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Dynamic Stability of Two Tractor Front-End Loader Systems

Michael J. Bader
University of Georgia

Linus R. Walton
University of Kentucky, lwalton@ca.uky.edu

Larry G. Wells
University of Kentucky, larry.wells@uky.edu

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M. J. Bader, L. R. Walton, L. G. Wells

Abstract. *A one-quarter scale model tractor and loader were designed, fabricated, and used in an experiment to evaluate two tractor-loader configurations relative to stability using roll angle as the means of assessment. The non-conventional loader system consisted of a conventional loader attached to a steerable carrier which in turn was attached to the tractor by the front axle and drawbar of the tractor. The stabilizing axle for the non-conventional loader system was the front axle as opposed to the conventional system in which the stabilizing axle was the rear axle. The experiment showed that the non-conventional loader had an inherent advantage in stability as assessed by roll angle. The roll angle was smaller at higher velocity than at lower velocity because the tires underwent greater deformation at the higher velocity than at the lower velocity.* ***Keywords.*** *Tractor, Front-end loader, Stability, Safety, Scale model.*

The sweep rake was the fore-runner of the front-end loader. A review of company archival literature by King (1992) showed that Deere & Co. first manufactured their sweep rake and hay stacker in 1930. A 1931 photo of a Case industrial tractor is shown with a fabri-form loader which is a scoop using wire cable to slide up and down on two vertical members (Letournean, 1992). Deere & Co. introduced their hydraulic manure loader in 1953 (King, 1992). The front-end loader has progressed from the initial perception of a one-operation implement to the multi-purpose implement it is today.

With the introduction of round balers and the experimentation with mechanically harvesting burley tobacco, the role of a tractor outfitted with a front-end loader is changing. The front-end loader is being used to lift, transport, and stack round hay bales. These bales can weigh up to 900 kg and, if stacked, may need to be raised to a height of 3.8 m. The front-end loader (Bader et al., 1990; Casada et al., 1987; Wells et al.,1990; Walton et al., 1985) is used as a means of handling frames of burley tobacco that weigh from 800 to 1200 kg and may need to be stacked to a height of 4.6 m. The stability of the tractor-loader combination needs to be analyzed under these new usages.

The standard or conventional front-end loader is attached to the main frame of a tractor. In the loader's raised position, the center of gravity of the tractor-loader system is raised and moved backward, thus causing the system to become less stable. An alternate tractor-loader system has been developed which uses a steerable carrier upon which the loader is mounted (Walton et al., 1985). Since this loader is not mounted rigidly to the main frame of the tractor, but instead to the frame of the steerable carrier, the response of this system will be different than that for the standard tractor-loader-load system.

A better understanding of the transient motions of a standard tractor-loader combination is needed to determine the kinds and sizes of loads which can be safely transported. It is important to determine whether or not the steerable carrier is more stable and thereby safer than the conventional tractor-front-end loader system.

The overall objective of the research was to compare the dynamic stability (as measured by roll angle) of a wheeled agricultural tractor equipped with a front-end loader mounted to the front axle as opposed to the conventional front-end loader mounting arrangement. Specific objectives were to:

1. Design and fabricate scale models of two tractor-loader systems.
2. Experimentally determine the dynamic response of each system to various destabilizing conditions.
3. Interpret kinematic response of both systems with regard to safety of operation.

Materials and Methods

Experimental Model Tractor and Loader

The advantages of a small-scale tractor and loader are the small area required for operation and ease of physically handling the model while those of a larger scale model are a more realistic construction of the tractor. A one-quarter scale model was chosen for this research because the smallest pneumatic tire available was in this range of scaling. It was not possible to scale all of the tractor parameters such as spring constant of the tires, damping coefficients, and mass moments of inertia. However, parameters such as velocity, wheel base, wheel diameter, mass, etc., could be scaled adequately.

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The authors are **Michael J. Bader**, ***ASAE Member Engineer,*** Associate Extension Professor, Department of Biological and Agricultural Engineering, University of Georgia, Tifton, Ga.; **Linus R. Walton**, ***ASAE Member Engineer,*** Chair and Professor, and **Larry G. Wells**, ***ASAE Member Engineer,*** Professor, Department of Biosystems and Agricultural Engineering, University of Kentucky, Lexington, Ky. **Corresponding author:** Linus R. Walton, Dept. of Biosystems and Agric. Engineering, University of Kentucky, 128 Agricultural Engineering Bldg., Lexington, KY 40546-0276; tel.: (606) 257-3000; fax: (606) 257-5671; e-mail: <lwalton@bae.uky.edu>.

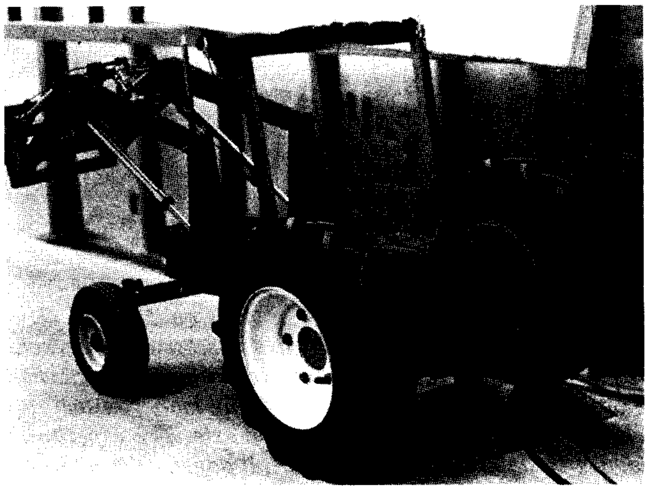

Figure 1–Scale model tractor.

The model tractor and loader (fig. 1) was scaled using a typical 100 hp tractor. Component specifications were:

1. Front tires: Size 2.80 × 4.0 pneumatic tires.
2. Rear tires: Size 4.80 × 8.0 pneumatic tires.
3. Electric motor: 1/6 hp dc.
4. Clutch: Electric clutch brake.
5. Transmission: Modified 4-speed manual transmission.
6. Differential: 4.6 to 1 ratio.
7. Hydraulic cylinders modeled from John Deere model 158 loader. Cylinder construction is shown in figure 2.

Each of the two loader systems were attached to the same model tractor. The loader was mounted to the main body of the model tractor to study the conventional loader mounting system as shown in figure 3. In the conventional tractor-loader-load system, the loader frame and load move with the main tractor body. To study the nonconventional loader mounting, the loader was attached to the front tractor axle by a metal frame as shown in figure 4. This frame was also attached to a pivot point at the rear of the tractor behind the differential housing. In the nonconventional loader system, the loader frame and load move in relation to the front axle.

Since the loader is mounted to the main tractor body in the conventional system, the load on the front-end loader reduces the forces on the rear tires which provide stability for the tractor- loader-load system. When the rear wheel transverses a bump, the subsequent load shift may cause overturn. For the nonconventional tractor-loader-load system, the load on the front-end loader increases the force on the front tires and provides stability for the tractor-loader-load system. As a front wheel traverses a bump, the load shift will work against the system, but the greater initial force on the stabilizing wheels should theoretically provide more resistance to overturn.

The model tractor was powered by an electric motor which was powered by a deep cycle 12 V automotive battery. The battery was mounted on a platform which was conveyed along the path of the tractor. This arrangement allowed a large battery to be used in the experimental tests, which, in turn, gave a more uniform voltage to be applied to the driving motor between runs.

The terrain (fig. 5) over which the models ran was a plane concrete surface with a broom finish and a sinusoidal-shaped obstacle input to the upslope tires. The test course consisted of a metal frame with a plywood floor and concrete on top of the plywood. The test course was

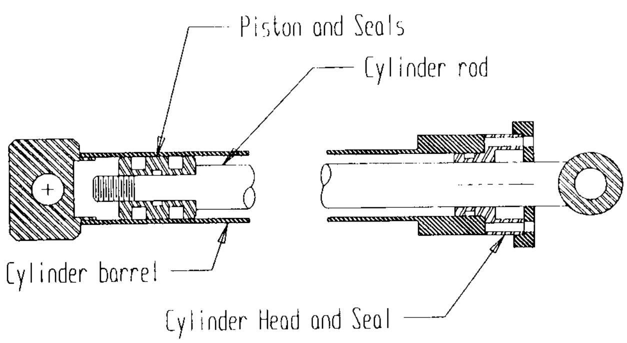

Figure 2–Loader cylinder construction.

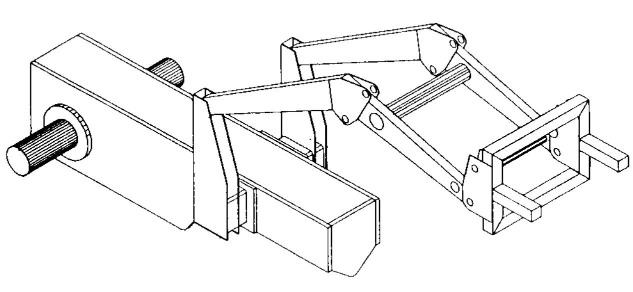

Figure 3–Conventional loader mounting system.

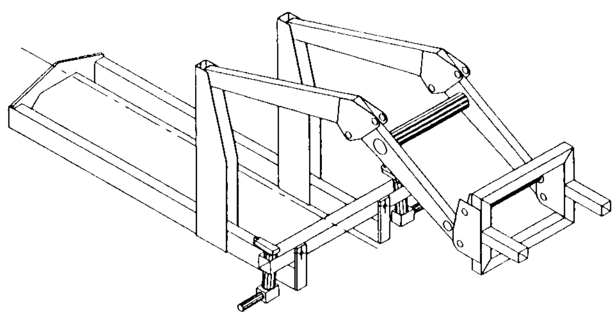

Figure 4–Nonconventional loader mounting system.

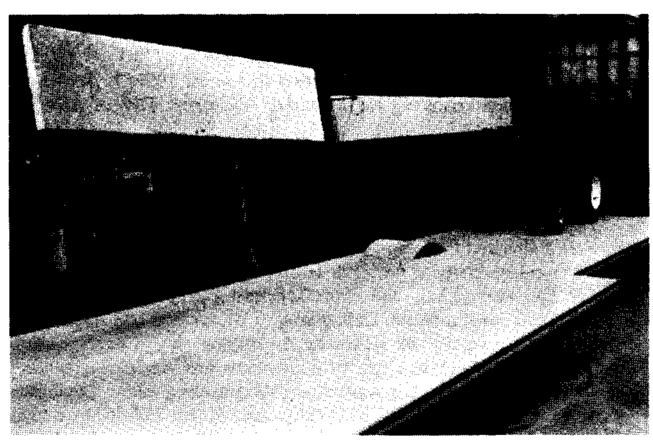

Figure 5–Experimental apparatus configuration.

hinged about one side so that the slope of terrain to be traversed by the models could be varied. The test course was 4.9 m long and was 1.2 m wide for the first 2.5 m of the course and 1.4 m wide for the remainder. The obstacle was described by the equation:

$$y = H \sin(\pi x/L) \quad (1)$$

where

y = height of obstacle at distance x (cm)
H = the height of obstacle (cm)
L = the length of obstacle (cm)
x = horizontal distance along the obstacle (cm)

Sonic Digitizer

A sonic digitizer was used to monitor the transient response of the two tractor-loader systems. Emitters mounted on the model tractor emitted a sound created by a small electrical arc. The sonic digitizer measured and recorded the time required for sound to travel from emitters to stationary microphones positioned above the surface. Using the speed of sound and the time measurements, distances from each microphone to each emitter were determined and stored in a computer. The position of each emitter was determined as a function of time to an accuracy of ± 0.5 mm in the x, y, and z directions, which in turn provided the position of the wheels and body of the tractor. Details concerning the sonic digitizer are provided by Bader et al. (1996).

Experimental Procedure

A series of experiments were conducted using one-quarter scale models of the two tractor-loader-load configurations to determine the relative stability of the two systems using roll angle as the assessment of stability. The roll angle is defined as the rotation of the body about its longitudinal axis. It is the roll angle plus the slope that, when large enough, causes overturn of the tractor. The front axle rotation angle was also determined to provide additional information. These experiments were conducted using the terrain described above. Each model (conventional and nonconventional loader) was placed perpendicular to the slope of the terrain table and made to traverse sinusoidal obstacles with heights of 1.9 and 3.8 cm. Treatment variables considered were terrain slope (10 and 15°), load weights (6 and 12 kg), load height (22.9 and 45.7 cm), and tractor velocity (first and second gear which correspond to model velocities of 34 and 110 cm/s and full-scale tractor velocities of 2.4 and 7.9 km/h). Each test was replicated three times. Differences among treatments were determined by analysis of variance.

The experiment began by selecting a loader configuration and a terrain slope. Once this combination of loader configuration and terrain slope was fixed, all combinations of load heights, load weights, tractor velocities, and obstacle heights were run. Once this set of tests was made, the slope was changed and tests of random combinations of load heights, load weights, tractor velocities, and obstacle heights were conducted on the second slope. The loader configuration was then changed and the process was repeated.

Results and Discussion

The analysis of variance showed that loader arrangement, tractor velocity, bump height, load height, and load weight had a significant effect on roll angle at the 1% level and also showed that tractor velocity, bump height, and load height had a significant effect on front axle rotation angle at the 1% level. In general, this discussion will focus on the effect of treatments and interactions on roll angle and will include a discussion of front axle rotation angle (θ) only when it helps to understand the effect of treatments on roll angle.

The mean values of roll angle and theta as a function of the treatments are shown in table 1. The experiment was able to discern differences in means of roll angle among treatments of less than 0.1°. A very important result was the non-conventional loader producing a lower roll angle than the conventional loader. The mean difference was only 0.09° but the difference was as much as 1.13° for maximum load, load height, bump height, and velocity. The non-conventional loader arrangement has an inherent advantage over the conventional loader from the standpoint of safety with all other things being equal. The advantage in safety can be enhanced even further by lengthening the wheel base and widening the front wheels of the carrier. These are decisions that engineers make when designing a carrier for safety as opposed to designing a tractor for versatility.

There was no difference between roll angle on the 10° and 15° slopes (table 1). Clearly, the slope contributes to overturns because the roll angle and slope act together to produce the instability that creates the overturn. This result shows that the slope neither exacerbated nor retarded the angle of roll.

The surprising result was that the mean roll angle was 12% lower for the high tractor velocity than for the low tractor velocity. An explanation for this result can be found by noting that mean front axle rotation angle was 19% lower for the high velocity than for the low velocity. This indicated that the higher velocity placed a greater force on the tires which caused more deformation as they traversed

Table 1. Mean values of roll angle and front axle rotation angle as a function of loader arrangement, slope, bump height, load weight, load height, and velocity

Treatments			Roll Angle (°)	Front Axle Rotation Angle (°)
Loader arrangement:				
Non-conventional		(L1)	3.37a*	2.98a
Conventional		(L2)	3.46b	2.95a
Slope: 10°		(S1)	3.40a	2.92a
15°		(S2)	3.43a	3.01a
Bump height:	1.9 cm	(B1)	2.31a	1.91a
	3.8 cm	(B2)	4.52b	4.02b
Load weight:	6 kg	(W1)	3.38a	2.94a
	12 kg	(W2)	3.45b	2.99a
Load height:	22.9 cm	(H1)	3.37a	2.88a
	45.7 cm	(H2)	3.46b	3.05b
Tractor velocity:	34 cm/s	(V1)	3.63a	3.28a
	110 cm/s	(V2)	3.20b	2.65b

* Means within a treatment followed by different letters are significantly different at the 1% level.

the bump which in turn produced a lower roll angle. In other words, the tires absorbed more of the shock at higher velocities than at lower velocities. The reduction in roll angle with increased velocity was consistent across all combinations of treatments with the singular exception of the combination of larger load, higher loader height, higher bump and higher slope treatment which caused the roll angle to increase by 9.7% as the velocity increased.

Bump height had the greatest effect on roll angle as was expected. The increase in roll angle was approximately equal to the increase in bump height, when the bump height doubled, the roll angle doubled.

Both amount and height of load had an influence on the angle of roll (table 1). The heavier and higher load caused a 2% greater angle of roll than the lighter and lower load, respectively. In these instances the lateral acceleration of the load was exacerbated as the tractor traversed the bump by the increased load and increased lever arm, respectively.

Significant second-order interactions involving roll angle were loader arrangement-load weight (L×W), loader arrangement-velocity (L×V), slope-bump height (S×B), slope-velocity (S×V), bump height-load height (B×H), bump height-velocity (B×V), and load height-velocity (H×V). Mean values of roll angle as a function of significant second order treatment interactions are shown in table 2. The loader arrangement and load weight interaction was significant because the nonconventional loader (L1) showed no change in roll angle with increased load but the mean roll angle increased by 0.13° (9%) for the conventional loader (L2) as the load was doubled. The design of the nonconventional loader system resulted in the load being attached to the front axle so the added load would not affect the roll of the main tractor body whereas the conventional loader is attached directly to the main tractor body thus increasing the effect of load on the magnitude of roll angle.

The loader arrangement — velocity (L×V, table 2) interaction was significant because the conventional and nonconventional loader had the same mean roll angle at the lower velocity but the nonconventional loader had a lower mean roll angle at higher velocity than did the conventional loader. Apparently, the nonconventional loader design exhibited moderately greater stability at a higher velocity than the conventional loader resulting in the significant loader arrangement-velocity interaction.

Table 2. Mean values of roll angle for significant second-order treatment interactions

Treatment	Roll Angle (°)	Treatment	Roll Angle (°)
L1×W1*	3.37	S1×B1	2.33
L1×W2	3.37	S1×B2	4.46
L2×W1	3.39	S2×B1	2.29
L2×W2	3.52	S2×B2	4.57
L1×V1	3.63	S1×V1	3.66
L1×V2	3.12	S1×V2	3.13
L2×V1	3.63	S2×V1	3.60
L2×V2	3.28	S2×V2	3.27
B1×H1	2.30	H1×V1	3.63
B1×H2	2.33	H1×V2	3.11
B2×H1	4.45	H2×V1	3.63
B2×H2	4.58	H2×V2	3.29
B1×V1	2.49		
B1×V2	2.14		
B2×V1	4.77		
B2×V2	4.26		

* See table 1 for explanation of treatments.

The slope-bump height interaction (S×B, table 2) showed that the 15° slope caused a larger increase in roll angle as bump height was increased from 1.9 cm to 3.8 cm than the 10° slope. This result was probably caused by instability at the combination of high slope and high bump height.

The slope-velocity interaction (S×V, table 2) showed that the reduction in roll angle with increased velocity was greater at the lower slope than at the higher slope. Since it was established earlier that the reduction in roll angle with increased velocity was a function of force on the tires, it is clear that the greater force on the tires perpendicular to the surface occurs at the lower slope and a lesser force on the tires occurs at the higher slope.

The interaction of bump height and velocity (B×V, table 2) was significant because the magnitude of roll angle decreased more for the high bump than for the low bump. This was probably caused by a difference in tire forces and deformation for the combinations of bump height and velocity.

The interaction of bump height and load height (B×H, table 2) showed that increase in roll angle was greater from low to high load height as the tractor-loader traversed the high bump as compared to the low bump. This effect is probably a function of load acceleration which would be greater for the higher bump height than for the lower bump height.

The interaction of load height and velocity (H×V, table 2) showed that the decrease in roll angle with increased velocity was greater for the low load height than the high load height. This result was probably caused by greater acceleration of the load at the higher load height and high bump height.

Conclusions

Conclusions based on this research were as follows:

1. The scale model tractor front-end loader system permitted effective evaluation of system performance.
2. The nonconventional loader system was superior to the conventional front-end loader system from a safety standpoint as assessed by roll angle.
3. The tires deformed at high velocity to the extent that roll angle was less at the higher velocity than at the low velocity.
4. Larger load weight and higher load height produced larger roll angles.
5. Slope became a significant factor on roll angle only at higher bump heights when the bump height and slope combined to produce instability.

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