DESIGN OF A ROUND HAY BALE ACCUMULATOR

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Ste-Anne-de-Bellevue, December 9th, 1993

ABSTRACT

An infinite number of design possibilities are constantly lurking in the back of our mind. It is left to each individual to grasp it and develop it.

This project is about a system which will help the farmer to increase its overall field efficiency with respect to haymaking. It will enable him to bale, pick and stack large round bales in one operation. Its stacking capacity will vary from three to five bales, depending on their size. This quantity should be enough to carry the bales at the end of the field to be dropped off at the same place. Its adapting capabilities to other balers and its manoeuvrability in the field should make it a good piece of machinery.

TABLE OF CONTENTS

LIST OF FIGURES 4
LIST OF TABLES 4
1. INTRODUCTION 5
2. OBJECTIVE 6
3. MATERIALS AND METHODS 7
- 3.1 Friction factor estimation 7
- 3.2 Determination of dumping angle 7
- 3.3 Flexural stresses 8
- 3.4 Cylinder required forces 8
- 3.5 Rolling resistance 9
- 3.6 Ground pressure 9
- 3.7 Use of computer aided drawing program in the design 10

4. RESULTS AND DISCUSSION 10
- 4.1 Description of alternatives 10
- 4.2 Working of the dumping trailer 12
- 4.3 Friction factor estimation 14
- 4.4 Determination of dumping angle 14
- 4.5 Flexural stresses 15
- 4.6 Cylinder required force 18
- 4.7 Turning angle 19
- 4.9 Rolling resistance 22
- 4.10 Ground pressure 22
- 4.11 Cost of the machine 23
- 4.12 General discussion 23

5. CONCLUSION 25
REFERENCES 26
APPENDICES 27

LIST OF FIGURES

Drawing 1.1: Conventional round baler	p.5
Drawing 4.1: The elevated baler	p.10
Drawing 4.2: The pinned hydraulic mid-section	p.11
Drawing 4.3: Hooked hydraulic mid-section	p.11
Drawing 4.4: Dumping trailer	p.12
Drawing 4.5: View of the carrier out of the trailer	p.13
Drawing 4.6: View of the trailer in dumped position	p.13
Drawing 4.7: Determination of the optimal dumping angle	p.14
Drawing 4.8: Dimensions of the dumping trailer	p.17
Drawing 4.9: View of the carrier with associated hydraulic cylinders	p.19
Drawing 4.10: Maximum turning angle when carrier at rest position	p.19
Drawing 4.11: Maximum turning angle when carrier is deployed	p.20
Drawing 4.12: Hydraulic system	p.20
Drawing 5.1: Overall view of the dumping trailer	p.25

LIST OF TABLES

Table 4.1: Weighing of alternatives	p.12
Table 4.2: Experimental results from pulling tests	p.14
Table 4.3: Some important flexural stresses	p.15
Table 4.4: Required cylinder capacity	p.18
Table 4.5: Rolling resistance for different sinkage values	p.22
Table 4.6: Material cost approximation	p.23

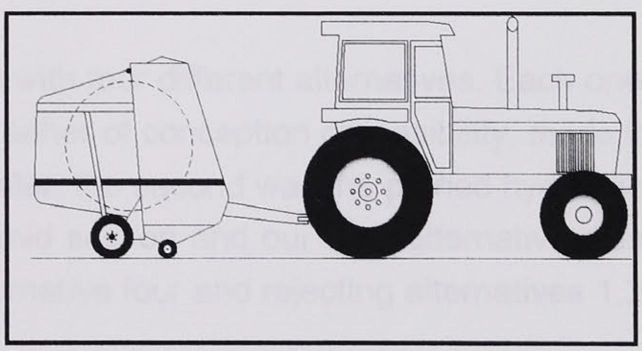
1. INTRODUCTION

Agriculture has always played a major role in the development and prosperity of a country. Therefore, it was important to keep up with the technology which increased farms efficiencies on many points. The need to decrease manual work on farms in the last century, due to economical reasons, was one of the main reasons for this incredible burst of development in machinery.

Hay has always been the most important forage on a farm. It was also one of the most demanding activity concerning hand labour. It is simple then to understand the need for a better hay handling system. Many different types of balers came onto the market since the turn of the century but none that decreased significantly the labour demand.

When the conventional square baler came out, with its high density bale, it revolutionised the haymaking process. As the years passed the machines got faster, more reliable and efficient. But square bales still required a certain amount of hand labour to handle all these small bales. Developments in handling methods inevitably lead to systems that do not require manual work. Such systems are available for square bales but the point to remember here is the number of bales involved.

The round baler was a good alternative to this problem. It produced a bale equivalent to approximately 15 conventional square bales. These bales were too heavy for man-handling. Therefore, many systems came on the market to handle them. All these systems require the farmer to go through the whole field a second time just to pick up the bales that are scattered all over the field. The problems encountered with this system are that it causes extra compaction, time loss and a few other points.



Drawing 1.1: Conventional round baler

This aspect about round bales was not discussed in any of the books used in our reading. They do not mention that handling large round bales is a problem. This is were we came up with the idea of having some kind of mechanism which could pick and stack round bales as they come out of the baler, without even touching the ground. There are some existing systems, but they are limited to either the bale size and/or the number of bales they can contain.

These where the points that had to be developed, while bearing in mind that it had to be very simple and as versatile as possible.

2. OBJECTIVE

The objective of this design is to come up with a mechanism that could pick and stack round bales as they come out of the baler. his mechanism has to be simple and reliable enough to make it worth the trouble.

We had to overcome a few design problems. Such as low ground clearance, limited working space, heavy loads, different field conditions, different balers to adapt to and many more. Since we did not have the chance to build a prototype, we were only able to establish its feasibility with our own practical knowledge and with the assistance of our drawings, which are to scale, to see any anomalies in the concept.

Considering all these factors, we came up with four different alternatives. Each one had a certain potential but some major problems, either of conception or feasibility, made it oblivious. The first alternative was the elevated baler, the second was the pinned hydraulic mid section, the third was the hooked hydraulic mid section and our final alternative was the dumping trailer. The reasons for choosing alternative four and rejecting alternatives 1,2 and 3 will be discussed later on in this report.

Now that we know what the design problems and criteria are, we can elaborate on then now.

3. *MATERIALS AND METHODS*

3.1 Friction factor estimation

Prior to the design of the actual picker and stacker, some tests were performed on the field. The purpose of those tests was the determination of the friction factors involved when pushing/pulling a typical round bale on different surfaces. The tests were performed on a beef farm where round bales are used for haylage. The particular bale used was a silage bale of 5 feet width and 4 feet diameter. It was not covered by a plastic for it was a surplus and wasn't probably going to be fed to animals.

The method used was of a great simplicity. The bale was pulled by a tractor on different surfaces representative of the conditions in alternatives 1 to 4. The device used to estimate the force involved was a simple cylinder tied at one end to the bale and at the other end to the tractor. A gage was measuring the pressure in the cylinder. Knowing the effective area of the cylinder, the pulling force can be determined easily. The particular cylinder used had an effective area of 1.46 square inches. Once the pulling force has been determined, a friction factor can be calculated using a simple free-body-diagram. The weight of the bale was determined by lifting it with the cylinder tied in between the tractor and the bale.

Five tests were performed:

1. On a wood plank wagon;
2. On the ground (to find a max. friction factor);
3. On a rough steel wagon;
4. On a smooth steel surface;
5. Lifting of the bale.

3.2 Determination of dumping angle

Only tests 4 and 5 were required for the design of the chosen alternative. Test 4 allowed the determination of a friction factor for a smooth steel surface. Test 5 measured the actual weight of the bale. Knowing these two components, a free-body-diagram of the bale on a slope permitted the determination of a critical dumping angle at which the weight of the bale overcomes the friction. The critical angle found in this manner was 13.5 degrees and an angle of 15 degrees was therefore used in the design, allowing for 1.5 degree of safety. Calculations are shown in appendices.

3.3 Flexural stresses

In the design of the chosen alternative, only the most important flexural stress analyses were conducted. The main equation used was :

$$\sigma \max = \frac{MC}{I}$$

For example, the maximum flexural stress was calculated in the cantilever part of the dumping frame (down position) for different possible loads. In a similar fashion, the maximum flexural stress was calculated for the fixed frame under different loads. Another calculation involved the strength of the carrier's forks. Unfortunately, because of the size of the prototype and the number of parts included, it was not possible to actually calculate the exact shape and type of material used for all of them. However, some most important calculations are included in appendices.

3.4 Cylinder required forces

Some hydraulic cylinders were incorporated in the design of the machine itself. Four of them are required for that particular design. The two small cylinders that are incorporated in the carrier are used to rotate the forks and make it possible to reach the point where the bale falls. When contracting the forks, a large moment is created around their axis of rotation. The cylinders are hinged 2 inches from the center of axis and the bales create a moment at around 20 inches off the center of the axis. Therefore, the combined force of the cylinders must be greater or equal to ten times the reaction force of the bale. These two cylinders are called rotational cylinders.

The operation of the carrier also requires another cylinder to make it roll along its tracks (in the c-beams). This is called the translational cylinder. Its required force is simply calculated by multiplying the maximum possible weight of bales by the friction factor. In this factor. In this case, a friction factor twice as large as the experimental one was used in order to account for possible bad sliding conditions.

Finally, the dumping operation requires another cylinder. It has to be able to lift a whole load of bales and the dumping platform itself. It's required force was calculated by analysing the force required to lift the dumping frame at the very beginning of the dumping process. This is the point where the angle between the dumping frame and the cylinder is the smallest, requiring the largest force from the cylinder. As the dumping frame is moving up, the line of action of the cylinder is aligning with the rotational path of the reaction force, requiring less and less force from the cylinder. All the calculations concerning the cylinder forces are demonstrated in the appendices.

3.5 Rolling resistance

An important issue in this project is the amount of additional power required from the tractor to operate the system. The additional pulling force due to rolling resistance has been calculated.

Knowing the formula of the ratio of rolling resistance:

$$\frac{R}{N} = \frac{\Delta t + z}{d}$$

where:

Δt = r-rl
z = sinkage
d = diameter of a wheel
R = rolling resistance
N = normal force (total weight)
r = unloaded radius
rl = loaded radius

R can easily be determined for different sinkage.

3.6 Ground pressure

Today's farmers are getting more and more concerned with the damaging action of heavy machinery on their soil. For that reason, the pressure on the ground has been estimated for a maximum load situation. In order to achieve this, the total weight must be known or estimated. This number is then divided by the total contact area of the tires with the ground.

3.7 Use of computer aided drawing program in the design

The computer aided drawing program Autocad was used in each steps of the design. From hand sketches, each part of the machine was drawn to scale in 3 dimensions on Autocad. By drawing them to scale in 3 dimensions, the fitting of each part could be checked and movement of mechanisms could be simulated to verify if they are entering in contact with other parts of the machine. Drawing this machine in 3 dimensions was almost like building a prototype of it. The use of this design procedure facilitates also the printing of drawings, since many views of the machine can be printed just by turning the 3 dimensional drawing. Finally, the presentation and demonstration of the mechanisms and details of the machine are easier with this approach.

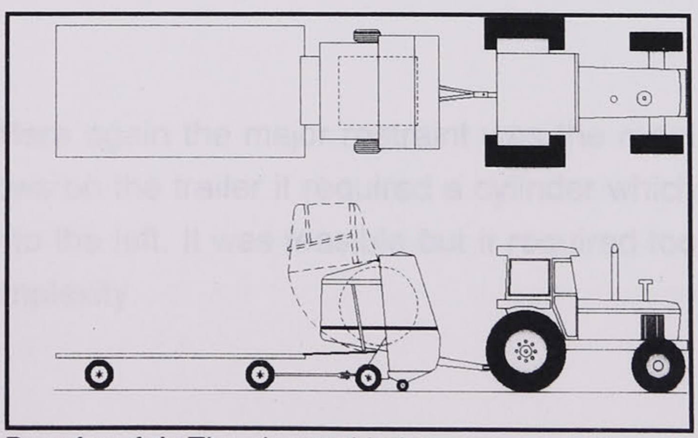
4. RESULTS AND DISCUSSION

4.1 Description of alternatives

We came up with four different alternatives in our analysis. All four alternatives had a certain potential feasibility. As mentioned in the introduction, the alternatives are:

1. The elevated baler
2. The pined hydraulic mid section
3. The hooked hydraulic mid section
4. The dumping trailer

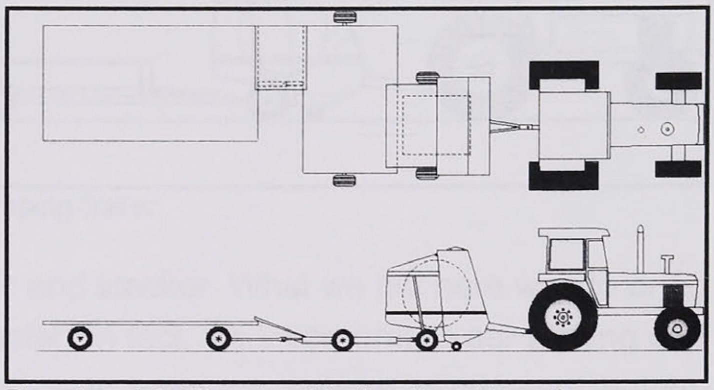
The elevated baler (see drawing 4.1) was our first idea. The concept was good but it required a complete change in the balers design and construction. This would only be feasible if the manufacturers themselves made the change. If you think about it, it is easier to bring up loose hay an



Drawing 4.1: The elevated baler

extra three feet than to bring up a whole bale which weights 1000 pounds.

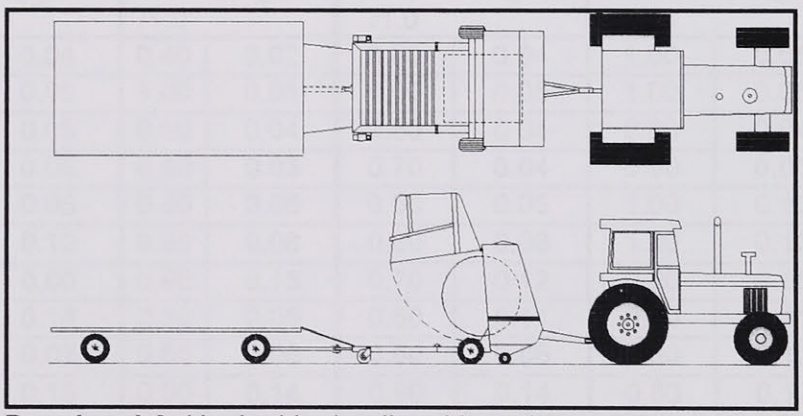
The pined hydraulic mid-section (see drawing 4.2) was the second alternative. This design consisted of a mechanism which would pick the bale has it came out of the baler, rotate it 90 degrees and carry up onto the trailer. Here again the idea was good, since it was possible to use a conventional trailer to stack the bales. This meant a great decrease in costs.



Drawing 4.2: The pined hydraulic mid section

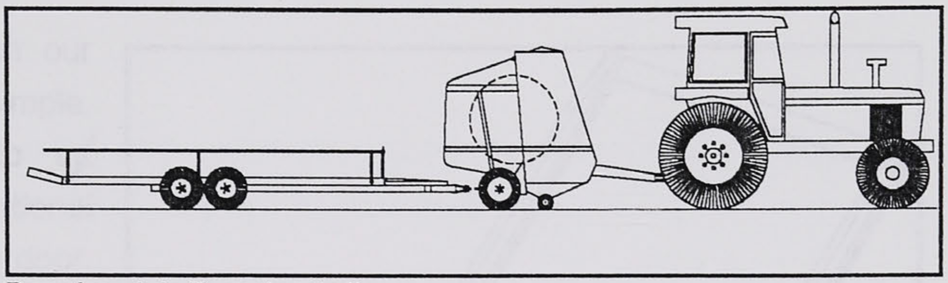
The problem here was that we ended up with three hitch points from the tractor to the trailer. This would hinder a proper manoeuvrability in the field while balling. Another major problem here was the limited space under the mid-section. It was almost impossible to get a working system in such a restrained area.

The third alternative was the hooked hydraulic mid-section (see drawing 4.3). The concept in this case is the same as for alternative two. The difference is that now it is welded onto the bailer. Therefore, it will increase its manoeuvrability in the field since we only have two hitch points instead of three. The other difference is in the picking and stacking mechanism itself. In this design, we did not rotate the bale 90 degrees. It was simply taken up the way it came out and stacked in two individual rows on a conventional trailer.



Drawing 4.3: Hooked hydraulic mid-section

This is were the problems appeared. Here again the major restraint was the room availability. To be able to stack two different rows on the trailer it required a cylinder which was able to push the bale either to the right or to the left. It was feasible but it required too much designing and calculations due to the complexity.



Drawing 4.4: Dumping trailer

Finally, we came up with our fourth alternative which is the dumping trailer (see drawing 4.4). This is by far the simplest system. It is a mixture of what we had in mind for alternatives two and three and the concept of a conventional picker and stacker. What we did here was to bring down the top of the trailer to the level of the baler. In fact, we incorporated our picking and stacking device into the trailer. This enabled us to keep the bale at a constant height, which reduced the magnitude of the forces implied in stacking the bales on the trailer. The table 4.1 illustrates the weighing of each alternatives. Based on this the fourth alternative was chosen, which is the dumping trailer.

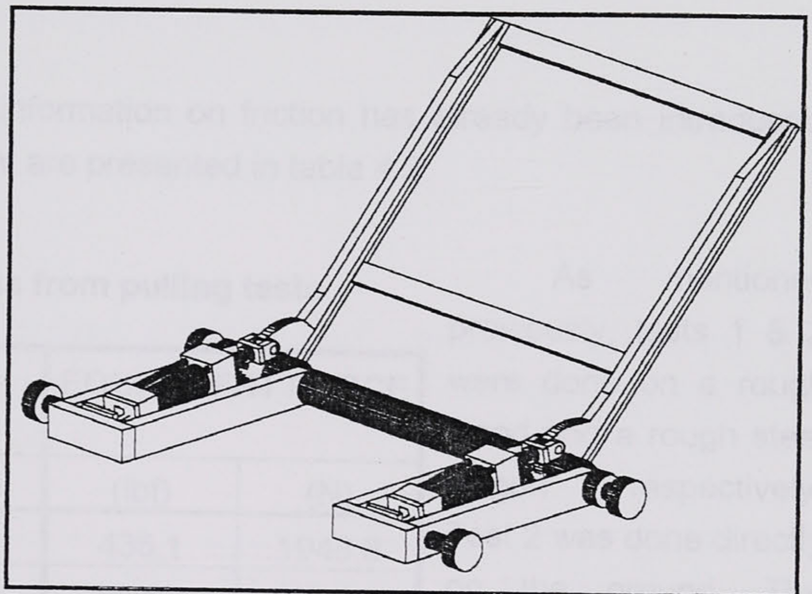
Table 4.1: Weighing of alternatives

Criteria	Weight	Alternative 1		Alternative 2		Alternative 3		Alternative 4	
		score /1.0	weighted	score /1.0	weighted	score /1.0	weighted	score /1.0	weighted
easiness of operation	0.05	0.80	0.04	0.40	0.02	0.80	0.04	1.00	0.05
easiness of maintenance	0.05	1.00	0.05	1.00	0.05	1.00	0.05	1.00	0.05
reliability	0.05	1.00	0.05	0.80	0.04	0.80	0.04	0.80	0.04
simplicity of design	0.05	1.00	0.05	0.60	0.03	0.70	0.04	0.90	0.05
low compaction	0.10	0.50	0.05	0.50	0.05	0.50	0.05	1.00	0.10
adaptability to actual bales	0.10	1.00	0.10	0.80	0.08	0.80	0.08	1.00	0.10
adaptability to actual balers	0.17	0.00	0.00	0.90	0.15	0.70	0.12	0.90	0.15
number of hitching points	0.18	1.00	0.18	0.30	0.05	0.60	0.11	0.60	0.11
required pull from tractor	0.10	0.70	0.07	0.60	0.06	0.60	0.06	1.00	0.10
cost	0.15	1.00	0.15	0.90	0.14	0.90	0.14	0.80	0.12
sum	1.00		0.74		0.67		0.72		0.87

4.2 Working of the dumping trailer

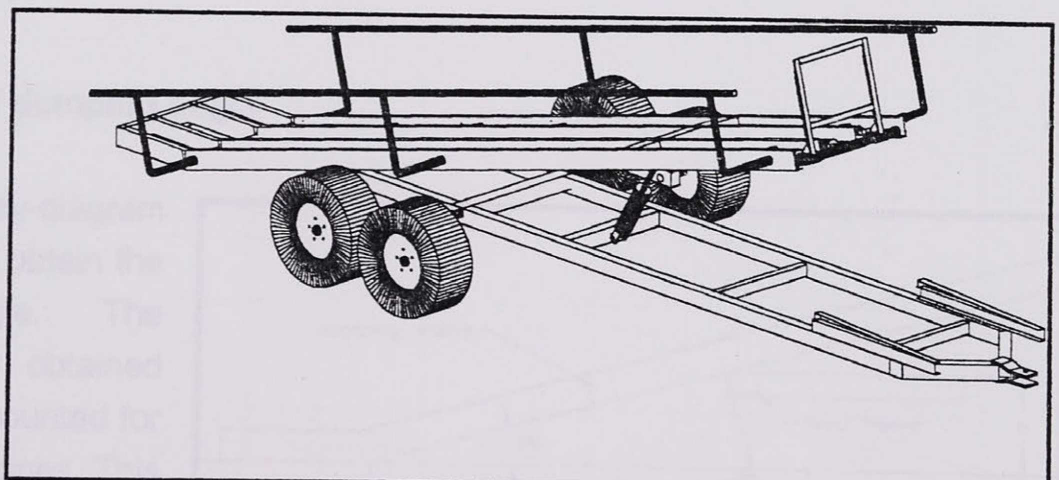
Now that we know what our choice is, we will be able to explain how it works. As we can see on drawing 4.5, the picking and stacking mechanism, which is called the carrier, is situated in front of the trailer. The carrier is activated by two cylinders. The rotational cylinders which deploys and retract the fork and the translational cylinder which pushes the carrier forward and pulls it back to its rest position.

The sequence in which our mechanism will operate is simple. First, the fork is brought to an horizontal position by the rotational cylinders, to clear the unloading door. Then the carrier is pushed forward until it is completely under the unloading door. When this is done, the unloading door of the baler opens and releases the bale which falls onto the fork. Now the rotational cylinders will bring the fork back to its vertical position. Then the translational cylinder pulls the carrier back and in the process pulls the bale with it. When the carrier is completely back the bale is now stacked in front of the trailer. When the second bale is being pulled back, it will push the first one further towards the rear of the trailer. Once the carrier has reached its rest position, the unloading door is closed.



Drawing 4.5: View of the carrier out of the trailer

When the end of the field has been reached it is then possible to unload the trailer by dumping it. The tail gate of the trailer has exactly a 15 degree angle to it (see drawing 4.6). The purpose of the tail gate is to prevent the round bales to fall out. The reason for a 15 degree tail gate is that the maximum dumping angle is 15 degrees. This will help the bales to roll out of the trailer.



Drawing 4.6: View of the trailer in dumped position

4.3 Friction factor estimation

The method used for getting information on friction has already been introduced. The results obtained in the experiment are presented in table 4.2.

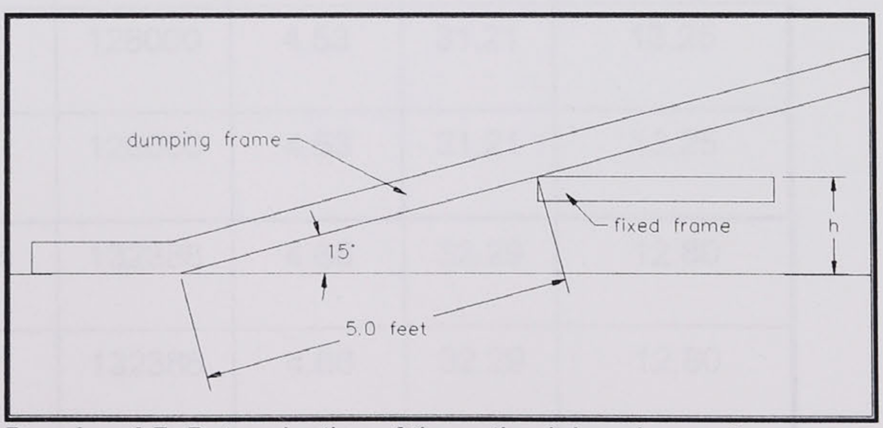
Table 4.2: Experimental results from pulling tests

DESCRIPTION	PRESSURE ON GAGE		EQUIVALENT FORCE	
	(psi)	(MPa)	(lbf)	(N)
TEST 1	300.0	2.07	438.1	1948.8
TEST 2	450.0	3.10	657.2	2923.2
TEST 3	400.0	2.76	584.1	2598.4
TEST 4	150.0	1.03	219.1	974.4
BALE LIFTING	625.0	4.31	912.7	4060.0

As mentioned previously, tests 1 & 3 were done on a rough wood and a rough steel wagon respectively. Test 2 was done directly on the ground. The results of tests 4 & 5 were the ones used in the calculations. Test 4 was typical of the friction factor found on the designed machine. Test 5 (bale lifting) was important since the weight of the bale has to be known for the making of a free-body-diagram. The experimental weight of the bale used is quite standard for a silage bale of that size (5' width and 4' dia.).

4.4 Determination of dumping angle

Such a free-body-diagram was drawn in order to obtain the critical dumping angle. The value of 15 degrees obtained that way was then accounted for in the design of the frames. This angle set the length of the part of the dumping frame that is beyond the fixed frame (see drawing 4.7). Knowing the distance from the ground to the bottom of the dumping frame and the dumping angle, a simple trigonometric calculation would tell the length of the



Drawing 4.7: Determination of the optimal dumping angle

hypotenuse. The hypotenuse corresponds to the part of the dumping frame that is beyond the fixed frame. The length of the hypotenuse came out to be 5 feet (ignoring the inclined part of the dumping frame).

4.5 Flexural stresses

Some of the most important flexural stresses were calculated, principally the stresses in the two major frames of the machine under different loading conditions. The results of these calculations are presented in table 4.3.

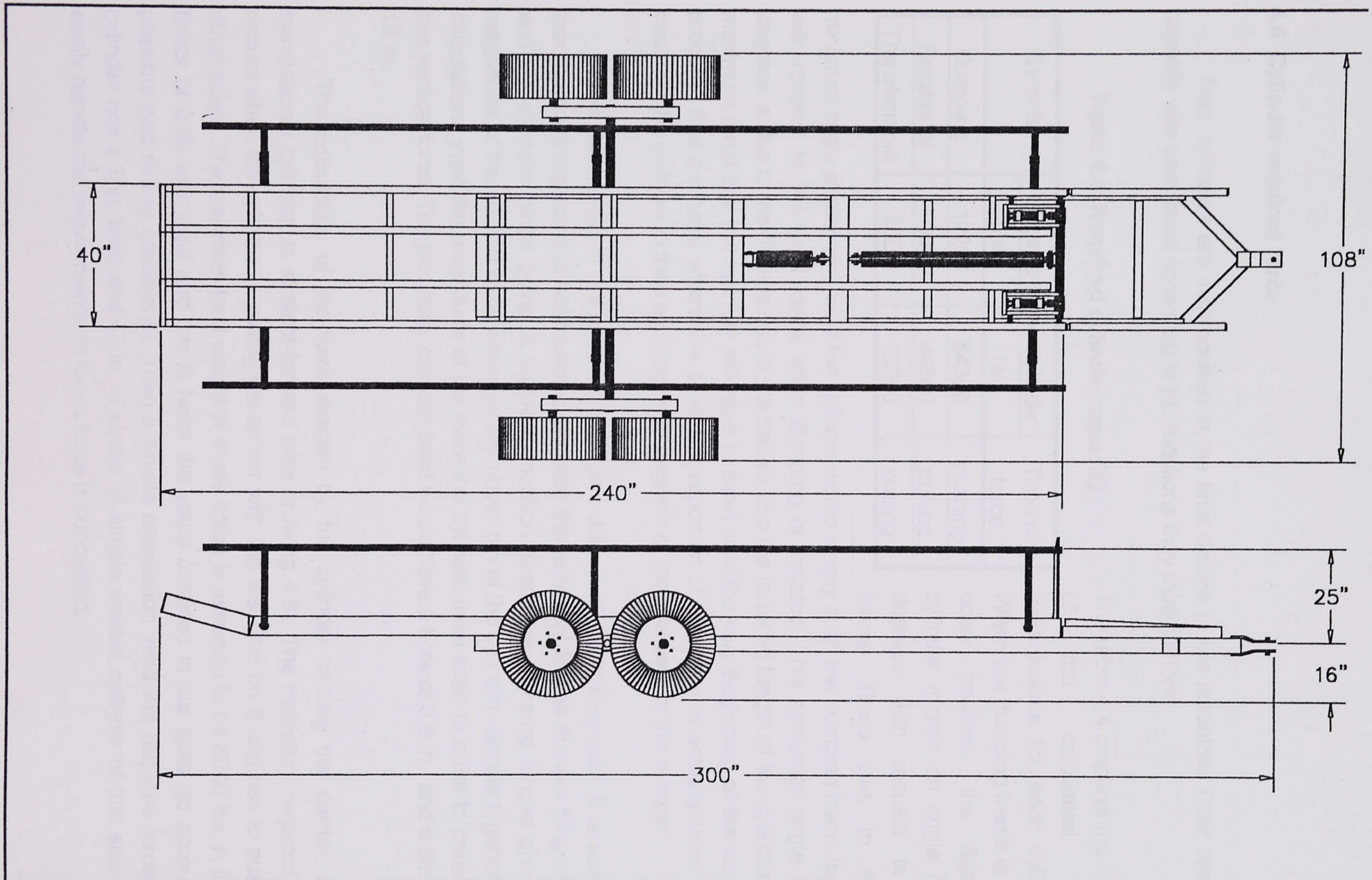
Table 4.3: Some important flexural stresses

Description	Load condition	Weight/bale (lbs)	Max. moment (in. - lbs)	Max. stress per beam (ksi)	Max. stress per beam (Mpa)	Safety factor with 20% C steel
Cantilever part of dumping frame	5 x 4' dia. bales	1000	72000	6.01	41.42	10.00
	4 x 5' dia. bales	1500	81000	6.76	46.61	8.90
	3 x 6' dia. bales	2000	96000	8.01	55.23	7.50
Fixed frame full load with dumping frame down	5 x 4' dia. bales	1000	106629	3.77	26.01	15.90
	4 x 5' dia. bales	1500	128000	4.53	31.21	13.25
	3 x 6' dia. bales	2000	128000	4.53	31.21	13.25
" " " " " dumping mode	4 x 5' dia. bales	2500	132386	4.68	32.29	12.80
	3 x 6' dia. bales	3000	132386	4.68	32.29	12.80

As seen in table 4.3, the chosen beams provide for good safety factors in all cases. However, since the machine will go on the field and be subjected to all kinds of conditions, it may be required to resist to some stresses higher than those accounted for in the design. At this point, the choice of the principal beams should be commented. The reader can refer to drawing 4.8 to see dimensions and details of the dumping trailer.

Concerning the dumping frame, C5 x 6.7 metal beams were selected for the following reasons: The main body of the carrier must move on a track above the fixed frame. It couldn't be at the same level as the fixed frame because it would have to go through it when moving to its picking position. On the other hand, the bales, which are sliding on top of the dumping frame, must not touch the top of the carrier, especially when it is rotating. Since the overall thickness of the body of the carrier is 4.5 in. the C-beams must be thicker and this explains the choice of C5 x 6.7 beams. They were the lightest 5 in. C-beams available. It was decided to use four of them in the dumping frame in order to provide good attachment for the translational cylinder of the carrier.

The choice of rectangular 4 in. x 3 in. of thickness .25 in. for the fixed frame was done for the following reasons: It is a type of beam widely used in dumping trailer frames. It provides the possibility of welding an axle that goes through it at two different places (the two sides of the beam). It also provides a greater moment of area (I) than a C-beam for a given thickness. Only two of those beams are required in the fixed frame.



Drawing 4.8: Dimensions of the dumping trailer

4.6 Cylinder required force

Four cylinders are incorporated in the final design of the machine. Their required capacity was calculated according to the functions they must perform.

Table 4.4: Required cylinder capacity

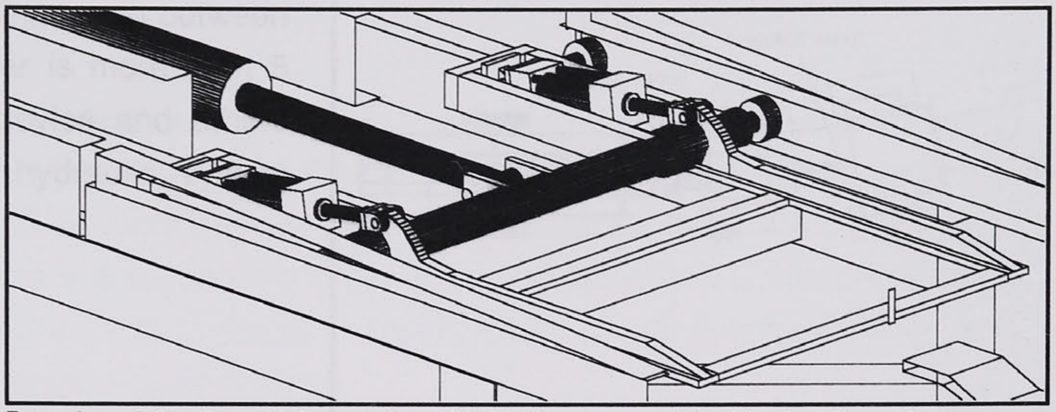
Cylinder	Required force/ cylinder		Type of force
	(lbf)	(N)	
Dumping	12200	54300	pushing
Rotational	10000	44500	pulling
Translational	3000	13500	pulling

The table 4.4 displays the results of the calculated force requirements for each cylinder. When the dumping frame is in its down position, the dumping cylinder makes an angle of 11 degrees with respect to the frame. There has to be a minimum angle at rest otherwise the cylinder would simply push the dumping frame forward with respect to the fixed frame when dumping is initiated. This particular angle of 11 degrees is due to the dimensions of the frames and the required length of the cylinder. The maximum force that the cylinder will have to exert is at the very beginning of the dumping process. It is the point where the smallest proportion of the force is actually used to lift because the cylinder is then pushing at 79 degrees of the real path of the dumper.

The calculation of the required force was done in the following way: It is assumed that the dumping frame is not touching the fixed frame except for the dumper hinge points and the cylinder hinge point. A moment equation about the dumping hinges gives the magnitude of the vertical force acting on the upper pin of the cylinder. Simple trigonometric calculations yield the magnitude of the force the cylinder must exert in order to counteract this vertical force. The particular cylinder used would have a bore of 3.5 in. and a stroke of 12 in.

The calculation of the force exerted by the cylinder moving the carrier, called translational cylinder, is straight-forward (see drawing 4.9). The maximum required force occurs when the cylinder is pulling the carrier with the last bale on it and has to push the other bales. The maximum total weight of those bales is assumed to be 6000 lbs. A friction factor of 0.50 was used and this is twice the value obtained in our tests (to account for possible bad sliding conditions). Then a simple calculation yield the required force. This cylinder has a 3 in. bore and 42 in. of stroke. A simple welded cylinder of that size could easily handle the requirements as far as force is concerned.

Two small cylinders are used to rotate the forks on the carrier (see drawing 4.9). The calculation of their required force was obtained by a simple moment equation

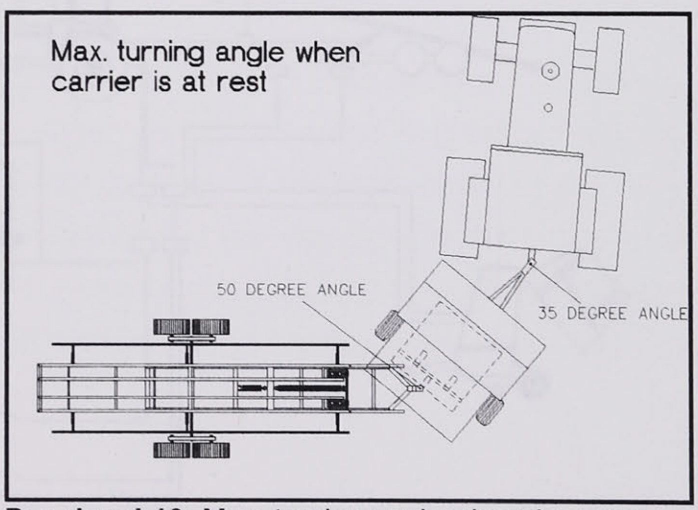


Drawing 4.9: View of the carrier with its three associated hydraulic cylinders

about the axis of rotation. Since the cylinders are pinned near the center of axis and the bale reaction force is acting approximately ten times farther, the combined force of the cylinder had to be greater than ten times the reaction force. This yielded the result shown in the above table. The size of the particular cylinder suggested for this task would be cylinders of 2.5 in. bore and 3 in. stroke. According to specifications of the manufacturer, they should be able to handle the requirements.

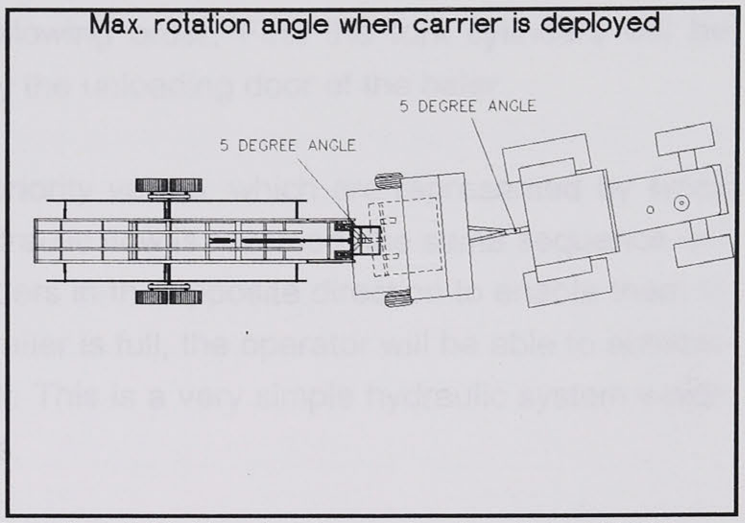
4.7 Turning angle

One problem arose in the latter part of the design. This problem concerns the limitation of the turning angle between the baler and the trailer. The two tracks in which the carrier rolls extends up to the front of the trailer frame. The back of the baler where the trailer is hooked is very low, therefore there is a risk of contact between the tracks and the baler in turns. By simulating a turn of the assembly tractor-baler-trailer, it had been found that the maximum turning angle



Drawing 4.10: Max. turning angle when the carrier is in rest position

available between the baler and the trailer is 50 degrees when the carrier is in its rest position, which is adequate. But when the carrier is in its forward position and the fork in its down position, the maximum turning angle between the baler and the trailer is 5 degrees. Passed this angle, the fork and the bottom of the baler would enter in contact and possibly damage themselves. To prevent this, a safety device connected to the hydraulic system will be installed, preventing the unloading of a bale when the turning angle between the baler and the trailer is more than 5 degrees. This safety device and others are described in the hydraulic system section.

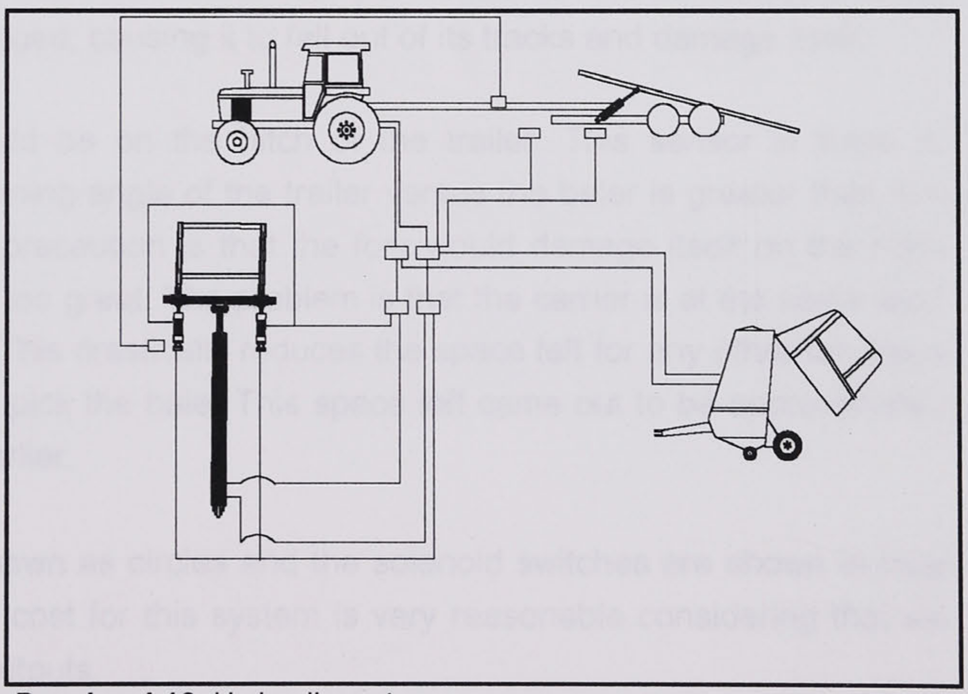


Drawing 4.11: Max. turning angle when the carrier is deployed

4.8 The hydraulic system

This part caused us some slight problems due to the a very large number of devices available on the market. This was also the part in which none of the team members were very familiar with. But we finally came up with an idea that seemed reasonable and which kept costs down.

The major problem we had was the limited number of hydraulic outlets on a conventional farm tractor. We have a total of four cylinders in our system for only two hydraulic outlets on the tractor. Therefore, we came up with the idea of having one outlet for the dumping cylinder and the other outlet to activate the other three cylinders.



Drawing 4.12: Hydraulic system

These three cylinders are going to be activated sequentially. Which means that when the operator turns on the hydraulic it will activate the cylinders in a pre-set sequence.

This sequence will be performed in the following order. First the fork cylinders will be activated, then the carrier cylinder and finally the unloading door of the baler.

This sequence will be directed by priority valves, which are represented by small red squares on drawing 4.12 . When the hydraulic flow is reversed, the same sequence will repeat itself but now it will activate the cylinders in the opposite direction to enable them to retract. Once this process is done and the trailer is full, the operator will be able to activate the dumping cylinder of the trailer to empty it. This is a very simple hydraulic system which does not require very expensive components.

One important factor we had to bear in mind here is that, we had to find some way to prevent certain cylinders to activate under certain conditions. First we had to make sure that the dumping cylinder could not be engaged unless the carrier was at its rest position. This was accomplished by putting an electric sensor on the carrier which would activate a solenoid switch. This would automatically disengage the dumping cylinder by blocking the oil flow .

A second sensor would be installed on the dumping cylinder itself. This sensor will disactivate the carrier if the cylinder is not at its rest position. This is to make sure that the carrier is not accidentally engaged, causing it to fall out of its tracks and damage itself.

The third sensor would be on the hitch of the trailer. This sensor is there to disengage the carrier if the turning angle of the trailer versus the baler is greater than five degrees. The reason for this precaution is that the fork would damage itself on the hitch bar of the baler if the angle is too great. The problem is that the carrier is at the same level as the hitch bar of the trailer. This drastically reduces the space left for any other functions than for the fork to come and pick the bale. This space left came out to be approximately the five degrees mentioned earlier.

These sensors are shown as circles and the solenoid switches are shown in blue squares in drawing 4.12. The cost for this system is very reasonable considering that we are only using two hydraulic outputs.

4.9 Rolling resistance

A simple rolling resistance estimation was conducted using the formula described in the materials and methods section. The results are presented in table 4.5.

Table 4.5: Rolling resistance for different sinkage values

Sinkage		Total weight		Rolling resistance	
(in.)	(cm)	(lbf)	(N)	(lbf)	(N)
0.00	0.00	10000.00	44482.00	357.10	1588.00
1.00	2.54	10000.00	44482.00	714.30	3176.00
2.00	5.08	10000.00	44482.00	1071.40	4768.00
3.00	7.62	10000.00	44482.00	1428.60	6356.00

The typical conditions when harvesting hay are such that the sinkage shouldn't be above 2 inches at worst. However, calculations have been made for sinkage up to 3 inches. in this worst case, the rolling resistance was calculated as 6356 N which should not require reinforcement of baler's frame. However, some tests should be made on prototypes to see if the baler`s frame are capable of withstanding this force. In the case where the system is used in bad sinking conditions, reinforcement of the baler's frame should be considered.

Those results of rolling resistance are also useful in determining the additional power required to pull the loaded trailer. For example, if the travel speed when harvesting is 1.5 m/s (5.4 km/h), the required power for pulling the system would vary from 2.3 kW with 0 sinkage to 9.5 kW with 3 in. sinkage. Again, it is a reasonable range and it would probably not require a change of tractor.

4.10 Ground pressure

The estimation of the ground pressure was done under the assumption that the total weight of the loaded picker and stacker was 10 000 lbs (4545 kg). The surface of contact of the tires with the ground is approximated by a formula commonly used in compaction studies. The width of the tire is multiplied by one half the diameter to give the contact area for each tire. In this particular case, the area for each tire was calculated to be 154 $in.^2$ (994.8 cm^2). Dividing the total weight by the total area of four tires resulted in a ground pressure of 16.2 psi (112 kPa). This value falls within the range of ground pressure exerted by typical farm machinery.

4.11 Cost of the machine

This is an approximate cost evaluation of the trailer. The material costs of the machine were separated in 4 items as illustrated in table 4.6. The labour costs encountered by the assembly of the machine are not written on the table. This is because this evaluation is even more approximate than the material costs evaluation. But a price range for labour was figured out, which would be between 5 000.$ and 10000.$, for a total price of fabrication between around 10 000.$ and 15 000.$.

Table 4.6: Material cost approximation

Components	Cost
Metal and wheels	4 000.$
Hydraulic cylinders	1 500.$
Solenoid switches	500.$
Hydraulic valves	250.$
Total	6 250.$

It is to be noted that the prices given for material are retail prices and the labour costs are for the building of one unit. If the trailer could be produced in series and in large quantities, the cost of material and labour may decrease a lot. Therefore it can be supposed that a realistic retail price range for one unit could be between 10 000.$ and 15 000.$.

4.12 General discussion

Up to now, there are only two possibilities for the farmer to gather the round bales. The first one is to pass through the field with two tractors and a trailer. One of the tractor's is equipped with a loader and the other one to pull the trailer. This requires two operators and it also requires an extra passing throughout the entire field to gather the bales.

The second possibility that he has is to buy a special trailer designed to pick and stack the bales without the need of the loader. The different systems investigated all had the same limitations. These limitations are the number of bales it can carry and the fact that you have to pass in the entire field again.

This is the difference with our design. It can only carry up to five bales but there is a reason for this. The objective of our trailer is not to carry the bales directly home but rather to gather them all at the same place in the field. We evaluated that with a capacity of five bales, it would be enough to reach the end of an average field. By delivering the bales at the end of the field, the farmer will not have to go through the entire field to gather his bales. This means a reduction in time loss and compaction.

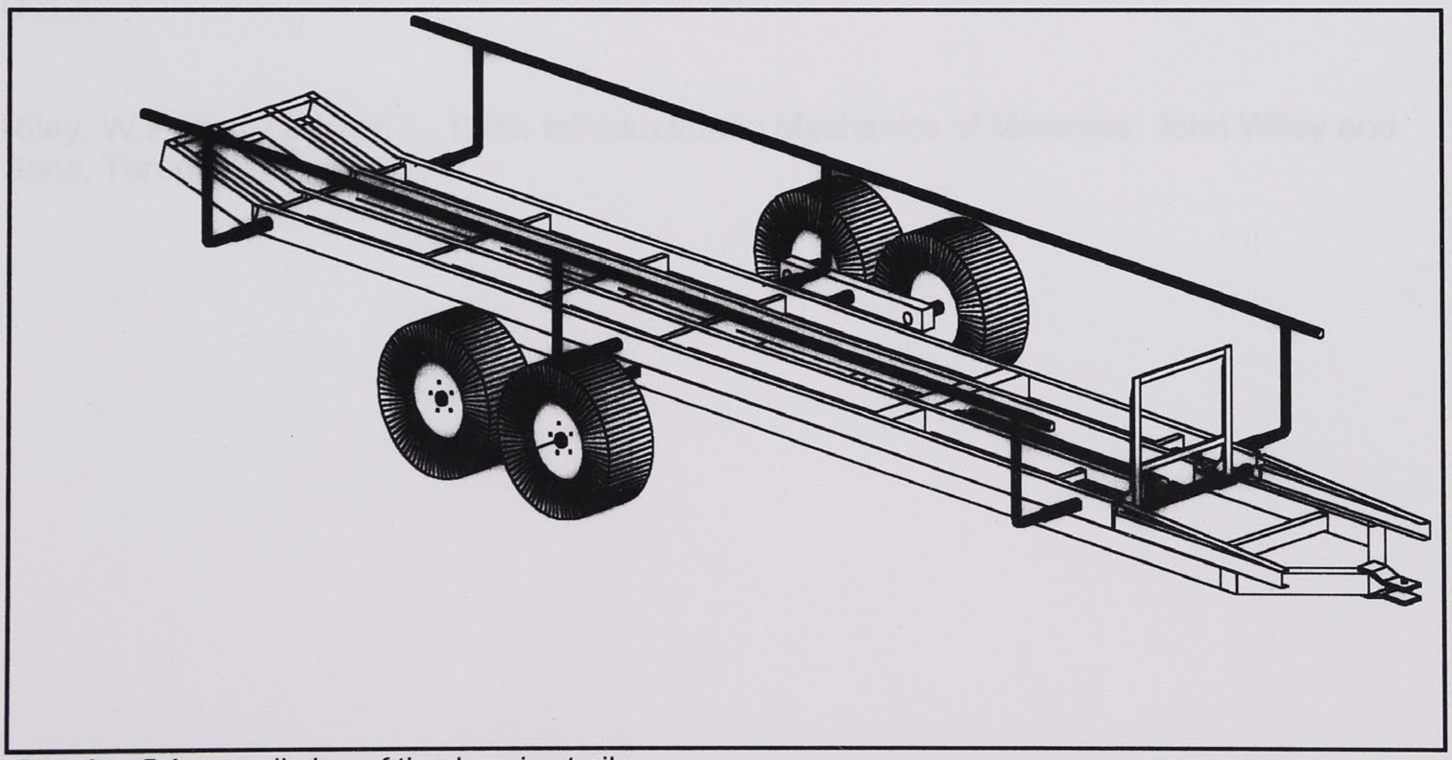
There might be some values that are not entirely accurate. Since we did not test our design, we had to assume certain values which might be off slightly from the exact value. The time limiting factor did not permit us to go into too many details in our design calculation process.

The overall cost of the trailer, which is between \$ 10 000 and \$ 15 000, is probably too high for what it is designed for. The only way that such a machine can be profitable is if the amount of bales handled is large enough. There must be a way to make a cheaper machine which does the same job. The reason why this design is so expensive is that the metal needed to build the trailer corresponds to 2/3 of its base costs.

5. CONCLUSION

To conclude this report, It is possible to say that the objective was fully achieved. The mechanism obtained is simple and does not require a large increase in horsepower demand from the tractor. From all the alternatives obtained, it was the most efficient for the job it had to perform (see drawing 5.1). The price range of \$10 000 to \$15 000 is somewhat high but it was expected. This machine could be brought onto the market, maybe at a somewhat lower price, to help the farmer to be more efficient.

All the design criteria were met. All the components should be strong enough to withstand the possible loads applied. It is possible that some assumptions are not exact but this will not have a great influence on the general outcome.



Drawing 5.1: overall view of the dumping trailer

REFERENCES

Bell, B. 1989. Farm Machinery. Farming Press Book, Ipswich, U.K.

Culpin, C. 1984. Farm Machinery. Collins, London, U.K.

Grainger, W.W. Inc. 1992. Fall 1992 General Catalog. Grainger, Soyth Vermont, U.S.A.

Jacobs, C.O. and Harrell, W.R. 1983. Agricultural power and Machinery. McGraw-Hill Book Company, New-York, U.S.A.

Juvinall, R.C. and Marshek, K.M. 1991. Fundamentals of Machine Component Design. John Wiley and Sons, Toronto, Canada.

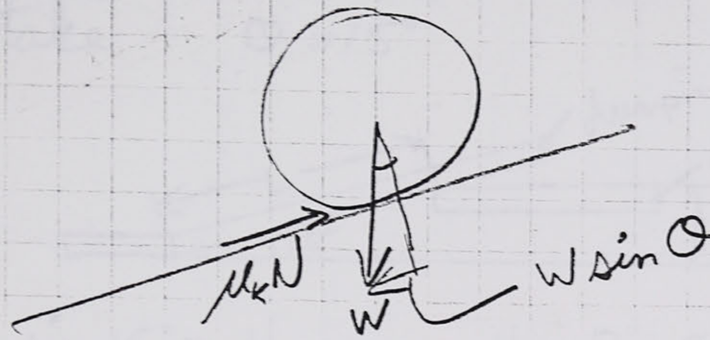
Meriam, J.L. and Kraige, L.G. 1987. Engineering Mechanics. John Woley and Sons, Toronto, Canada.

Mills, R.K. 1986. Implement and Tractor. Intertec Publishing Corporation, Overland Park, U.S.A.

Riley, W.F. and Zachary, L. 1989. Introduction to Mechanics of Materials. John Wiley and Sons, Toronto, Canada.

APPENDICES

Critical dumping angle calculation



$N = W\cos\theta$

What is μ_k

Wood wagon : 350 psi press

weight of bale : ~~625~~ 625 psi

$N = 625\,psi\,A$

$\mu_k N = 350\,psi\,A$

$\mu_k = \frac{\mu_k N}{N} = \frac{350}{625}$

$\mu_k = 0.56$

Result of test 4

read. press. 150 psi $\mu_k = \frac{\mu_k N}{N} = \frac{150}{625} = 0.24$

need $W\sin\theta > \mu_k N = \mu_k W\cos\theta$

$\cancel{W}\sin\theta > \mu_k \cancel{W}\cos\theta$

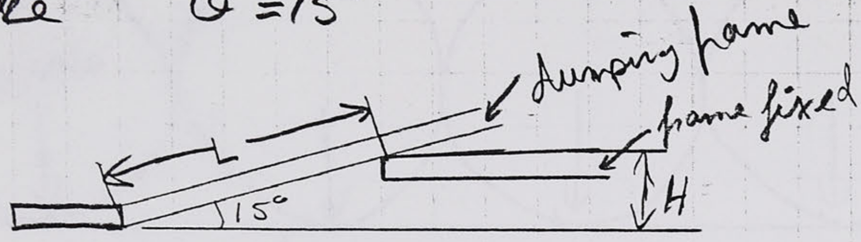
$\tan\theta > \mu_k = 0.24$

$\theta_{crit} = \tan^{-1}(0.24) = 13.5°$

we'll use 15° for safety

Calculation of hinge point for dumping

take $\theta = 15°$



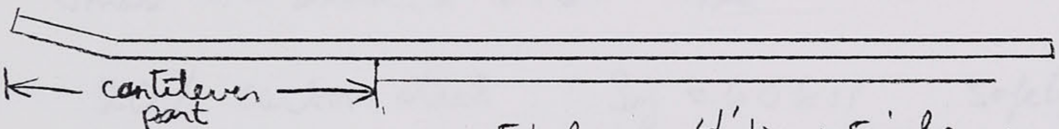
$\sin 15° = \frac{H}{L}$ $\qquad H = 12'' + 5'' = 17''$

$L = \frac{17''}{\sin 15°} = 66''$ use $60''$ → because don't want to touch the ground

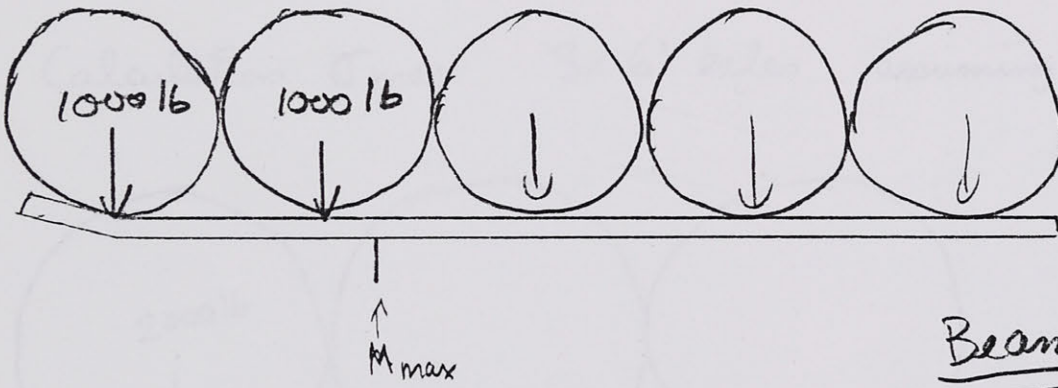
$H = \sin 15 \times 60'' = 15.5''$

So $L = 60''$ a clearance of $1.5''$ is obtained

Calcul. σ_{max} $\qquad$ 5×4' bales



5 bales 4' dia × 5' large 1000 lb each max



5 in thickness

6.7 lb/ft linear weight

Beam C5×6.7 $I = 7.49\ in^4$

$M_{max} = (1000\ lb)(12\ in) + (1000\ lb)(60\ in) = 72000\ in \cdot lb$

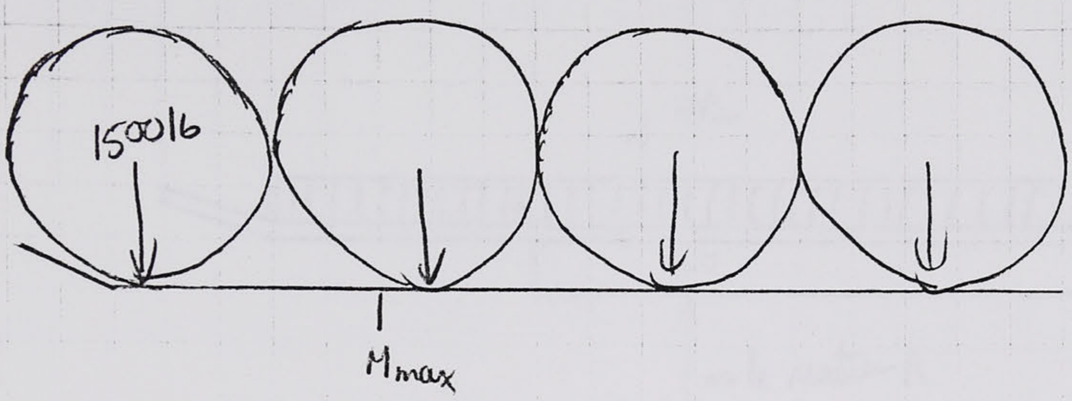
$\sigma_{max} = \frac{Mc}{I} = \frac{(72000\ in \cdot lb)(2.5\ in)}{7.49\ in^4} = 24032\ lb/in^2$

σ_{max} one beam $= 6008$ psi $= 6$ ksi

assume a 20% carbon steel with $S_y = 60$ ksi

safety factor of 10

Calculation σ_{max} 4×5' bales



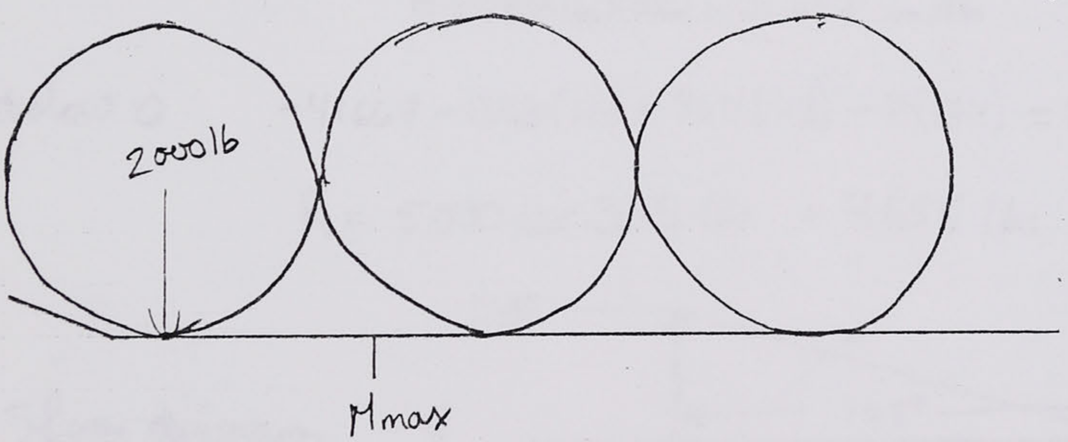
$M_{max} = (1500\text{ lb})(54\text{ in}) = 81000\text{ in}\cdot\text{lb}$

$\sigma_{max} = \frac{Mc}{I} = \frac{(81000\text{ in}\cdot\text{lb})(2.5\text{ in})}{7.49\text{ in}^4} = 27036\text{ lb/in}^2$

σ_{max} one beam $= 6760\text{ lb/in}^2$

20% carbon steel $S_y = 60$ ksi safety factor = 8,9

Calculation σ_{max} 3×6' bales assuming they're all at back for worst case



1 psi = 6894.76 Pa

1,0 ksi = 6894760 Pa

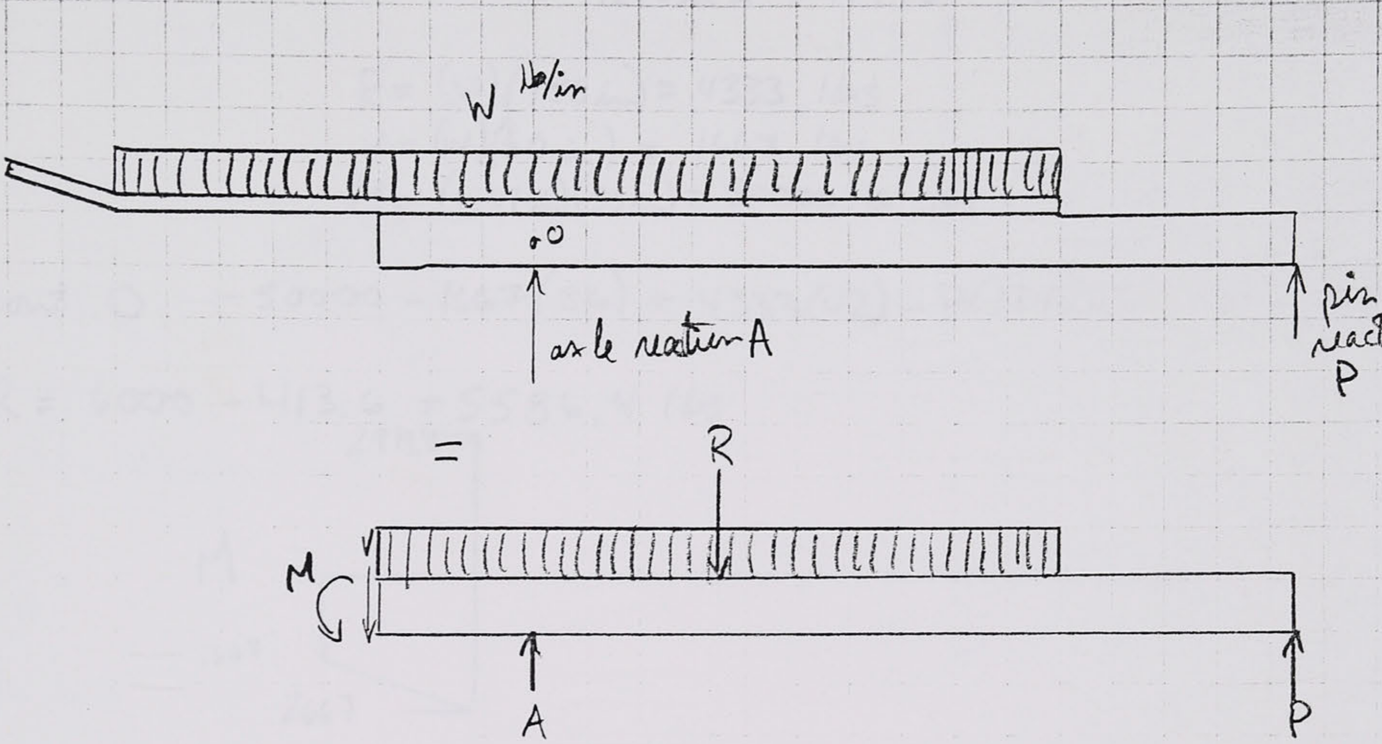
$M_{max} = (2000\text{ lb}) \times (48\text{ in}) = 96000\text{ in}\cdot\text{lb}$

$\sigma_{max} = \frac{Mc}{I} = \frac{(96000\text{ in}\cdot\text{lb})(2.5\text{ in})}{7.49\text{ in}^4} = 32043\text{ lb/in}^2$

σ_{max} one beam = 8,01 ksi

20% carbon steel $S_y = 60$ ksi safety factor = 7,5

STRESS IN FIXED FRAME



5x1000 lb bales

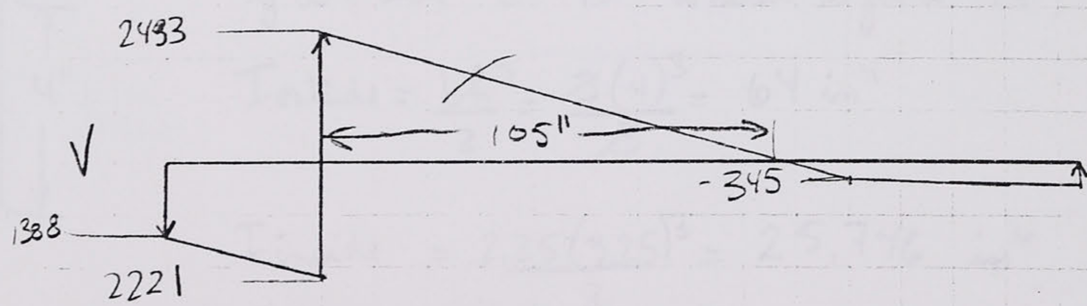
$$W = 5000 \text{ lb}/216 \text{ in} = 23,15 \text{ lb/in}$$

$$R = (W)(156 \text{ in}) = 3611 \text{ lbs}$$
$$V = (W)(60 \text{ in}) = 1388 \text{ lbs}$$
$$M = (1388 \text{ lbs})(30 \text{ in}) = 41\,667 \text{ in}\cdot\text{lb}$$

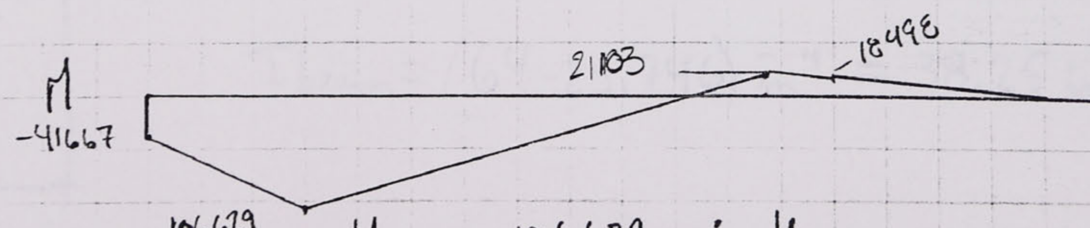
Moment about O $\quad -41667 - 1388(36) + 3611(42) - P(174) = 0 \qquad P = 345 \text{ lbs}$

$$R = 5000 \text{ lbs} - 345 \text{ lbs} = 4655 \text{ lbs}$$

Shear diagram



Bending moment diagram



$M_{max} = 106629 \text{ in}\cdot\text{lb}$

4×1500 lb bales $\quad w = 6000\ lb/216\ in = 27.778\ lb/in$

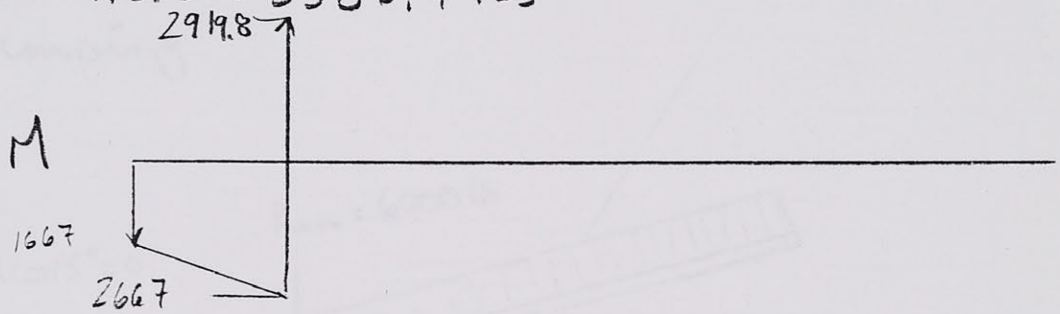
$R = (w)(156\ in) = 4333\ lbs$

$V = (w)(60\ in) = 1667\ lbs$

$M = (1667\ lbs)(30\ in) = 50000\ in\cdot lbs$

Moment about O $\quad -50000 - 1667(36) + 4333(42) - P(174) = 0 \quad P = 413.6\ lb$

$R = 6000 - 413.6 = 5586.4\ lbs$

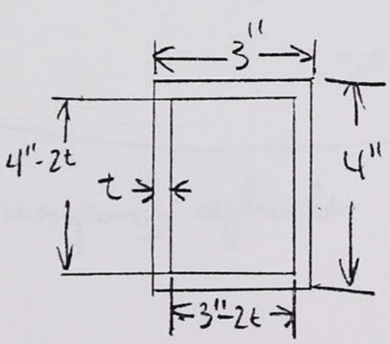


right there $\quad$ Mmax can be calculated

$M_{max} = 1667 \times 36 + 1000\frac{(36)}{2} = 128\,000\ in\cdot lb$

3×2000 lb bales $\quad$ same thing $\quad M_{max} = 128\,000\ in\cdot lb$

We use a 4"×3" exterior dim. beam $\quad$ we need the thickness



if we use $t = 3/8"$ which is quite std.

$I_{outside} = \frac{bh^3}{3} = \frac{3(4)^3}{3} = 64\ in^4$

$I_{inside} = \frac{2.25(3.25)^3}{3} = 25.746\ in^4$

$b = 3 - 3/4 = 2.25$

$h = 4 - 3/4 = 3.25$

$I_{beam} = (64 - 25.746)\ in^4 = 38.25\ in^4$

if $t = 0.25"$ $\quad I_{in} = 35.73\ in^4$

$I_{beam} = 64 - 35.73$

$= 28.27\ in^4$

The max. Moment of all cases = $128^{\times 10}$ in·lb

t=0.375 $\quad \sigma_{max} = \frac{MC}{I} = \frac{(128000 \text{ in·lb})(2 \text{ in})}{38.25 \text{ in}^4} = 6692.8$ psi$/2 = 3346$ psi/beam

~~252.5~~

t=0.25 $\quad \sigma_{max} = \frac{(128000 \text{ in·lb})(2 \text{ in})}{28.27 \text{ in}^4} = 9055$ psi$/2 = 4527$ psi/beam

~~252.9~~

if 20% C $\quad S_y = 60$ ksi $\quad$ Safety factor = 17.9 & 13.25

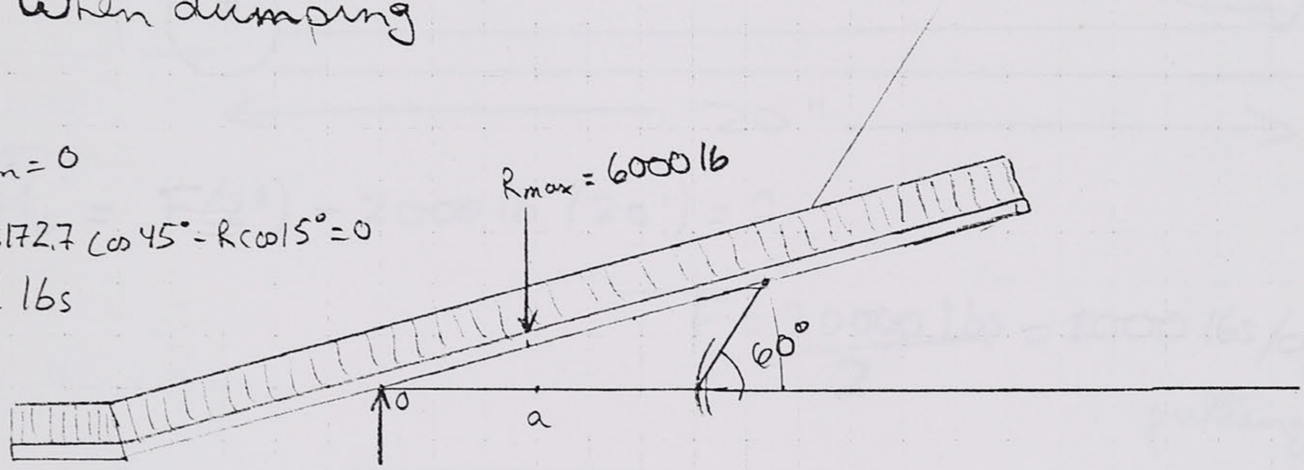
use t = 1/4"

When dumping

Fy on C beam = 0

00 cos 15° − 3172.7 cos 45° − R cos 15° = 0

R = 3677.4 lbs

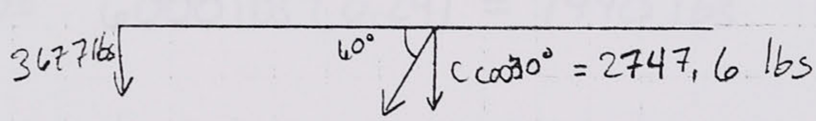


Moment about 0 $\quad (6000)\cos 15°(36") - C(\cos 45°)(93") = 0$

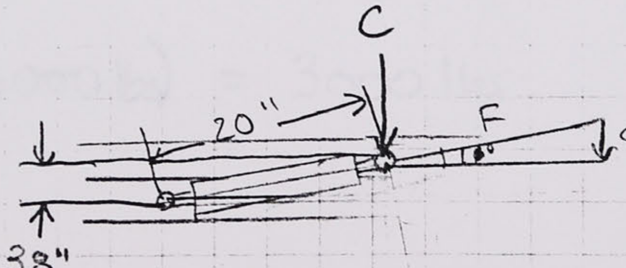
Moment about a at position a $\quad C = 3172.7$ lbs

= 3677.4 lbs (36 in) =

frame



Dumping cylinder force



$\frac{c}{F} = \sin\theta$

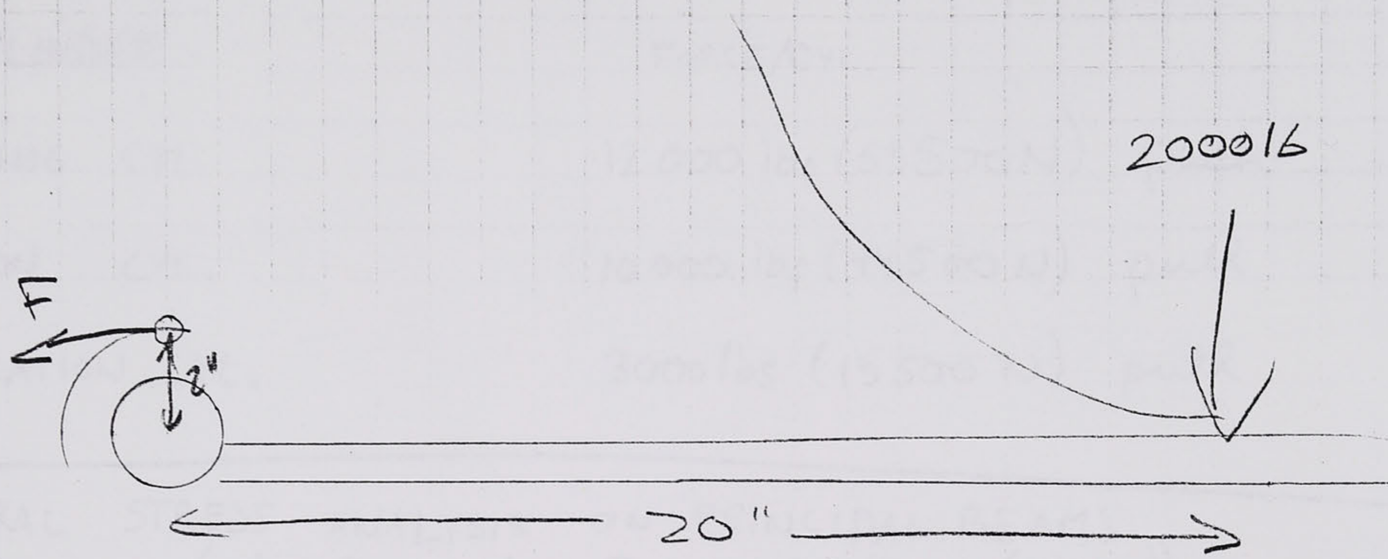
$F = \frac{2322}{\sin 11°} = \boxed{12172}$

pushing

Moment about O

$6000(36") - C(93") = 0 \quad C = 2322$

rotational cylinders



$M_O = F(2") - 2000\,lb\,(20") = 0$

$F = \dfrac{20000\,lbs}{2} = 10000\,lbs$/cylinder

pulling

---

translational cylinder

$\mu_K = 0.24$

worst $N = 6000\,lbs$

$F = \mu_K N = 6000\,lbs\,(0.24) = 1440\,lbs$ pulling

assume bad $\mu_K = 0.5$

$F = 0.5\,(6000\,lbs) = 3000\,lbs$

RESULTS

REQUIRED CYLINDER CAPACITY

CYLINDER	FORCE/CYL.
DUMPING CYL.	12 000 lb_f (53500 N) push
ROTATION CYL.	10 000 lb_f (44500 N) pull
TRANSLATION CYL.	3000 lbs (13 500 N) pull

FLEXURAL STRESS ANALYSIS ON PRINCIPAL BEAMS
(using 20% carbon steel $S_y = 60ksi$ (414 MPa))

Beam	Max. stress.*	Safety factor
Dumping C5×6.7 frame	8.0 ksi (55230 MPa)	7.5
Fixed 4in. × 3in. frame	4.5 ksi (31210 MPa)	13.3

* considering only the weight of the bales

Rolling Resistance

$$\frac{R}{N} \approx \left(\frac{\delta_T + Z}{d}\right)$$

11-15 tires $b = 11"$
$d = 28" = 2r$ $r = 14"$
$r_L = 13"$

$N = \underset{\text{trailer}}{4000} + \underset{\text{bales}}{6000} = 10000 \text{ lbs}$

$\delta_T = r - r_L = 14" - 13" = 1"$

if $Z = 0"$ $R = 10000 \text{lbs} \left(\frac{1" + 0"}{28"}\right) = 357.1 \text{ lbs}$

$Z = 1"$ $R = 10000 \text{lbs} \left(\frac{1" + 1"}{28"}\right) = 714.3 \text{ lbs}$

$Z = 2"$ $R = 10000 \text{lbs} \left(\frac{1" + 2"}{28"}\right) = 1071.4 \text{ lbs}$

$Z = 3"$ $R = 10000 \text{ lbs} \left(\frac{1" + 3"}{28}\right) = 1428.6 \text{ lbs}$

Slide

Rolling resistance for diffrent sinkage values

sinkage	Rolling resistance
0	357 lb (1588 N)
1 in. (2.54 cm)	714 lb (3176 N)
2 in. (5.08 cm)	1072 lb (4768 N)
3 in. (7.62 cm)	1429 lb (6356 N)

il compaction

$P = \frac{N}{A}$ A for wheel use $L = \frac{d}{2} = \frac{28''}{2} = 14''$

$b = 11''$

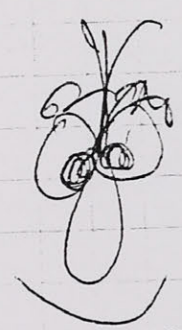
$A = b \times L = 14'' \times 11'' = 154\ in^2$

$P = \frac{10000\ lbs}{4 \times 154\ in^2} = 16.2\ psi = 112\ kPa$

Slide

Estimated ground pressure
at full load = 10000 lbs (4545 kg)
= 16.2 psi (112 kPa)

ick 1 — 1) Titre ✓
2 — 2) obj ✓
3 — 3) Bak ✓
4 — 4) Alt. ✓
5 — 5) Design ✓
7 — 6) exp. rest ✓
11 — 7) cyl. capacity ~ ✓
12 — 8) ~~Flexural~~ stress ~ ✓
10 — 9) Rolling Res.
13 — 10) Est grnd P ~ cut.
Rick 9 — 11) Cost
14 — 12 Conclusion



Slides photos

Test 1 ✓
" 2 ✓
" 3 ✓
Poid balle ✓
devant NH ✓
derrière NH ✓
devant JD ✓
derrière JD ✓
Flexiballe ✓