2 \text{ (very good)}$$

IX) Rod buckling calculation for belt tightener

(Method described by Juvinall and Marshek, 1991, p. 187-95.)

ρ = Radius of gyration
d_m = Smallest diameter of column (minor diameter of threaded rod)
A = Cross sectional area of bolt
I = Moment of interial with respect to buckling-bending axis
L = Length of column
L_e = Equivalent length of column
E = Modulus of Elasticity
S_{cr} = Maximum column stress
S_y = Yield stress of material
P_{max} = Maximum load on column

d_m = 0.5135(5/8")	A = 0.226in^2
L = 6 in	S_y= 60 ksi
E = 30 000 ksi	L_e= 0.8L (fixed ends)

$$I = \frac{d_m^4}{64} = 798 \times 10^{-6}\ in^4$$

$$\rho = \sqrt{\frac{I}{A}} = \frac{d}{4} = \frac{0.5135\ in}{4} = 0.128\ in$$

$$\text{Tangent Point} = \sqrt{\frac{2\pi^2 E}{S_y}} = 99.34$$

$$\frac{le}{\rho} = \frac{0.8*6\ in}{0.128\ in} = 37.5$$

Solve using Johnson's Equation

$$S_{cr} = S_y - \frac{S_y^2}{4*\pi^2 E}\left(\frac{le}{\rho}\right)^2 = 60 \times 10^3 - \frac{(60\times10^3)^2}{4\pi^2 30\times10^6}(37.5)^2 = 55.72\ ksi$$

$$P_{\max} = S_{cr} * A = 55.72\ ksi * 0.226\ in^2 = 12590\ lbs$$

$$SF = \frac{12590\ lbs}{2500} = 5.03 \text{ (very good)}$$

X) Bevel Gears in Gearbox

(Method described by Juvinall and Marshek, 1991, p. 257-270, 550-627.)

A) *Geometry and Forces*

ω_p = Rotational speed of pinion
ω_g = Rotational speed of gear
N_p = Number of teeth on pinion
N_g = Number of teeth on gear
P = Diametral Pitch
d_p = Pitch diameter of pinion
d_g = Pitch diameter of gear
γ_p = Pitch cone angle of pinion
γ_g = Pitch cone angle of gear
b = Face width
L = Pitch cone length
F = Resultant tooth force
F_t = Tangential tooth force
d_{av} = Average diameter
V_{av} = Average tangential velocity
$\dot{W}$ = Power

$$\text{Gear Ratio} = \frac{\omega_p}{\omega_g} = \frac{N_g}{N_p} = \frac{d_g}{d_p} = \tan\gamma_g = \cot\gamma_p$$

Given:

Gear Ratio = 4.6
d_p = 2 in
N_p = 12 teeth
P = 6

Obtain:

γ_p = 12.3°
γ_g = 77.7°
N_g = 55 teeth
d_g = 9.17 in

For Tooth Width: $b \leq \frac{10}{P}$ $b \leq \frac{L}{3}$

For Pinion. L = 4.7 in
For Gear, L = 4.7 in

b = 1 3/8 in

$$d_{av} = d - b\sin\gamma$$

$$V_{av} = \pi d_{av}\omega$$

$$F_t = \frac{33000\dot{W}}{V_{av}}$$

For pinion: ω_p = 459 rpm (for a belt speed of 16 ft/s)

d_{av} = 1.71 in
V_{av} = 205 ft/min
F_t = 282 lbs

For gear: ω_g=99.78 rpm

d_{av} = 7.83 in
V_{av} = 205 ft/min
F_t = 282 lbs

B) *Gear Tooth Bending Strength*

σ	= Gear bending stress
J	= Geometry factor (Fig. 16.13)
K_v	= Velocity factor
K_o	= Overload factor (Table 15.1)
K_m	= Mounting factor (Table 16.1)
S_n	= Endurance Limit (stress)
S_n'	= Standard R.R. Moore Endurance Limit
C_l	= Load factor
C_g	= Gradient factor
C_s	= Surface factor (Fig. 8.13)
k_r	= Reliability factor
k_t	= Temperature factor
k_{ms}	= Mean stress factor

$$\sigma = \frac{F_t P}{bJ} K_v K_o K_m$$

For Pinion:

J	= 0.24 (Fig 16.13)
K_v	= 1.1 (Fig 15.24)
K_o	= 1.5 (Table 15.1, Light shock power source, medium shock driven machinery)
K_m	= 1.5 (Table 16.1, overhung design)

σ =12.69 ksi

For Gear:

J	= 0.18 (Fig 16.13)
K_v	= 1.1 (Fig 15.24)
K_o	= 1.5 (Table 15.1, Light shock power source, medium shock driven machinery)
K_m	= 1.5 (Table 16.1, overhung design)

σ =16.92 ksi

$$S_n = S_n' C_l C_G C_s k_s k_r k_{ms}$$

S_n'	= 102 ksi (Fig. 8.6, for SAE 5150 Hardness, H_B of 375)
C_l	= 1(bending)
C_g	= 1 (P is bigger than 5)
C_s	= 0.64 (Fig. 8.13, machined)
k_r	= 0.814 (99% reliability)
k_t	= 1 (Temperature is always less than 160 degrees Fahrenheit)
k_{ms}	= 1.4 (two way bending)

S_n = 74.39 ksi

$$SF_{bending} = \frac{S_n}{\sigma_{\max}} = \frac{74.39\ ksi}{16.92\ ksi} = 4.40$$

C) *Gear Tooth Fatigue Strength*

σ_H = Surface fatigue stress
C_p = Elastic coefficient
I = Geometry Factor
ϕ = pressure angle
S_H = Gear tooth surface fatigue strength
S_{fe} = Surface fatigue strength (Table 15.5)
C_{Li} = Life Factor (Fig. 15.27)
C_R = Reliability Factor (Table 15.6)
Bhn = Brinell Hardness number

$$\sigma_H = C_p \sqrt{\frac{F_t}{b * d_p * I} K_v K_o K_m}$$

$$I = \frac{\sin\phi \cos\phi}{2} \frac{R}{R+1} \Rightarrow R = \text{gear ratio}$$

For Pinion:

ϕ =20 degrees
C_p = 2300
I = .132
K_v = 1.1 (Fig 15.24)
K_o = 1.5 (Table 15.1, Light shock power source, medium shock driven machinery)
K_m = 1.5 (Table 16.1, overhung design)

σ_H = 100 ksi

For Gear:

ϕ = 20 degrees
C_p = 2300
I = 0.132
K_v = 1.1 (Fig 15.24)
K_o = 1.5 (Table 15.1, Light shock power source, medium shock driven machinery)
K_m = 1.5 (Table 16.1, overhung design)

σ_H = 47.1 ksi

$$S_H = S_{fe} C_{Li} C_R$$

S_{fe} = 0.4(Bhn)-10ksi = .4(375 Bhn)-10ksi = 140 ksi
C_{Li} = 0.95 (Fig. 15.27)
C_R = 1.00 (Table 15.6 for 99% reliability)

S_H = 133 ksi

$$SF_{fatigue} = \frac{S_H}{\sigma_{H\max}} = \frac{133\ ksi}{100\ ksi} = 1.33$$

XI) Spring Size Calculation for rear raking apparatus suspension

(Method described by Juvinall and Marshek, 1991, p. 427-444)

d = wire diameter
D = Spring diameter (Total Diameter-d)
F_{solid} = Force Required to Compress spring To Solid Height
k = Spring Constant
τ_{solid} = Design Shear Stress
K_s = Stress Correction Factor
C = D/d
G = Modulus of Rigidity (11600 ksi)
N = Number of active turns
N_t = Number of total turns
L = Spring Length

Design Parameters

k =500 lbs/in
Maximum force on spring is 750 lbs (estimate)
Desire spring to have 1.5 inches of deflection between maximum force and free length
Spring Need to Fit in a circular hollow section of 1.25 inches inner diameter and around
a 0.75 inch rod
D+d should be less than 1.25 in by a diametral clearance of .1D

Clash allowance of 10% of maximum working deflection

$$= 0.1 * \frac{750\ lbs}{500\ lbs/in} = 0.15\ in$$

$$F_{solid} = 750\ lbs + 500\ lbs/in * 0.15\ in = 825\ lbs$$

$$(a)\ \tau_{solid} = \frac{8F_{solid}D}{\pi d^3}K_s \qquad (b)\ \tau_{solid} = \frac{8*F_{solid}}{\pi d^2}CK_s$$

Using (a):

Assume D = 1.125 in, d=0.3 in for value of τ_{solid}
From Figure 12.4, estimate K_s=1.13
From Figure 12.7, Tensile strength (Su) of 175 ksi. $\tau_{solid} = 0.45*S_u = 78.75$ ksi (no preset)

d = 0.324 in

Now use result in Eq (b) to recalculate D:

τ_{solid} for a wire of 0.324 in diameter is similar to that for a wire of 0.3 in
From (b), CK_s=3.94
Use in figure 12.4 to get C = D/d = 3.6

D = 1.167 in

Diametral Clearance Required is 0.1*1.167 in =0.117 in
Diametral Clearance Available
= (Inner diameter of pipe) - D+d = 1.62-(0.324+1.167) = 0.129 in

$$k = \frac{d^4G}{8D^3N} \qquad N_t = N(\text{Ground Ends}) \qquad L = N_t d + \frac{F_{solid}}{k}$$

N_t = 20.11 turns
L = 8.17 in

When under normal load (500 lbs), Spring length is **7.17** inches

XII) Strength of parts used to suspend the rear of the rake

(Equations and Coefficients taken from Juvinall and Marshek,1991, p.354-395, 99-143)

Estimate Maximum Force to be 1500 lbs (750 lbs per support)

A) Force on threads on nuts supporting suspension bar

F_{max} = Maximum force that nut can support
S_y = Yield strength (1030 rolled steel used)
d = Major diameter of thread
t = Thickness of nut

given:
d = 3/4",
S_y of 1030 rolled steel = 50 ksi, t = 0.66"

$$F_{\max} \approx \pi d * (0.75t)S_y$$

F_{max} **= 58 316 lbs**

$$SF_{threads} = \frac{58316\ lbs}{750\ lbs} = 77.75$$

B) Tension on Suspension bar

σ = tensile stress on rod
F = Force on bar (lbs)
A = Cross-sectional Area of Bar
S_y = 60 ksi, 1040 rolled steel

$$\sigma = \frac{F}{A} = \frac{750\ lbs}{\frac{\pi(0.75\ in)^2}{4}} = 1.70\ ksi$$

$$SF = \frac{60\ ksi}{1.7\ ksi} = 35.3$$

C) Pins connecting suspension bars to raking mechanism

τ_{max} = Shear stress pin can support (1040 steel used)
τ = Shear stress caused by load
S_y =Yield Strength of metal
F = Force on bar (lbs)
A = Cross-sectional Area of Bar

Given:

S_y =60 ksi

$$\tau_{\max} = 0.58S_y = .58 * 60\ ksi = 34.8\ ksi$$

$$\tau = \frac{F}{A} = \frac{750\ lbs}{\frac{2*\pi*(0.5\ in)^2}{4}} = 1.91\ ksi$$

$$SF = \frac{34.8\ ksi}{1.91\ ksi} = 18.2$$

D) Pins connecting lift arms to suspension bar

τ_{max} = Shear stress pin can support (1040 steel used)
τ = Shear stress caused by load
S_y =Yield Strength of metal
F = Force on bar (lbs)
A = Cross-sectional Area of Bar

$$\tau_{max} = 0.58S_y = .58 * 60\ ksi = 34.8\ ksi$$

$$\tau = \frac{F}{A} = \frac{750\ lbs}{\frac{2*\pi*(0.375\ in)^2}{4}} = 3.4\ ksi$$

$$SF = \frac{34.8\ ksi}{3.4\ ksi} = 10.24$$

E) Force on Lift Arms

σ_e	= Bending stress
σ	= Equivalent stress
M	= Maximum moment
c	= distance from centroidal axis to point of maximum stress
F	= Force applied on beam
d	= Distance to force
b	= Width of beam
h	= Height of beam

Given:

M	= F*d = 750 lbs*6 in = 4500 in-lbs
c	=1 in

$$\sigma_e = \sigma = \frac{M*c}{I}$$

$$I = \frac{bh^3}{12} = \frac{(0.25\ in)*(2\ in)^3}{12} = 0.166\ in^4$$

$$\sigma_e = \frac{4500\ in\text{-}lbs*1\ in}{0.166\ in^4} = 27.1\ ksi$$

$$SF_{lift\ arms} = \frac{48\ ksi}{27.1\ ksi} = 1.77$$

F) Force on Control Bar

σ	= Bending stress
σ_e	= Equivalent stress
M	= Maximum moment
c	= distance from centroidal axis to point of maximum stress
F	= Force applied on beam
d	= Distance to force
b	= Width of beam
h	= Height of beam

Given:

M = F*d = 1500 lbs*5 in = 7500 in-lbs
c = 1 in

$$\sigma_e = \sigma = \frac{M*c}{I}$$

$$I = \frac{bh^3}{12} = \frac{0.5\ in*(2\ in)^3}{12} = 0.333\ in$$

$$\sigma_e = \frac{7500\ in\text{–}lbs*1\ in}{0.333\ in^4} = 22.52\ ksi$$

$$SF_{control\ arm} = \frac{48\ ksi}{22.52\ ksi} = 2.13$$

G) Force on crank bar threads

F_{max} = Maximum force that nut can support
S_y = Yield strength (1020 rolled steel used)
d = Major diameter of thread
t = Thickness of nut

given:

d = 5/8"
S_y of 1030 rolled steel = 50 ksi,
t = 0.55

$$F_{\max} \approx \pi d * (0.75t)S_y$$

F = 40 497 lbs

$$FS_{crank\ threads} = \frac{40497\ lbs}{1500\ lbs} = 27.0$$

H) Stresses on control bar (see diagram)

τ = Shear stress caused by load
σ_{max} = Maximum bending stress
σ_e = Equivalent stress
M = Maximum moment
T = Torque on beam
c(=r) = distance from centroidal axis to point of maximum stress
I = Moment of Inertia
J = Torsional Constant
S_y =Yield Strength of metal (48 ksi for 1020 as rolled)

Given:

M = 15 800 in-lbs
T = 4500 in-lbs
I = .867 in^4
J = 1.737 in^4
c (=r) = (diameter/2) = (2.375/2) in

$$\tau = \frac{Tr}{J} = \frac{750\ lbs*6\ in}{1.737\ in^4} * \frac{2.375\ in}{2} = 3.08\ ksi$$

$$\sigma_{\max} = \frac{Mc}{I} = \frac{15.8\times10^3\ in\text{-}lbs*(2.375/2)\ in}{0.867\ in^4} = 21.64\ ksi$$

$$\sigma_e = \sqrt{\sigma_{\max}^2 + 3\tau^2} = \sqrt{(21.64\ ksi)^2 + (3 * 3.08\ ksi)^2} = 23.53\ ksi$$

$$SF = \frac{S_y}{\sigma_e} = \frac{48\ ksi}{23.53\ ksi} = 2.04$$

I) Rod buckling calculation for belt tightener

(Method described by Juvinall and Marshek, 1991, p. 187-95.)

ρ = Radius of gyration
d_m = Smallest diameter of column (minor diameter of bolt)
A = Cross sectional area of bolt
I = Moment of interial with respect to buckling-bending axis
L = Length of column
L_e = Equivalent length of column
E = Modulus of Elasticity
S_{cr} = Maximum column stress
S_y = Yield stress of material
P_{max} = Maximum load on column

I = .048in^4
A=0.399in^2
L=54 in
S_y=48 ksi
E =30 000 ksi
L_e=0.8L (fixed ends)

$$\rho = \sqrt{\frac{I}{A}} = \sqrt{\frac{0.048\ in^4}{.399\ in^2}} = .347 in$$

$$\text{Tangent Point} = \sqrt{\frac{2\pi^2 E}{S_y}} = 111.07$$

$$\frac{le}{\rho} = \frac{0.8*54\ in}{0.347\ in} = 124.5$$

Solve using Euler's Equation

$$S_{cr} = \frac{\pi^2 E}{(\frac{Le}{\rho})^2} = 19.1\ ksi$$

$$P_{\max} = S_{cr} * A = 19.1\ ksi * 0.399\ in^2 = 7622\ lbs$$

$$SF = \frac{7622\ lbs}{1500} = 5.08$$

Appendix D

Sensitivity Analysis

Yearly Production (tonnes)	Discount Rate (R)	Present Worth Factor	Present Worth of Benefits in Dollars (machine life=15 years)						
			Increase in Harvesting Efficiency (decimal)						
			0.00	***0.01***	***0.02***	***0.03***	***0.04***	***0.05***	***0.06***
180	0.1	7.61	**-3761**	**-2734**	**-1707**	**-680**	**347**	**1373**	**2400**
270	0.1	7.61	**-3761**	**-2220**	**-680**	**860**	**2400**	**3941**	**5481**
360	0.1	7.61	**-3761**	**-1707**	**347**	**2400**	**4454**	**6508**	**8561**
450	0.1	7.61	**-3761**	**-1194**	**1373**	**3941**	**6508**	**9075**	**11642**
540	0.1	7.61	**-3761**	**-680**	**2400**	**5481**	**8561**	**11642**	**14722**
630	0.1	7.61	**-3761**	**-167**	**3427**	**7021**	**10615**	**14209**	**17803**
720	0.1	7.61	**-3761**	**347**	**4454**	**8561**	**12669**	**16776**	**20883**
810	0.1	7.61	**-3761**	**860**	**5481**	**10101**	**14722**	**19343**	**23964**
900	0.1	7.61	**-3761**	**1373**	**6508**	**11642**	**16776**	**21910**	**27044**

Table D1 Sensitivity Analysis for Present Worth of Benefits (machine life = 15 years)

Yearly Production (tonnes)	Discount Rate (R)	Present Worth Factor	Present Worth of Benefits in Dollars (machine life=12 years)						
			Increase in Harvesting Efficiency (decimal)						
			0	***0.01***	***0.02***	***0.03***	***0.04***	***0.05***	***0.06***
180	0.1	6.81	**-3681**	**-2762**	**-1842**	**-922**	**-2**	**918**	**1838**
270	0.1	6.81	**-3681**	**-2302**	**-922**	**458**	**1838**	**3217**	**4597**
360	0.1	6.81	**-3681**	**-1842**	**-2**	**1838**	**3677**	**5517**	**7357**
450	0.1	6.81	**-3681**	**-1382**	**918**	**3217**	**5517**	**7817**	**10116**
540	0.1	6.81	**-3681**	**-922**	**1838**	**4597**	**7357**	**10116**	**12876**
630	0.1	6.81	**-3681**	**-462**	**2758**	**5977**	**9197**	**12416**	**15635**
720	0.1	6.81	**-3681**	**-2**	**3677**	**7357**	**11036**	**14716**	**18395**
810	0.1	6.81	**-3681**	**458**	**4597**	**8737**	**12876**	**17015**	**21155**
900	0.1	6.81	**-3681**	**918**	**5517**	**10116**	**14716**	**19315**	**23914**

Table D2 Sensitivity Analysis for Present Worth of Benefits (machine life = 12 years)

Note 1: Formula for Series Present Worth Factor(SPWF) calculated as follows:

$SPWF=((1+R)^n-1)/(R(1+R)^n)$ *(From Midwest Plan Service)*

where

R=interest rate

n=period of time

Note 2: a) The price of the crop is assumed to be $75 per tonne dry matter

b) Maintenance costs are estimated to be $100 per year more for new machine, compared to a conventional machine

c) Purchase price is assumed to be $3000 more for a new machine, compared to a conventional machine