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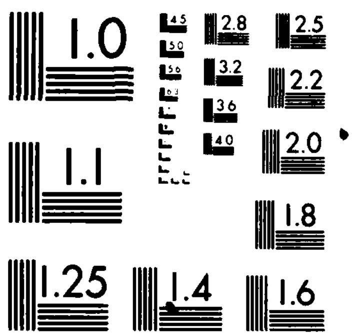

AD-A172 368

RESEARCH
DEVELOPMENT
MANUFACTURING

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Final Report

For

45 MPH - 6,000 Pound

and 10,000 Pound

Rough Terrain Fork Lift Truck

Feasibility Study

Contract Number

DAAK70-85-C-0111

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CATERPILLAR INC.

DEFENSE PRODUCTS DEPARTMENT

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Report ____________________

45 MPH 6,000-POUND AND 10,000-POUND ROUGH TERRAIN FORK LIFT TRUCK FEASIBILITY STUDY

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24 June 1986

Final Report
Contract DAAK70-85-C-0111

Prepared for
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This study was initiated to determine feasibility of developing 6K and 10K Rough Terrain Fork Lift Trucks with a 45 miles per hour road capability. This would allow the RTFLT's to be self deployable, thus eliminating the trailering and towing requirement. Computer simulation was used to analyze the 6K and 10K RTFLT's performance, ride, and handling and to determine the major component requirements. Component suppliers were then contacted to determine component commercial availability and compatibility. Both the 6K and 10K RTFLT's are capable of being developed to go 45 mph without new component technology. The major vehicle components for both the 6K and 10K RTFLT's are available in commercial production today by at least one manufacturer. Minor component design changes requiring a year or less would be required by other manufacturers.

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1 INTRODUCTION

This final technical report, prepared by Caterpillar Inc., for the U.S. Army Belvoir Research & Development Center under Contract DAAK70-85-C-0111, describes the feasibility of developing 6K and 10K Rough Terrain Fork Lift Trucks (RTFLT) with a road speed capability of 45 miles per hour (mph).

Currently, the U.S. Army's RTFLTs have to be trailered or towed to work locations during deployment or relocation. The objective of this contract was to determine the feasibility of developing RTFLTs which are self-deployable and capable of road speeds up to 45 mph without sacrificing any of the performance capabilities during material handling operations on rough terrain. This feasibility was to be determined through an analytical and investigative study that addressed the following:

- Technical problems that would arise due to 45 mph speed requirements on an improved surface;
- Trade-offs, if any, required to meet the 45 mph requirement and their effect on vehicle performance;
- Performance characteristics and specifications for the vehicles' major components;
- Commercial availability and compatibility of those components; and
- Inherent differences in capability to achieve the 45 mph due to machine type and size, (6K versus 10K), and type of steering system, (Ackermann versus articulated).

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2 DISCUSSION

For the purpose of analyzing the feasibility of 45 mile per hour rough terrain fork lift trucks, five somewhat independent categories have been defined. These are the powertrain, suspension, steering system, tires, and other affected systems. Each of these categories required a different analysis approach and are discussed separately in this report.

2.1 Powertrain

One of the first areas that need to be examined in determining the feasibility of rough terrain fork lift trucks with 45 mile per hour capability is the machine's powertrain. Each of the major components in the powertrain must be checked for its ability to handle the speed and torque required during normal material handling operation and high speed roading. Following is a discussion of the approach taken for the analysis of the powertrain and the results of the analysis.

2.1.1 Computer Models

Powertrain performance was evaluated using a number of in-house computer programs run on a mainframe computer and spreadsheet programs created for this job that run on personal computers. The primary computer analysis program used to predict powertrain performance was an in-house engine/torque converter (E/T) matching program. Engine lug curve points (speed and torque) were matched to given torque converter characteristics (speed ratio, torque ratio, primary torque) to compute torque converter output speed and torque. To calculate vehicle performance, specific vehicle and application parameters were input. Input data included engine power rating, transfer gear ratio, bevel gear ratio, final drive ratio, tire rolling radius, transmission gear ratios, mechanical efficiency, and vehicle wind drag loads. The computer program used these inputs to compute rimpull or wheel power versus vehicle speed. Wheel power versus vehicle speed was used to graphically compare various powertrain options.

2.1.2 Operating Conditions

Operating conditions were established based on the current 6K and 10K RTFLT military specifications, the 45 mph RTFLT military specifications, and Caterpillar experience with vehicles of this type. The RFP defined the following specifications used in vehicle performance calculations: vehicle weight, rated load, grade, and speed. Since the RTFLT's will be used for material handling at low speed, it is important for the operator to maintain accurate control of machine speed. If the top speed in first gear is too fast, vehicle speed will be difficult to control when preparing to engage a load. The first gear speeds specified in Table 1 are equivalent to those in commercial vehicles of this type.

Other operating conditions considered included rolling resistance and wind drag. Rolling resistance for roading operation was estimated at 50 pounds per ton of vehicle weight (2.5 percent equivalent grade). This rolling resistance corresponds to operation on a firm, smooth, stabilized roadway surface. During material handling operation, rolling resistance was estimated at 80 pounds per ton of vehicle weight (4.0 percent equivalent grade). This rolling resistance represents a rutted dirt roadway that flexes under load. Wind drag force was calculated by estimating the frontal area of the vehicle and assuming a drag coefficient of 0.7. Wind drag force was included to calculate power requirements.

Commercially available construction type vehicles generally exceed the military specification of 75 hours mean time between failures. For this reason components judged to be commercially successful in applications similar to the 6K and 10K RTFLT were expected to exhibit acceptable component life.

The requirement to travel 40 mph while ascending a 3 percent grade was added by the project engineer that prepared the RFP for this feasibility study to insure that RTFLT performance will meet or exceed other convoy vehicle specifications. The additional specifications discussed above were also agreed upon by the project engineer. All of the feasibility study requirements are shown in Table 1.

Table 1. Operating Conditions Affecting or Affected by 45 mph Capability

Direction	Load	Grade %	Roll Res. %	Total MR* %	Current RTFLT Rqmts.	Feasibility Study
Forward	Rated	45	4.0	45	2 mph	2 mph
Forward	Rated	Level	2.5	2.5	15	15
Forward	None	Level	2.5	2.5	20	45
Forward	None	2	2.5	4.5	-	45
Forward	None	3	2.5	5.5	-	40
Reverse	Rated	Level	2.5	2.5	7.5	7.5
Max Gear Speed - 1st Forward					-	5.0 ±0.5 mph
- 1st Reverse					-	5.0 ±0.5 mph
Max Weight						
-10K Total					37,000 lbs	37,000 lbs
- Max per Axle (air transport)					20,000	20,000
- 6K Total					30,000	30,000
- Max per Axle (air transport)					20,000	20,000
Life, MTBF					75 Hrs	75 Hrs

* Total motion resistance does not include air drag

The current military RTFLT braking requirements are shown in Table 2. These requirements primarily pertain to low speed, off-highway operation.

Table 2. Current Military RTFLT Braking Requirements

	Current RTFLT Requirements		Feasibility Study
Service Brake Deceleration Capability			
Speed mph	Dist ft	Rate ft/s2	
10 Dry	15	7.2	No Change
10 Fade	15	7.2	
10 Wet	25	4.3	
EMERGENCY BRAKING CAPABILITY			
- Dead Engine Stops	6		No Change
- Speed	2 mph		
- Grade	30 %		

Table 3 is a list of some typical on-highway braking requirements for vehicles such as trucks and buses. This list was included for reference in case the military determines additional braking requirements are needed for on-highway RTFLT operation.

Table 3. Typical On-Highway Vehicle Braking Requirements

	Current RTFLT Requirements		Feasibility Study	
Service Brake Deceleration Capability Speed mph	Dist ft	Rate ft/s^2	Dist ft	Rate ft/s^2
20 SAE J992b (Truck,Bus)	-	-	35	12.3
26 SAE J1152 (Constr Eq)	-	-	70	10.4
45 DOT-NHTSA (Veh >10K)	-	-	173	12.6
Service Brake Horsepower Rating SAE J257 HP < 12+1.4*VehWt/1000				
- 6K	-		54 hp	
- 10K	-		64 hp	
Emergency Braking Capability				
SAE J992b (Truck,Bus)	-		5.5 ft/sec^2	
ANSI B56.6 RTFLT	-		35% of normal brake perf.	

2.1.3 Analysis

The powertrain analysis study was conducted to determine the performance required to meet the specified operating conditions. The first step taken was to establish the wheel power requirements. Wheel power is a function of grade, speed, rolling resistance, total vehicle weight, and wind drag.

Input power to other powertrain components can be calculated once wheel power has been calculated. Some general assumptions about the powertrain were made before this was done. First, the mechanical efficiency between the torque converter and the wheels was estimated at 88 percent. Second, torque converter efficiency was estimated at 82 percent unless a lockup converter was used in which case 100 percent converter efficiency was assumed. Third, idle implement pump power and transmission pump power combined was estimated to be 20 hp for both the 6K and 10K RTFLT's. With this information, net engine power was calculated.

A spreadsheet program was created to generate generic engine lug curves. With this spreadsheet a lug curve could be created with the following inputs: rated net engine power, rated engine speed, peak torque speed, and torque rise.

Definitions:

- rated engine power - maximum engine power, can be either net or gross rated engine power.
- gross engine power - power as set on a dynamometer.
- net engine power - gross power minus losses for fan, muffler, air cleaner, and alternator; net engine power is the maximum flywheel power available to propel the vehicle or operate the hydraulics.
- rated engine speed - speed at which maximum engine power occurs.
- torque rise - percent torque increase over engine torque at rated speed.
- peak torque speed - speed at which peak torque occurs, an engine can not be lugged below peak torque speed without stalling

The output data from the spreadsheet program included engine torque and power versus engine speed. This information was required for input to the primary performance analysis program, engine/torque converter matching program. Generic torque converter performance parameters were chosen to represent typical wheel loader torque converters for analysis purposes. Torque converters of this type and capacity are commercially available from a number of suppliers. The appropriate size torque converter was chosen to match the corresponding engine size. A typical wheel loader engine/torque converter match was used.

Computer runs were made for each engine/torque converter combination. Torque converter output speed, output torque, and output power for every point on the corresponding engine lug curve was calculated.

At this point specific transmission and axle combinations could be evaluated. Another spreadsheet program was created to calculate vehicle speed, rimpull, and wheel power based on the following inputs:

- transmission gear ratios
- bevel gear ratio
- final drive ratio
- tire rolling radius
- drive line efficiency

This spreadsheet program was then used to plot wheel power versus vehicle speed to determine how well a particular driveline option meets the operating conditions.

2.1.4 Requirements

Table 4 and Table 5 show the wheel power required to meet the specified operating conditions.

Table 4. 6K RTFLT Wheel Power Requirements

Operating Condition						Wheel Power
Grade (%)	Speed (mph)	RollRes (%)	VehWt (lbs)	Load (lbs)	WindDrg (lbs)	(hp)
0%	-7.5	2.5%	30,000	6000	7	18
0%	15	2.5%	30,000	6000	25	37
45%	2	4.0%	30,000	6000	0	86
0%	20	2.5%	30,000	0	45	42
0%	45	2.5%	30,000	0	227	117
2%	45	2.5%	30,000	0	227	189
3%	40	2.5%	30,000	0	179	195

Table 5. 10K RTFLT Wheel Power Requirements

Operating Condition						Wheel Power
Grade (%)	Speed (mph)	RollRes (%)	VehWt (lbs)	Load (lbs)	WindDrg (lbs)	(hp)
0%	-7.5	2.5%	37,000	10,000	7	24
0%	15	2.5%	37,000	10,000	26	48
45%	2	4.0%	37,000	10,000	0	113
0%	20	2.5%	37,000	0	46	52
0%	45	2.5%	37,000	0	235	139
2%	45	2.5%	37,000	0	235	228
3%	40	2.5%	37,000	0	186	237

Table 6 and Table 7 show the net engine power required corresponding to each specified operating condition. The low speed condition requiring the most power is operation up a 45 percent grade at 2 mph. The roading condition requiring the most power is operation up a 3 percent grade at 40 mph.

Table 6. 6K RTFLT Engine Power Requirements

Assumptions:
- 88% mechanical eff.,
- 82% torque converter eff (converter drv)
- 100% torque converter eff (lock up)
- 20 hp pump power

------Operating Conditions-----				Wheel Power	Net Eng Power TC Drv	Net Eng Power Lock Up
Grade (%)	Speed (mph)	RollRes (%)	Operating Wt. lbs.	(hp)	(hp)	(hp)
0%	-7.5	2.5%	36,000	18	45	41
0%	15	2.5%	36,000	37	71	62
45%	2	4.0%	36,000	86	140	118
0%	20	2.5%	30,000	42	79	68
0%	45	2.5%	30,000	117	182	153
2%	45	2.5%	30,000	189	282	235
3%	40	2.5%	30,000	195	290	242

Table 7. 10K RTFLT Engine Power Requirements

Assumptions:
- 88% mechanical eff.,
- 82% torque converter eff (converter drv)
- 100% torque converter eff (lock up)
- 20 hp pump power

------Operating Conditions-----				Wheel Power	Net Eng Power TC Drv	Net Eng Power Lock Up
Grade (%)	Speed (mph)	RollRes (%)	Operating Wt. lbs.	(hp)	(hp)	(hp)
0%	-7.5	2.5%	47,000	24	53	47
0%	15	2.5%	47,000	48	87	75
45%	2	4.0%	47,000	113	176	148
0%	20	2.5%	37,000	52	92	79
0%	45	2.5%	37,000	139	213	178
2%	45	2.5%	37,000	228	336	279
3%	40	2.5%	37,000	237	348	289

Table 8 is a summary of net engine power ratings used in the E/T matching program.

Table 8. Net engine power lug curves used in E/T Matching program

RTFLT	Application	Net Eng Power
6K	Low Speed Material Handling	150 hp
6K	High Speed Roading (TC drive)	290 hp
6K	High Speed Roading (lock up TC)	245 hp
10K	Low Speed Material Handling	185 hp
10K	High Speed Roading (TC drive)	350 hp
10K	High Speed Roading (lock up TC)	290 hp

A computer run was made for each engine/torque converter combination shown in Table 8.

Figure 1 and Figure 2 are plots of the 6K and 10K RTFLT performance requirements in terms of wheel power versus vehicle speed. Figure 3 through Figure 13 are plots of wheel power versus vehicle speed that can be achieved by all the generic 10K RTFLT powertrain options analyzed. All of the options meet or exceed the wheel power versus speed requirements plotted in Figure 2. Logarithmic plots were used because transmission gear spacing is more easily evaluated. 6K RTFLT powertrain options were not plotted because the conclusions drawn from the 10K RTFLT plots also apply to the 6K RTFLT. The 6K RTFLT power requirements are approximately 20 percent less than the 10K RTFLT powertrain requirements.

The points shown on the plots in Figure 1 and Figure 2 represent the required operating conditions for the 6K RTFLT and 10K RTFLT. "GR" stands for percent grade and "RR" stands for percent rolling resistance. COT is the coefficient of traction or sometimes referred to as pull/weight ratio. The wheel power curves are lines of constant rolling resistance plus grade resistance but with wind drag increasing with vehicle speed. The curve in the upper left corner of each plot is wheel power versus speed when the machine is operating at 0.6 coefficient of traction (approximately the traction limit of the machines off-road). These two plots of the requirements were used to check the feasibility of each powertrain option.

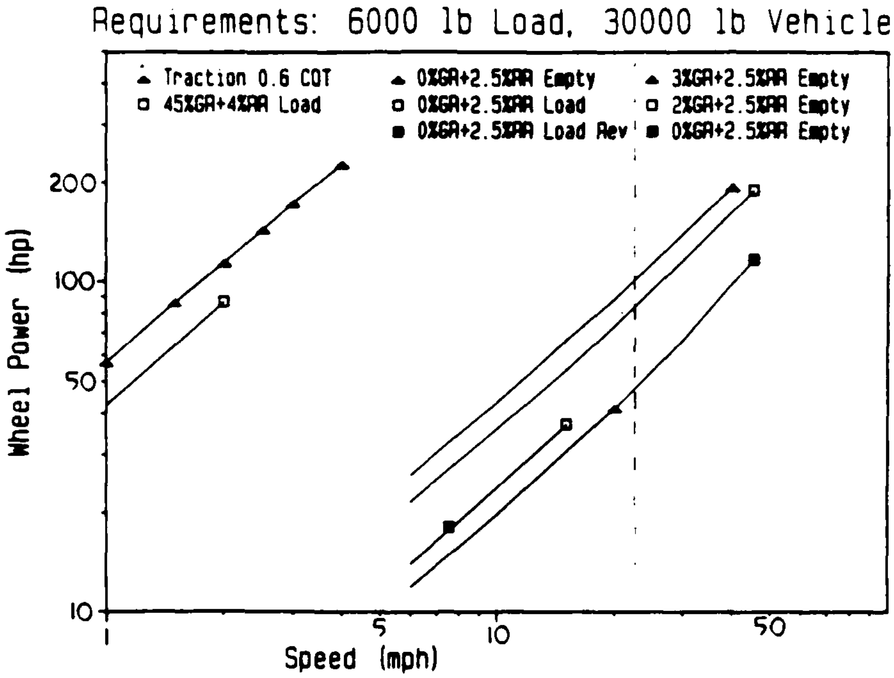


Figure 1. Wheel Power vs Speed for 6k RTFLT Requirements

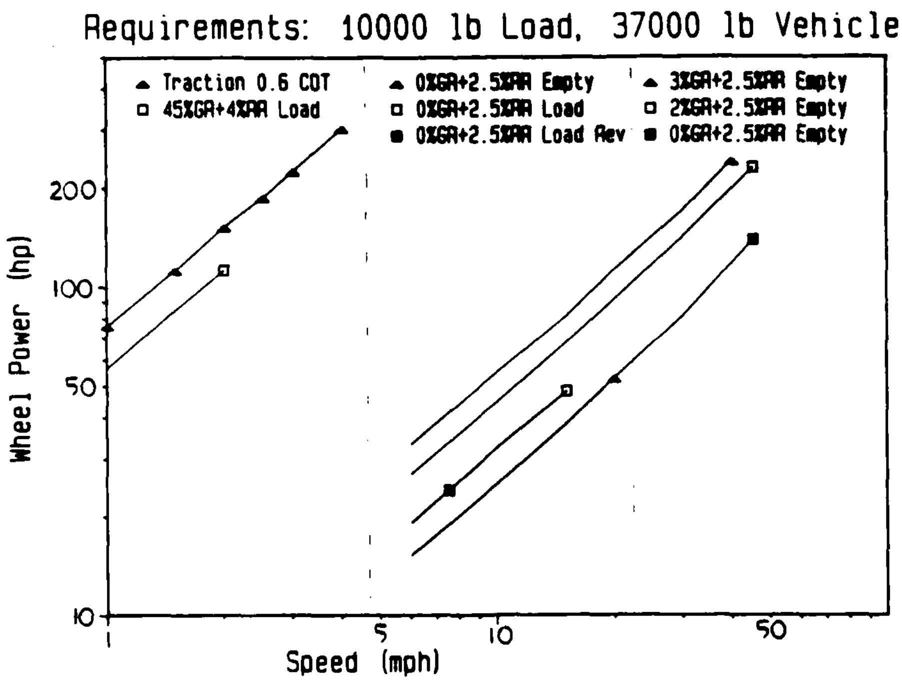


Figure 2. Wheel Power vs Speed for 10k RTFLT Requirements

Figure 3 shows a conventional 3 speed loader transmission with a fixed gear change. Although all performance requirements were met, control of vehicle speed during material handling would not be adequate since 1st gear runout is 10 mph. This option would not be acceptable.

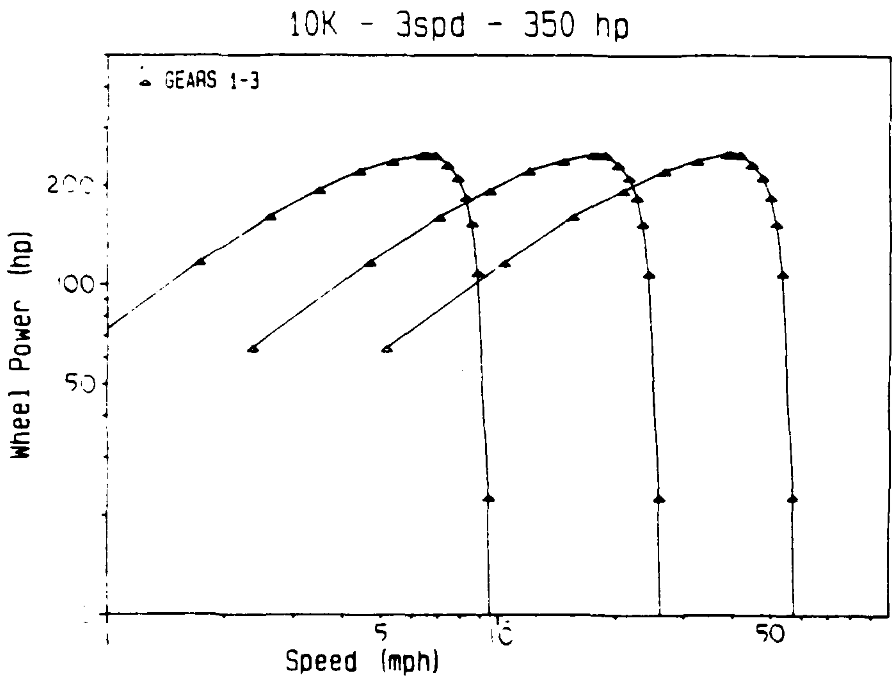


Figure 3. Wheel Power vs Speed for 10k RTFLT
3 Speed Trans with Fixed Gear Change

Figure 4 is a plot of a conventional loader transmission with the gear ratio changed in 3rd gear only. This option provides the desired speeds in 1st gear and 2nd gear for material handling. However, the gear spacing between 2nd gear and 3rd gear is too large to be able to upshift during roading. This transmission would be unacceptable.

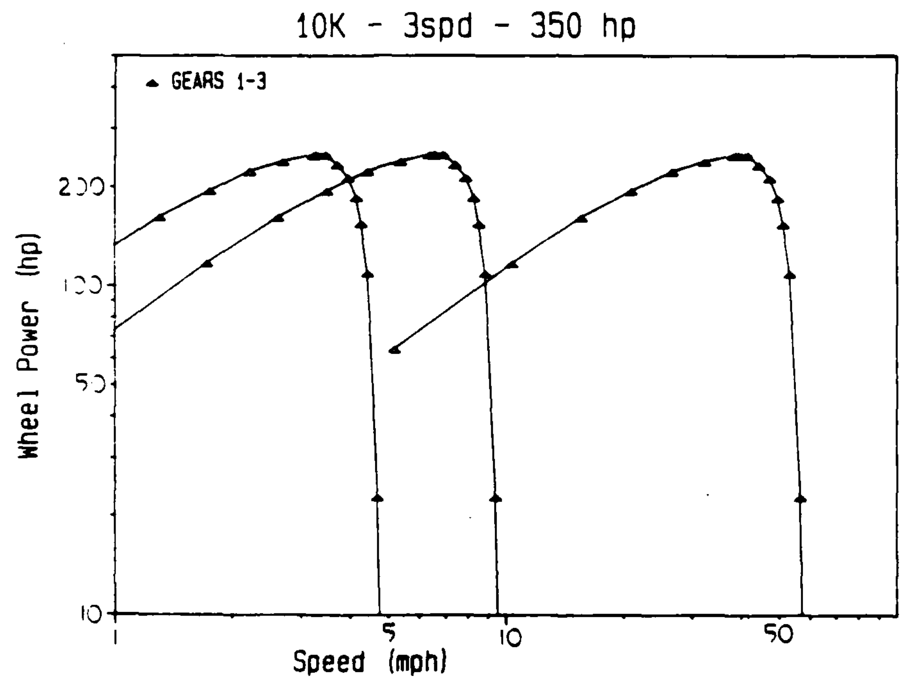


Figure 4. Wheel Power vs Speed for 10k RTFLT 3 Speed Trans with 3rd Gear Ratio Change

Figure 5 assumed the use of a conventional 3 speed loader transmission, however a 2 speed axle (or 2 speed range box) was added to obtain the required roading speeds.

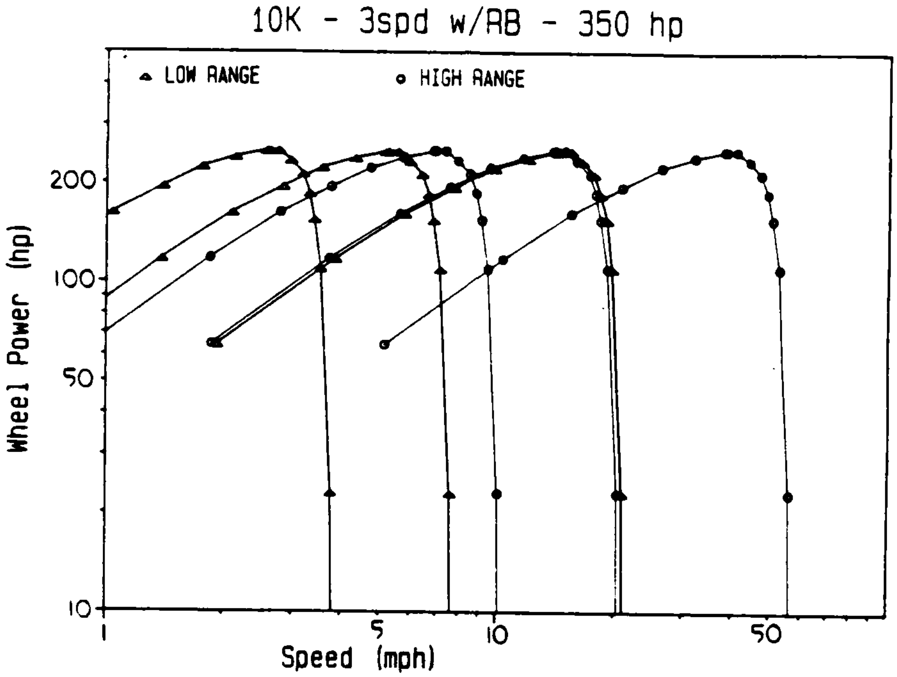


Figure 5. Wheel Power vs Speed for 10k RTFLT 3 Speed Trans with 2 Speed Axle

Figure 6 is a plot of the performance of the same transmission with a lock up torque converter. Note that the gain in powertrain efficiency during lock up operation allows the net engine power to be reduced 60 horsepower. The end result is a six speed transmission where gears 1-3 are used for material handling and gears 4-6 are used during roading. Both of these options would provide acceptable performance. The vehicle may have to be stopped before shifting from low range (gears 1-3) to high range (gears 4-6) depending on the design of the axle or range box.

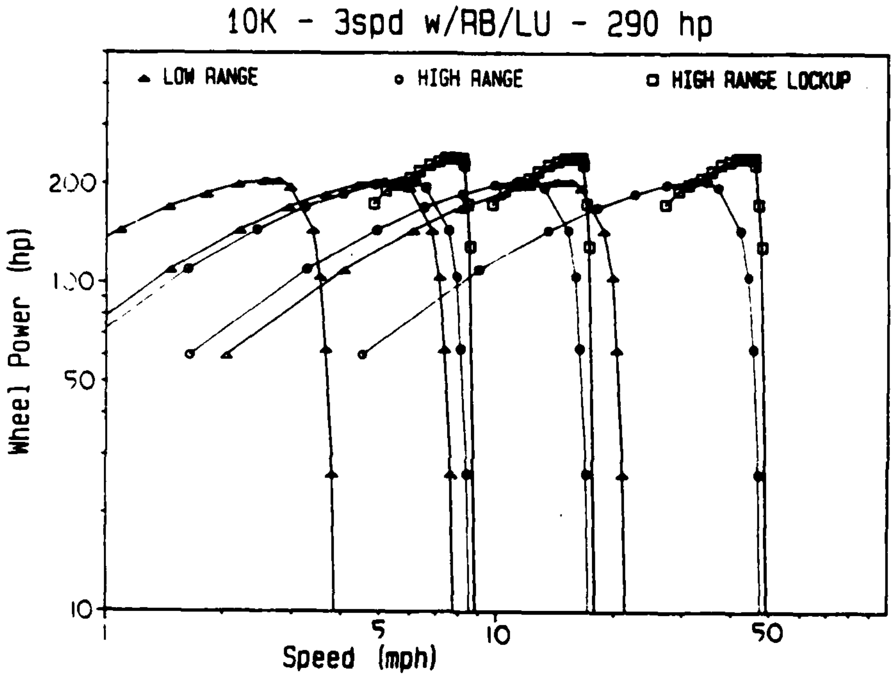


Figure 6. Wheel Power vs Speed for 10k RTFLT
3 Speed Trans with 2 Speed Axle and Lock Up TC

Figure 7 shows the performance of a true six speed transmission (such as a scraper or off-highway truck transmission) not commonly used in wheel loaders.

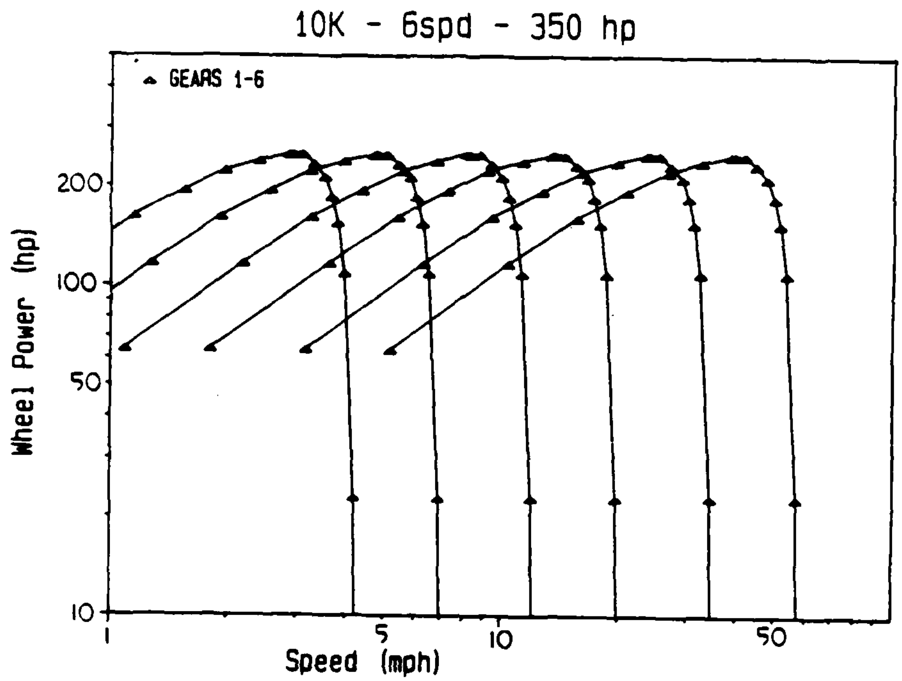


Figure 7. Wheel Power vs Speed for 10k RTFLT 6 Speed Trans

As with the 3 speed transmission shown in Figure 6, Figure 8 shows the performance of the 6 speed transmission when a lock up torque converter is used. The 6 speed transmission with or without lockup would provide excellent RTFLT performance including the ability to shift through all gears without stopping the vehicle. A transmission of this type would be somewhat larger than typical 3 or 4 speed loader transmissions.

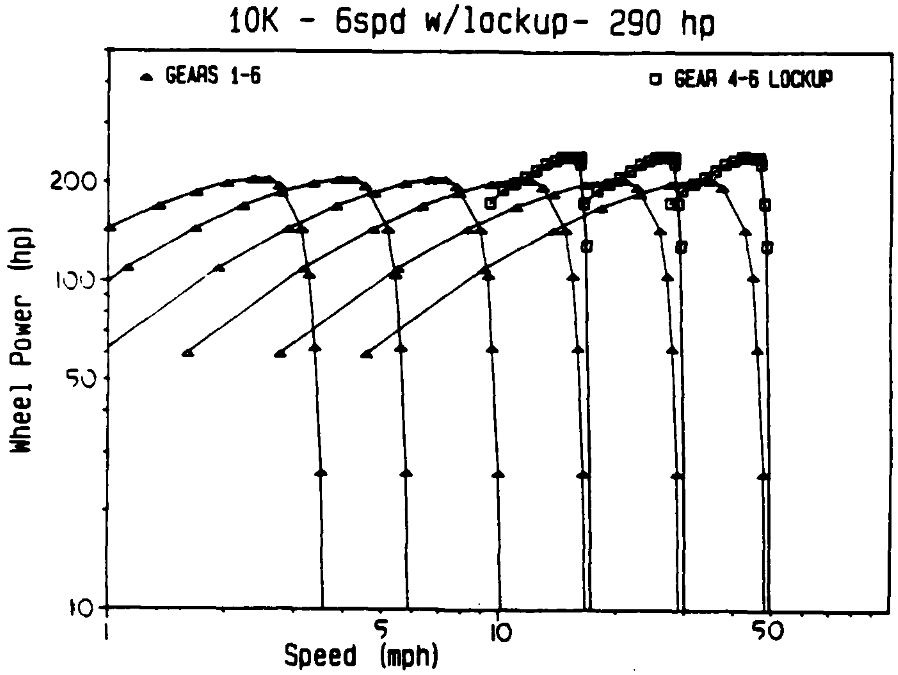


Figure 8. Wheel Power vs Speed for 10k RTFLT 6 Speed Trans with Lock Up TC

Figure 9 is a performance plot of a four speed transmission. The gear ratios were set so that the 2nd gear runout speed was less than 10 mph (to insure adequate control during material handling) and 4th gear speeds would meet or exceed the requirements.

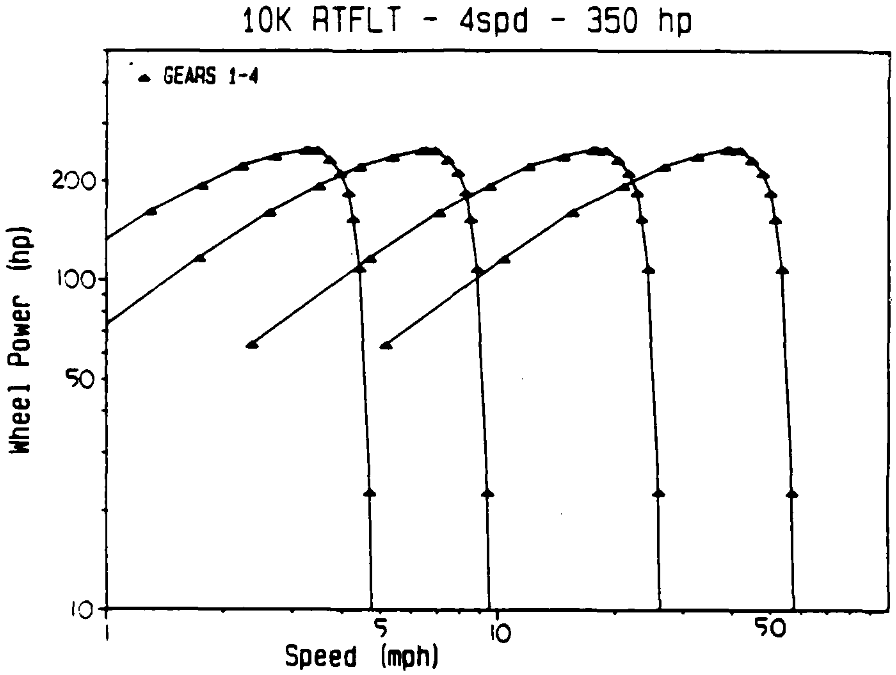


Figure 9. Wheel Power vs Speed for 10k RTFLT 4 Speed Trans with Lock Up TC

Figure 10 is similar to the four speed transmission in Figure 9 with the addition of torque converter lock up in 3rd and 4th gears. The lockup allows the machine to meet the high speed wheel power requirements with 17% less engine power.

Based on Figure 9 and Figure 10, a transmission with a minimum of 4 forward speeds is required. A four speed transmission would provide acceptable performance during all modes of operation.

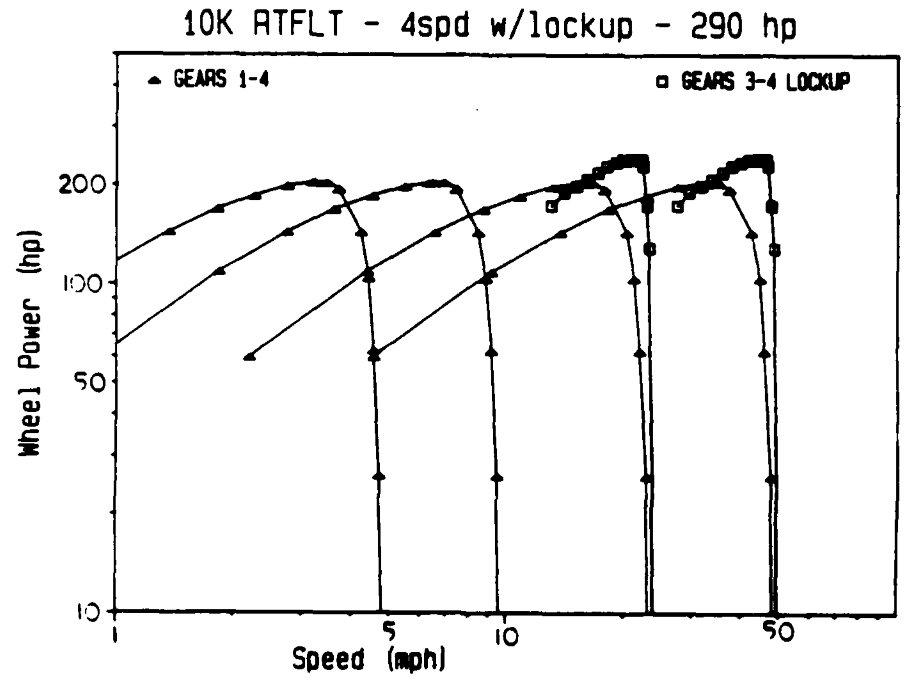


Figure 10. Wheel Power vs Speed for 10k RTFLT 4 Speed Trans with Lock Up TC

Figure 11 and Figure 12 show the resulting wheel power versus speed curves if a dual power engine is used with the 4 speed transmission. The low power setting used in the lower gears was assumed to be same as the M10A military forklift. The high power setting used in the higher gears was matched to the power required to meet the high speed roading requirements.

Use of a dual horsepower engine would allow driveline torques to be limited in the lower gears. For example in Figure 11, the 10K RTFLT requires 210 horsepower to meet the lowspeed performance requirements and 350 horsepower to meet the highspeed performance requirements. If the power were limited to 210 hp in 1st and 2nd gears, substantially smaller driveline components could be used. The additional 140 hp available in 3rd and 4th gears would be used to increase vehicle speed, not the torque in the driveline components. The engine must be sized to match the high power setting and a governor rack stop added to limit power in 1st and 2nd gears. A dual power engine could be used with any transmission option and is recommended for the 6K RTFLT and 10K RTFLT.

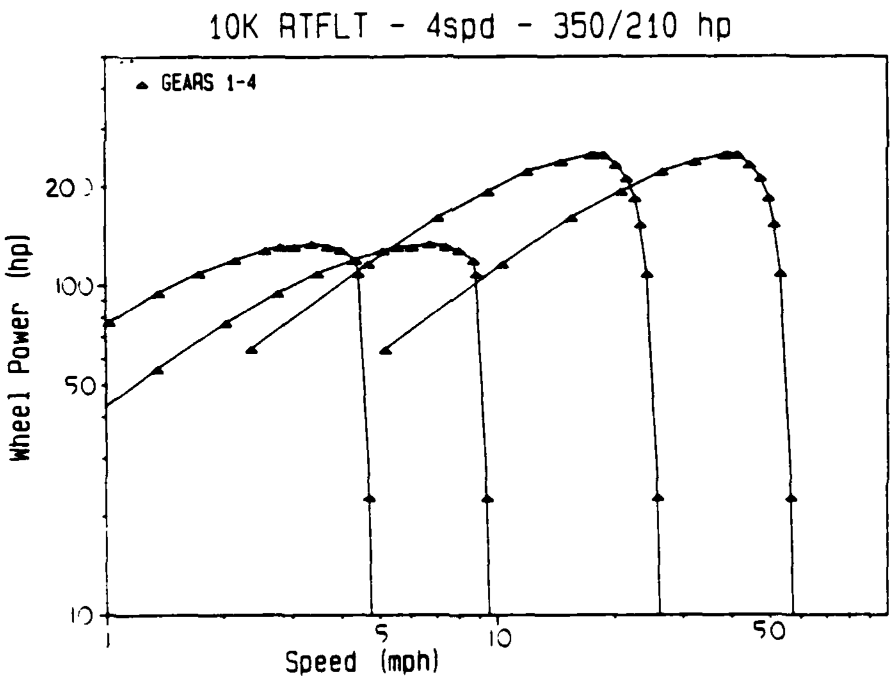


Figure 11. Wheel Power vs Speed for 10k RTFLT
4 Speed Trans and Dual Power Engine

Figure 12 is the same as the dual power engine option shown in Figure 11 with torque converter lock up used. The 17% power saving seen in the upper power number is the same as for the single power options shown in Figures 9 and 10.

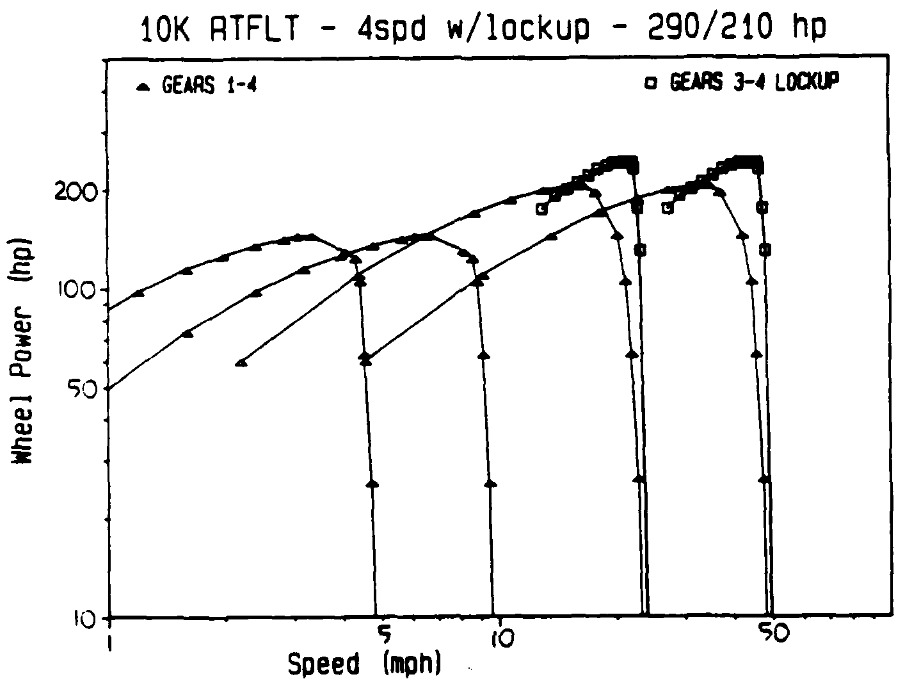


Figure 12. Wheel Power vs Speed for 10k RTFLT 4 Speed Trans with Lock Up TC and Dual Power Engine

A hydrostatic transmission consisting of variable displacement pumps, a two speed gear box or two speed axle, and variable displacement motors was analyzed. Although the performance curves shown in Figure 13 met the requirements, 400 net engine horsepower would be required and large hydrostatic components would be required. This would not be an economical powertrain.

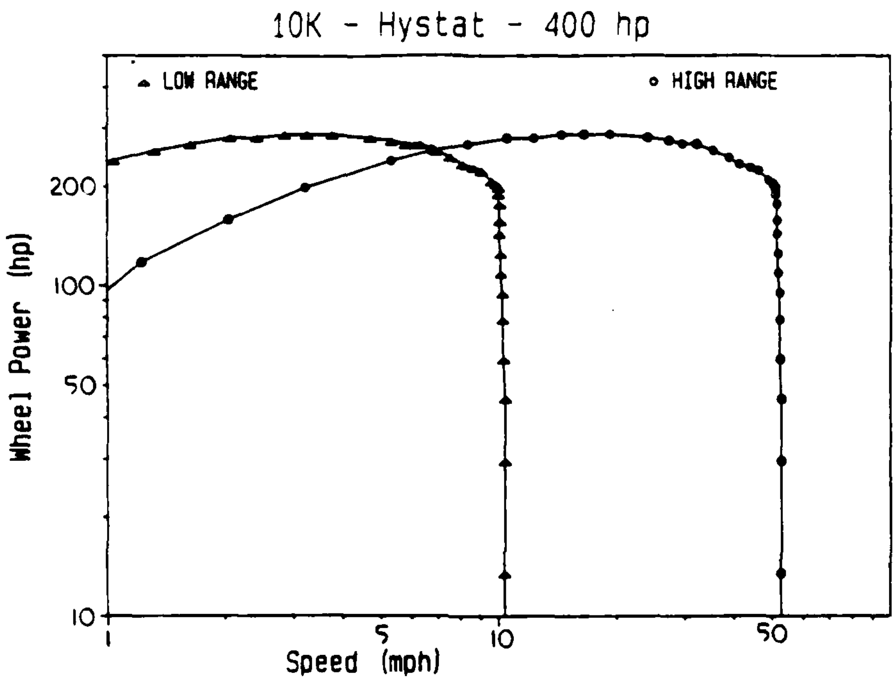
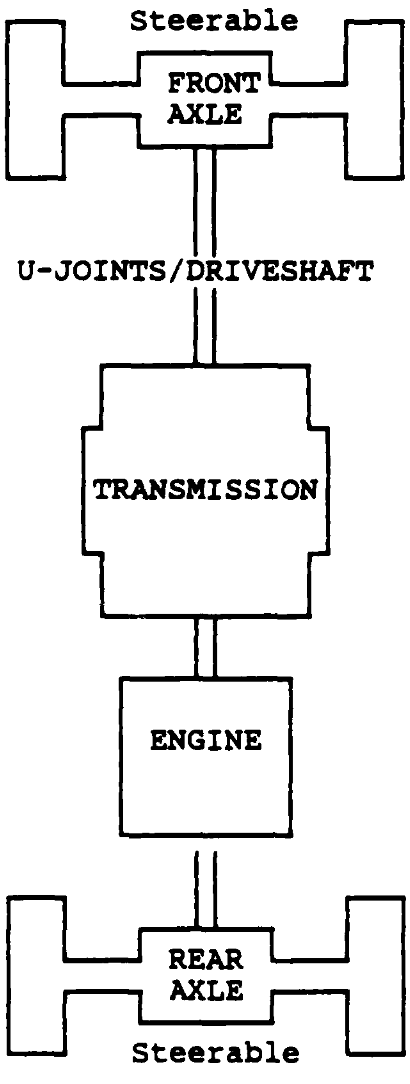


Figure 13. Wheel Power vs Speed for 10k RTFLT Hydrostatic Trans with 2 Speed Axle or Gear Box

Typical component operating requirements or specifications shown on Figure 14 through Figure 17 were developed to assure that some compatability between components can be achieved. These were used as a starting point in contacting suppliers for component availability. Two columns of values are shown. The first column shows conditions in a 1st gear material handling operation (2 mph) with rolling resistance equivalent to 45% grade plus 4% rolling resistance (45% total motion resistance). The second column shows conditions in high gear roading operation (45 mph) with rolling resistance equivalent to 2% grade plus 2.5% rolling resistance (4.5% total motion resistance not including air drag).

Figure 14 below shows the operating requirements or specifications for the 6K RTFLT incorporating one speed axles and a wide range transmission.

	1st Gear		High Gear	
Vehicle Speed	2		45	mph
Equivalent Grade	45		2	%
Rolling Resistance	4		2.5	%
Total Wheel Power	86		189	hp
Tire Rolling Radius		24.5		in
Axle Width/Track Width		95/80		in

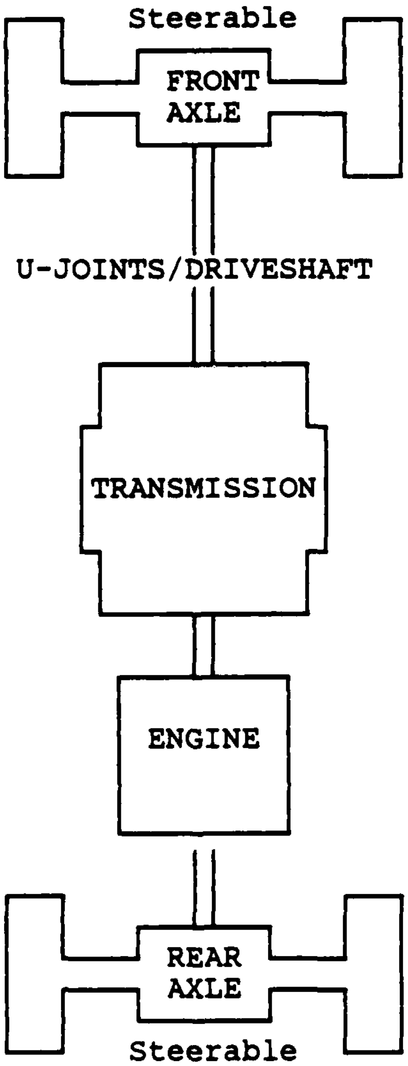


	1st Gear		High Gear	
Front Axle	43		189	hp
	14		309	rpm
	16100		3200	ft-lbs
	12.309		12.309	ratio
	30000		10000	cap lb
		Locking Differential		
U-Joints/Driveshaft	172		3803	rpm
	1300		260	ft-lbs
Transmission		13.21		range
	8.72		0.66	ratio
		6		for spds
		3		rev spds
		Yes		lockup
		5 mph		dir shift spd
Engine			235	net hp
			2500	rpm
Rear Axle			Disconnect	
	43		0	hp
	14		309	rpm
	16100		0	ft-lb
	12.309		-	ratio
	6000		20000	cap lb

Figure 14. 6K RTFLT Powertrain Specification Example
One Speed Axle and Wide Range Transmission

Figure 15 below shows the operating requirements or specifications for the 6K RTFLT incorporating a two speed front axle and either a one or two speed rear axle with a conventional transmission.

	1st Gear		High Gear	
Vehicle Speed	2		45	mph
Equivalent Grade	45		2	%
Rolling Resistance	4		2.5	%
Total Wheel Power	86		189	hp
Tire Rolling Radius		24.5		in
Axle Width/Track Width		95/80		in



	1st Gear		High Gear	
Front Axle (Steerable)	43		189	hp
	14		309	rpm
	16100		3200	ft-lbs
	16.72		12.309	ratio
		1.36		range
	30000		10000	cap lb
	Locking Differential			
U-JOINTS/DRIVESHAFT	234		38	rpm
	960		260	ft-lbs
TRANSMISSION	6.41		0.66	ratio
		9.71		range
		4		for spds
		4		rev spds
		Yes		lockup
		5 mph		dir shift spd
ENGINE			235	net hp
			2500	rpm
Rear Axle (Steerable)			Disconnect	
	43		0	hp
	14		309	rpm
	16100		0	ft-lb
	16.72		-	ratio
	6000		20000	cap lb

Figure 15. 6K RTFLT Powertrain Specification Example Two Speed Axle and Conventional Transmission

Figure 16 below shows the operating requirements or specifications for the 10K RTFLT incorporating a one speed axle and wide range transmission.

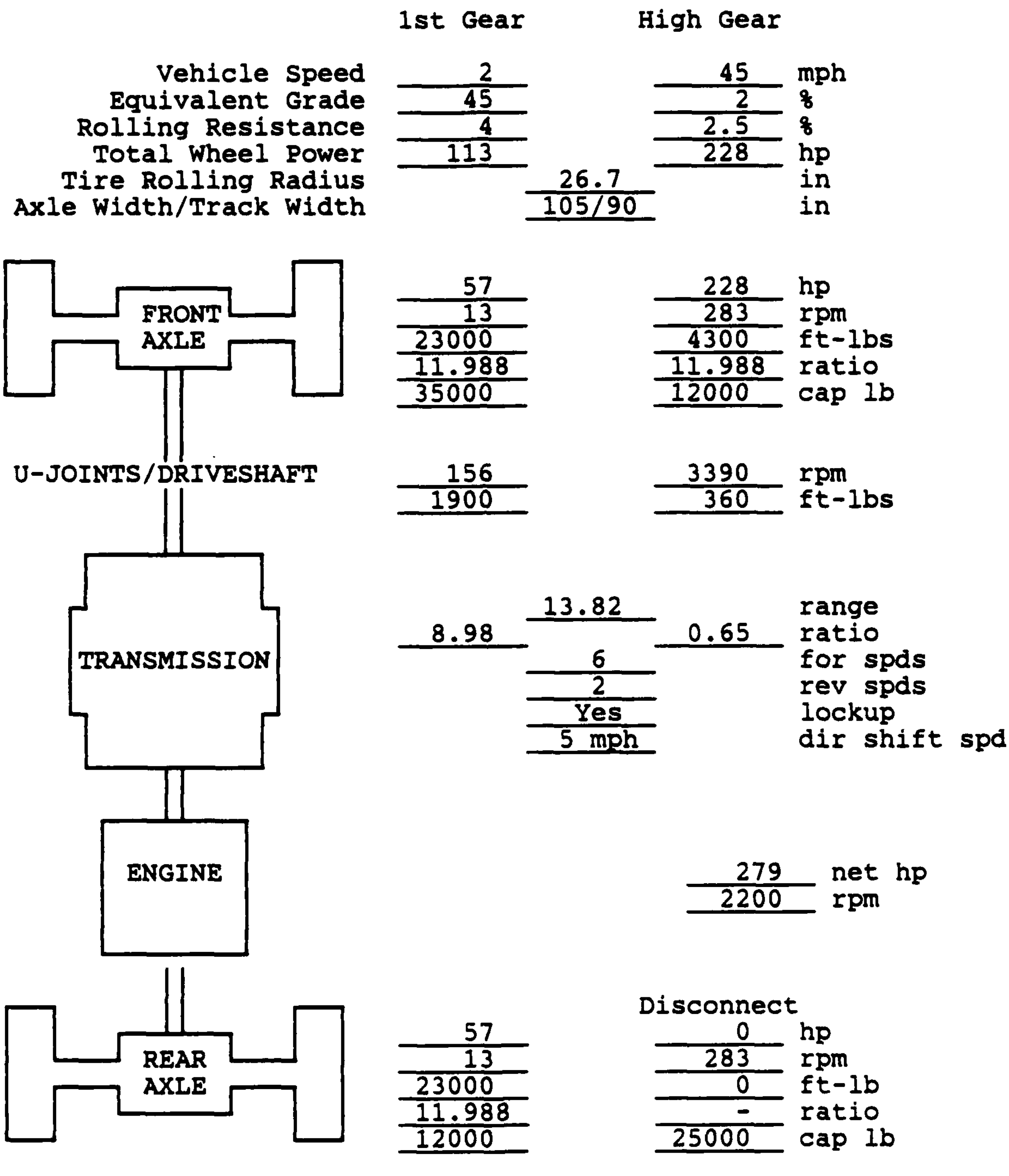
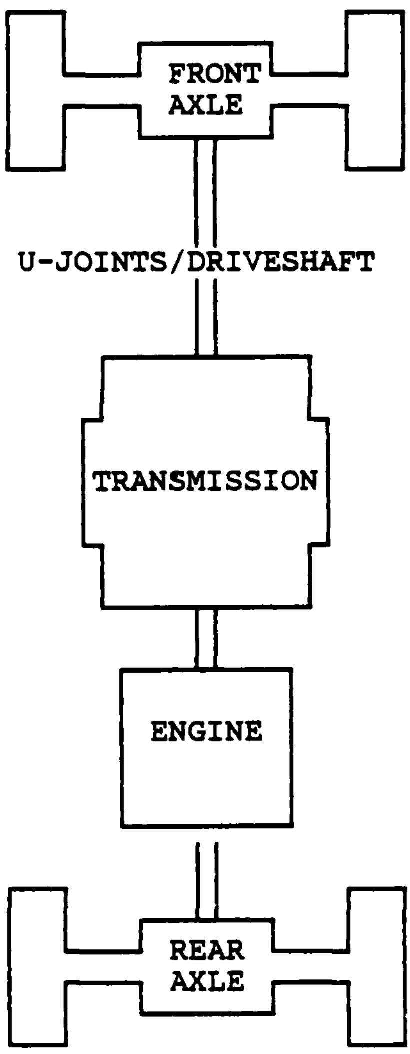


Figure 16. 10K RTFLT Powertrain Specification Example
One Speed Axle and Wide Range Transmission

Figure 17 below shows the operating requirements or specifications for the 10K RTFLT incorporating a two speed front axle and either a one or two speed rear axle and a conventional wheel loader transmission.



	1st Gear		High Gear	
Vehicle Speed	2		45	mph
Equivalent Grade	45		2	%
Rolling Resistance	4		2.5	%
Total Wheel Power	113		228	hp
Tire Rolling Radius		26.7		in
Axle Width/Track Width		105/90		in
FRONT AXLE	57		228	hp
	13		283	rpm
	23000		4300	ft-lbs
	18.30		13.42	ratio
		1.36		range
	35000		12000	cap lb
U-JOINTS/DRIVESHAFT	234		3798	rpm
	1257		320	ft-lbs
TRANSMISSION	5.98		0.58	ratio
		10.31		range
		4		for spds
		4		rev spds
		Yes		lockup
		5 mph		dir shift spd
ENGINE			279	net hp
			2200	rpm
REAR AXLE			Disconnect	
	57		0	hp
	13		283	rpm
	23000		0	ft-lb
	18.30		-	ratio
	12000		25000	cap lb

Figure 17. 6K RTFLT Powertrain Specification Example
Two Speed Axle and Conventional Transmission

2.1.5 Engine

2.1.5.1 Performance characteristics/specifications. The powertrain performance analysis modeled engines with specifications similar to those used in RTFLT's and wheel loaders today. Based on the characteristics of diesel engines commonly used in these applications today, the following specifications are recommended:

- rated engine speed 2000 to 2500 rpm
- torque rise 20 to 30 percent

Torque rise is related to the engine's ability to "hang on or lug" under overload conditions.

Table 9 is a list of the approximate gross and net engine power required to meet the power requirements for low speed and high speed operation.

Table 9. Estimate of Gross Engine Power Requirements

RTFLT	Application	Net Eng Power	Estimated Gross Eng Power*
6K	Low Speed Material Handling	150 hp	160 hp
6K	High Speed Roading (lock up TC)	245 hp	260 hp
10K	Low Speed Material Handling	185 hp	200 hp
10K	High Speed Roading (lock up TC)	290 hp	310 hp

* ≈5% added for fan, muffler, air cleaner, and alternator

2.1.5.2 Feasibility/availability. There are a large number of diesel engine suppliers in the range of power required for the 6K and 10K machines. Caterpillar has a 10.4 liter engine called the 3208 with a production rating up to 250 hp at 2600 rpm. This engine weighs 1450 lb and has a torque rise of 27 percent. The horsepower rating and/or rated speed could be modified to some degree to match up to the other powertrain components. The 3208 engine is an excellent match for the 6K RTFLT. This engine is used in medium range on-highway trucks and several models of Caterpillar construction machines. The Cat 3208 is also used in the Lull 2044 variable reach truck which is very similar to the 6K RTFLT. For the 10K RTFLT, the Caterpillar 3306 diesel engine has production ratings up to 300 hp at 2100 rpm. This rating is for the turbocharged engine with air to air aftercooling. This engine weighs 1975 lb and has a torque rise of 27 percent. Current applications include heavy duty on-highway trucks and more than 20 different Caterpillar construction machines (including wheel loaders).

Detroit Diesel has the 6V-53T-50FP engine available for both the 6K and 10K RTFLT. The Detroit Diesel 8.2 liter engine also is available for the 6K RTFLT. The 8.2 liter engine weighs 1120 lb. A third option for the 6K RTFLT would be the Cummins "C series" engine. This Cummins engine just went into production. Additional engine options for the 10K RTFLT include the Cummins L10 diesel engine. The L10 has a 300 hp rating at 2100 rpm in the turbocharged aftercooled version and weighs 1930 lb.

In general, engine models for the 6K and 10K RTFLT's are commercially available. Most of the suggested engine models from Caterpillar, Cummins, and Detroit Diesel are at their rated power limit at 250 hp in the 6K RTFLT and 300 hp in the 10K RTFLT. Exact power requirements can not be defined until a particular vehicle installation is chosen. Detailed information such as engine parasitic losses (air cleaner, muffler, alternator, fan), implement pumps, transmission pumps, and powertrain efficiency at low and high speed operation is required. Most of these engine ratings can probably be increased if the vehicle application is examined more closely and a dual horsepower governor is used to limit power during lower gear material handling operation. The high power setting would be used in the higher gear roading operation to meet the speed requirements. The engine must be sized to match the high power setting and a governor rack stop added to limit power in the lower gears. A dual power engine could be used with any transmission option and is recommended for the 6K RTFLT and 10K RTFLT. Dual power engines are currently used in motor graders by at least two manufacturers.

2.1.6 Transmission

2.1.6.1 Performance characteristics/specifications. Analysis of the RTFLT performance requirements discussed earlier indicated that to achieve adequate rimpull at the low speed and to achieve the required roading speed the drivetrain will need an overall range in the order of 12 or 13 to 1. Generally, wheel loaders use "soft" torque converters that allow engine speed to remain high. This arrangement provides more responsive hydraulics since implement pump speed is driven by the engine but inefficient roading operation. To reduce power requirements during roading a lock-up torque converter should be incorporated. The lock-up torque converter reduces the required overall drivetrain range to 10 or 11 to 1.

Analysis showed that when using single speed axles or typical 2-speed axles with 35% ratio change, the transmission should have a minimum of four speeds. With lock-up torque converter, the four speed transmission average steps (gear spacing ratio) would range from 1.9 with a 2-speed axle with 35% ratio change to 2.1 with a single speed axle. Although wheel loader applications usually limit the steps to 1.8, in the RTFLT application the 2.1 would probably be adequate. A 6-speed transmission would have average steps of 1.67 with a single speed axle. The 3-speed transmission with lock-up torque converter would have average steps ranging from 2.7 with a 2-speed axle with 35% ratio change to 3.2 with a single speed axle. Both options would be unacceptable.

Therefore to achieve good overall RTFLT performance, only the following options with lock-up torque converter and a minimum of 2 reverse gears should be considered:

- 6-speed transmission (1.7 avg. step) and 1 speed axle
- 4-speed transmission (2.1 avg. step) and 1 speed axle
- 4-speed transmission (1.9 avg. step) and 2 speed axle (1.35 step)

2.1.6.2 Feasibility/availability. Four speed powershift transmissions commonly used in 25,000 pound to 40,000 pound wheel loaders usually have an overall gear ratio range of 5:1 to 8:1 versus the 10:1 required with lock-up torque converter. These commercially available transmissions would have to be connected with a two speed transfer gear box or a two speed axle. Since transmissions of this type are readily available, the following discussion will be limited to the availability of wide range 4 or 6 speed transmissions (that is a transmission that could be used with a conventional one speed axle).

Caterpillar has a transmission currently used in the 615 scraper that has 6 forward speeds and capability for 2 reverse speeds. The overall ratio is 13.5 to 1 and transmission input power for the 615 scraper application is 250 hp.

Clark has recommended their 32000 series transmission (specifically the 13.5HR32654) with 6 forward speeds and 3 reverse speeds for both the 6K and 10K RTFLT. The overall ratio coverage is 9.35 to 1 and the advertised maximum input power is 225 hp. The transmission is designed for off-highway operation. This six speed powershift transmission is available with front and rear axle disconnect.

Funk Manufacturing has a 2000 series transmission with 6 forward speeds and 3 reverse speeds. The overall ratio coverage is 10.25 to 1 and the advertised maximum input power is 225 hp. However, this transmission is being used with a Cummins engine rated at 250 hp in one application. This powershift transmission is also available with an integral engine side axle disconnect or integral interaxle differential. The 2000 series transmission is a good match for the 6K RTFLT. If a dual horsepower arrangement is used on the 10K RTFLT, this transmission may be able to handle 300 hp in the higher gears. Funk Manufacturing has not responded to questions about operation at 200 hp in 1st, 2nd, and 3rd gears and 280 hp in 4th, 5th, and 6th gears.

Twin Disc has an 1130 series powershift transmission rated up to 325 hp. This 1130 is the smallest transmission produced by Twin Disc but it would be too large for the 6K RTFLT. In addition, the current maximum overall ratio is 7.73 to 1 but it could be increased to 8 or 9 to 1 with some development.

2.1.7 Driveshaft/Universal Joints

2.1.7.1 Performance characteristics/specifications. In wheel loaders with a top speed capability of 20 to 25 mph, transmission output speeds are ±20 percent of the maximum engine speed (2000 to 2500 rpm). In most cases, the major portion of the gearing changes required to operate at 45 mph will be done in the axle differential and final drives. Probably only 15 to 30 percent of the required 100 percent speed increase will be obtained in the transmission. Thus universal joints and driveshafts with maximum speed capabilities between 3500 and 4000 rpm will be required for the 45 mph 6K and 10K RTFLT's.

2.1.7.2 Feasibility/availability. The Dana Corporation is a major supplier of universal joints and driveshafts for commercial wheel loaders. Drive shaft speeds are commonly around 2500 rpm. Dana stated that universal joints and drive shafts are normally balanced to 3500 rpm and operation at 4000 rpm should not be a problem.

Rockwell International lists driveline specifications (maximum continuous torque) for 5000 hours of life at 3000 rpm. These universal joints and driveshafts are balanced at speeds above 3000 rpm and Rockwell could supply components for 3500 to 4000 rpm operation.

Universal joints and driveshafts used in production wheel loaders today are generally balanced to speeds around 3500 rpm by suppliers and operation at 4000 rpm is feasible. The 6K RTFLT and 10K RTFLT will not exceed these speeds at 45 mph.

2.1.8 Axles/Brakes

2.1.8.1 Performance characteristics/specifications.

Table 10. Approximate Maximum Axle Operating Loads

RTFLT	Application	Front Axle (lb) Empty	Front Axle (lb) Loaded	Rear Axle (lb) Empty	Rear Axle (lb) Loaded
6K	Low Speed Material Handling	10K	30K	20K	6K
6K	High Speed Roading (45 mph)	10K	-	20K	-
10K	Low Speed Material Handling	12K	35K	25K	12K
10K	High Speed Roading (45 mph)	12K	-	25K	-

Current wheel loader brakes are generally applied quite often during a working cycle, but always at relatively low speeds. Stopping the 6K or 10K RTFLT from 45 mph will generate higher power levels although the frequency of stops from this speed will be very low. Brakes with adequate heat absorbtion capability will be required for high speed stops.

2.1.8.2 Feasibility/availability. Rockwell International has one speed planetary steerable axles available for the 6K RTFLT and one speed planetary non-steerable axles available for the 10K RTFLT. Both of these axles are capable of 45 mph operation. Rockwell has also determined that two speed axles for both RTFLT's could be manufactured using final drive components from planetary axles and two speed differential components from on highway truck axles. Using the off-highway planetary axle housing, some additional machining on the differential housing would allow a two speed differential to be installed. A new two speed planetary axle would require 6 months design time with production requiring another 6 to 12 months. Generally, Rockwell two speed axles have about a 35 percent ratio change.

The Spicer Axle Division of the Dana Corporation does not have any production axles of the size required for the 10K RTFLT. They manufacture a large percentage of the axles used in the current 6K RTFLT market. Spicer evaluated 45 mph planetary axle operation and two speed planetary axles. They concluded one year would be required to design and build a one speed axle for the 6K RTFLT capable of 45 mph operation. This axle would probably require a pressure lubrication system in the final drives. An extensive development program would be required to produce a two speed planetary axle.

Clark Components Company has responded positively with regards to Clark's ability to provide axles for 45 mph operation in the 6K RTFLT and 10K RTFLT. They indicated that this could be done with existing components. They did not indicate axle model numbers, but listed total axle ratio for the 6K at 9.5:1 and 11:1 for the 10K.

Caterpillar Inc. manufactures articulated dump trucks with 35 mph capability. These trucks successfully use planetary non-steerable wheel loader axles similar to the size required for the 6K and 10K RTFLT's. The Caterpillar wheel loader axle design group estimates that some bearing changes in the final drive may be required for 45 mph operation.

Brake options include the following basic types:

- Dry Caliper Disc
- Dry Drum
- Enclosed Wet Disc

All three types of brakes are commonly used on wheel loaders today and available on most axles. The dry type brakes tend to be used on the smaller machines due to their relatively low cost. The enclosed wet type brakes tend to be used on the larger machines where large amounts of heat must be rejected. Dry drum type brakes can absorb more power than most dry disc brakes but are still lower cost than enclosed wet disc brakes. Most dry disc brakes will probably not provide adequate performance for high speed stops. Dry drum brakes are currently used on off-highway trucks that operate at speeds up to 35 mph. Good quality dry drum brakes should be acceptable for the RTFLT's. The desired brake specifications must be evaluated to determine if typical on-highway brake performance is required for the RTFLT's and worth the additional expense if enclosed wet disc brakes are required to meet the specifications.

2.1.9 Component Compatibility

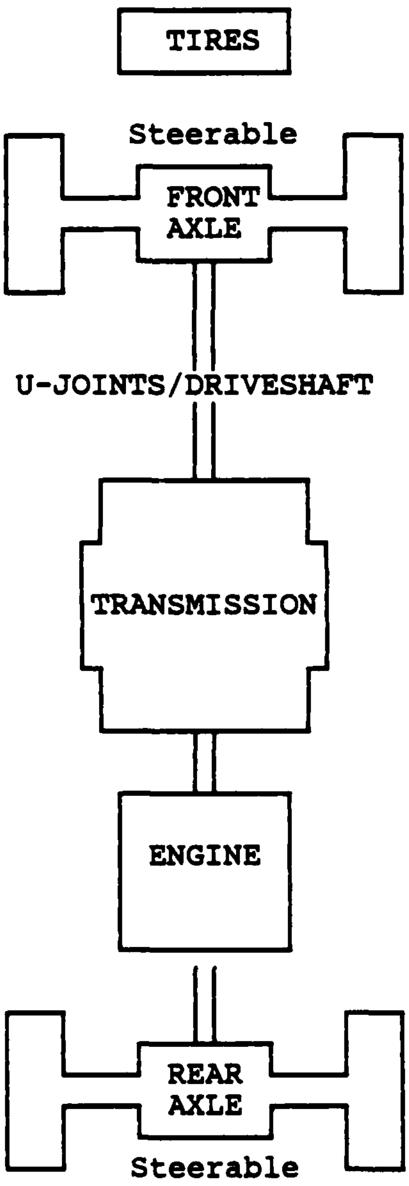
Caterpillar Inc. could build the 10K RTFLT using a modified version of a 950B wheel loader. The required powertrain component modifications could be completed in less than one year. Caterpillar could provide engines and transmissions for the 6K RTFLT.

Clark Components Company was asked to evaluate the availability of Clark powertrain components required to build the high speed 6K and 10K rough terrain forklift trucks. Clark concluded that they can meet the performance specifications using existing Clark components. Their analyses for this project were based on the use of Clark powertrain components and the Detroit Diesel 6V-53T-50FP engine.

Figures 18 through 21 provide a summary of suppliers responses concerning availability of components that meet the performance requirements determined in this feasibility study. The component summaries have been divided into two categories for each of the machine sizes. One is a list of currently available components and the other is a list of components that either will be available within one year or could be manufactured within one year if requested to do so. Combinations of components can be selected from these figures for an in-depth component mating study in the event that demonstration machines are to be built.

Figure 18 below lists the powertrain components that are currently available for use in the 6K rough terrain fork lift truck.

Vehicle speed capability on 45% grade = 2.0 mph
Vehicle speed capability on 2% grade = 45.0 mph



Components Available Today

Make	Model
Goodyear	17.5Rx25 AT2A
Michelin	17.5Rx25 XLB**
Clark	9.5:1 ratio (1 spd) Mod # not specified
Rockwell	PSC-204 (1 spd)
Dana	
Rockwell	
Caterpillar	615 Scraper
Clark	13.5HR32654
Funk	2000 Series
Caterpillar	3208
Cummins	C Series
Detroit Diesel	6V-53T-50FP or 8.2L
Navistar	DTA-466
Clark	9.5:1 ratio (1 spd) Mod # not specified
Rockwell	PSC-204 (1 spd)

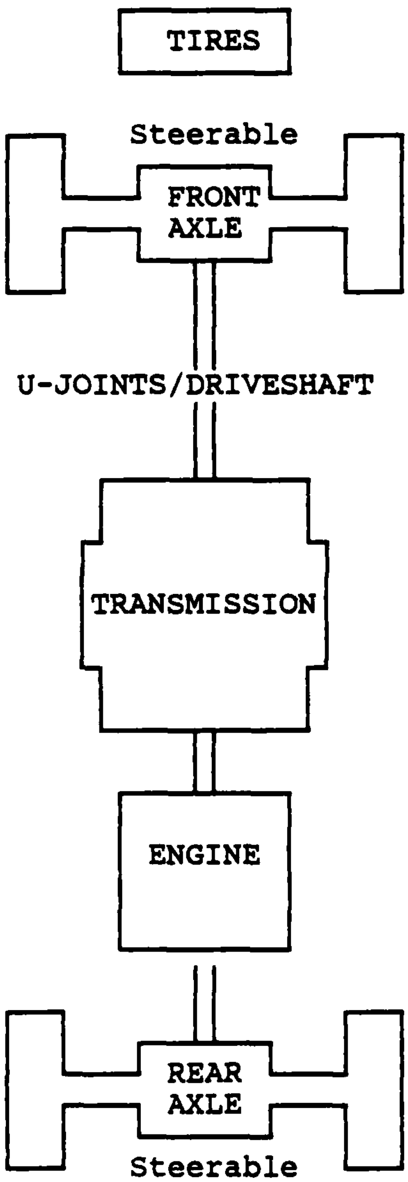
** part of tire designation

Figure 18. 6K RTFLT Current Powertrain Component Compatibility

Figure 19 below lists the additional powertrain components that can be made available for use in the 6K rough terrain fork lift truck within one year.

Vehicle speed capability on 45% grade = 2.0 mph
Vehicle speed capability on 2% grade = 45.0 mph

Additional Components Available in One Year

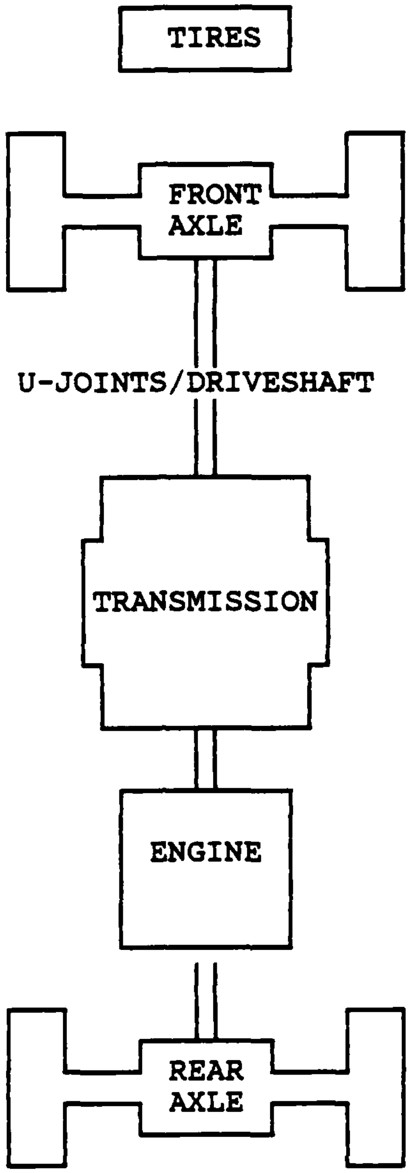


Make	Model	
Rockwell	PSC-204	(2 spd)
Dana	PS1350	(1 spd)
Rockwell	PSC-204	(2 spd)
Dana	PS1350	(1 spd)

Figure 19. 6K RTFLT Future Powertrain Component Compatibility

Figure 20 below lists the powertrain components that are currently available for use in the 10K rough terrain fork lift truck.

Vehicle speed capability on 45% grade = 2.0 mph
Vehicle speed capability on 2% grade = 45.0 mph



Components Available Today

Make	Model
Michelin	20.5Rx25 XLB**
Clark	1 spd, 11.1:1 ratio Mod # not specified
Rockwell	PRC1106 (1 spd)
Dana	
Rockwell	
Caterpillar	615 Scraper
Clark	13HR32654
Caterpillar	3306
Cummins	L10
Detroit Diesel	6-71TA
Clark	1 spd, 11.1:1 ratio Mod # not specified
Rockwell	PRC1106 (1 spd)

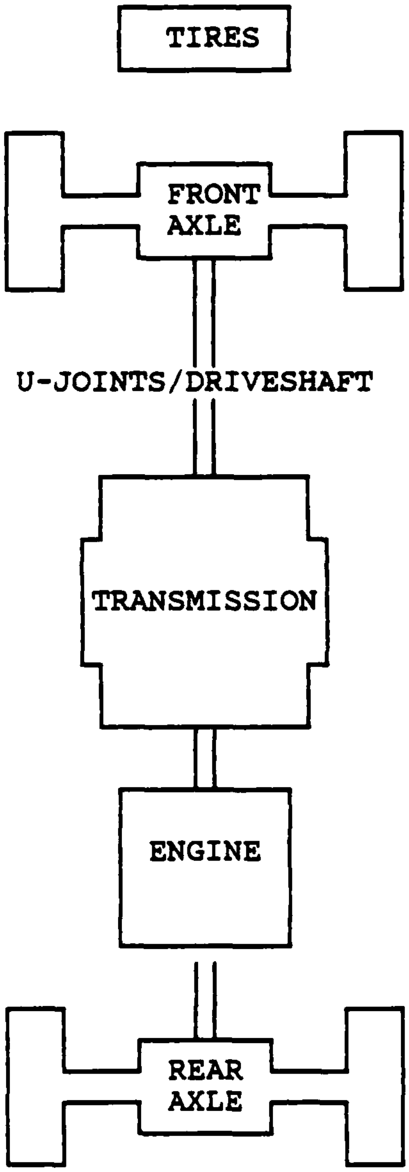
** part of tire designation

Figure 20. 10K RTFLT Current Powertrain Component Compatibility

Figure 21 below lists the additional powertrain components that can be made available for use in the 10K rough terrain fork lift truck within one year.

Vehicle speed capability on 45% grade = 2.0 mph
Vehicle speed capability on 2% grade = 45.0 mph

Additional Components Available in One Year



Make	Model
Goodyear	20.5Rx25 AT2A
Caterpillar	D30C Truck (1 spd)
Rockwell	PRC1106 (2 spd)
Twin Disc	1130
Caterpillar	D30C Truck (1 spd)
Rockwell	PRC1106 (2 spd)

Figure 21. 10K RTFLT Future Powertrain Component Compatibility

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2.2 Suspension

It was the general opinion that some form of vibration reduction device would be required on rough terrain fork lift trucks with 45 mile per hour capability. The vibration reduction device could be either an axle suspension or a suspended mass on the vehicle acting as a tuned harmonic damper. Several alternatives were evaluated for their ability to meet operator comfort and controllability limits at high speeds, encounter large obstacles at low speeds, and perform material handling operations on side slopes. Following is a discussion of the methods used to analyze the suspension options and the results obtained.

2.2.1 Computer Models

2.2.1.1 Ride analysis techniques in general. Ride analysis examines the impact of vehicle dynamic response to road excitations on operator comfort. In addition, some measure of vehicle control (vehicle wheel hop) can be inferred from ride analysis. The dynamic response of a vehicle and, of course operator comfort, are affected by a vehicle suspension. The objective of the rough terrain fork lift truck ride analysis was to determine the suspension characteristics required to provide operator comfort and vehicle control over rough road surfaces.

Several things can excite the response of the vehicle as it is driven over terrain. Some of these are listed below.

- Terrain or road surface roughness.
- Tire radial runout (the variation in tire vertical force as it is rolled on a flat surface at constant rolling radius.)
- Tire unbalance.
- Driveline torsional dynamics that induce vehicle lope.

The first three of these were studied in this work. For this analysis, the fourth was assumed to be alleviated by either disconnecting front or rear axle drive during roading and the addition of a suspension system.

Ride analysis can be performed using time domain analysis or frequency domain analysis. A frequency domain model was used to analyse the response of the vehicle when exposed to road roughness excitations. These excitations consist of a wide range of frequencies with a specific rms amplitude at each frequency. Time domain simulations for this type of analysis are very computer intensive and time consuming. Frequency domain analysis is very efficient whenever the system can be assumed to be linear (e.g. constant stiffness springs and small angle motions). These conditions exist for the analysis of excitations due to road roughness. A time domain model was used for any situations where the linear system assumption would not apply. For example, the linear system assumption can not be made for large obstacle encounters, sideslope stability, and handling analysis.

2.2.1.1.1 Frequency domain analysis. Frequency domain analysis has an advantage for general road surfaces. In frequency domain analysis, the road surface is described statistically. Tire runout and unbalance are described as sinusoidial functions at a frequency corresponding to the wheel rotational speed. Frequency domain analysis results should correspond more closely to measurements taken over extended operating periods than to relatively short operating period measurements. In addition, linear frequency response analysis is very computer time efficient, but cannot handle nonlinear effects. However, most ride phenomena are small displacement/ angle motions and remain relatively linear. Therefore, frequency domain analysis was chosen for most of the rough terrain fork lift truck ride study, although time domain analysis was used to analyze the large displacement motions encountered in obstacle impact response (e.g. traversing an 8 inch bump or 12 inch potholes).

2.2.1.1.2 Time domain analysis. In time domain analysis, the road surface profile (vertical height versus horizontal distance) is specified. Tire force variations due to radial runout and unbalance must be specified as functions of time or wheel angular rotation or speed. Time domain analysis is most advantageous when the road profile is known (e.g. a predefined test track) and the time required to traverse the surface is relatively short (e.g. less than 30 seconds). Time domain analysis is also capable of handling nonlinear effects. Unfortunately, time domain analysis is not well suited for evaluation of ride over long periods of time or on poorly defined road surfaces. Long analysis time periods require excessive computer time to complete the analysis. In addition, poorly defined road surface profiles require an estimation of what the surface will actually look like. The surface profile is also difficult to describe for random road surfaces.

2.2.1.2 Frequency domain models.

2.2.1.2.1 6K RTFLT model. The model components used in the 6K RTFLT ride analysis are shown in Fig. 22. The model is comprised of rigid body components interconnected with linear spring / damper pairs. The model is two dimensional (2D) and includes the pitch mode (rocking chair motion) and bounce mode (up and down motion) of the vehicle. These and other modes of vibration will be discussed more fully in 2.2.3.2. Roll modes and side to side motion were not included in the model.

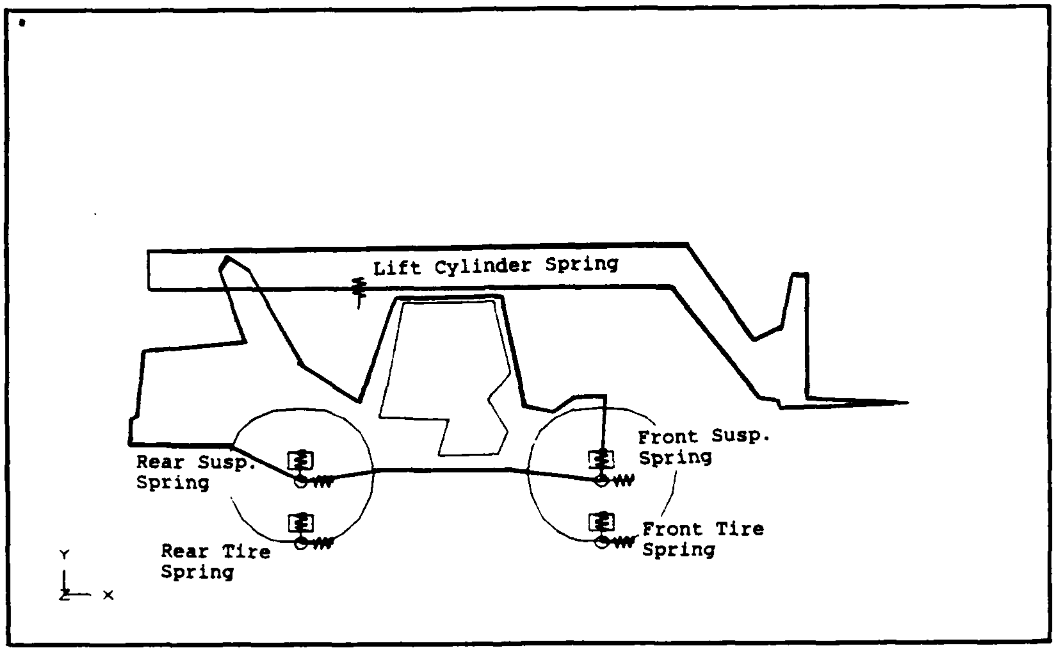


Figure 22. 6K RTFLT Ride Analysis Model

Several model variations were studied. These variations are listed in Table 11. In the unsuspended baseline model variation, only the tire spring / dampers were active. A front axle suspension version has the front suspension spring / damper active in addition to the tire spring / dampers. A third version has both front and rear axle suspensions active in addition to the tire spring / dampers. In the final version, the lift boom assembly is suspended on a hydraulic spring and acts as a harmonic damper. The axles are not suspended in the suspended boom version.

Spring / damper parameter values used in the study are listed in Table 12. These are nominal values. These values were varied over a wide range in the course of the study (See 2.2.3.4). The tire and suspension values are given per axle and not per suspension unit.

Rigid body component parameter values for the 6K RTFLT model are listed in Table 13. Characteristics of the 6K RTFLT include; 30,000 pound total empty vehicle weight, 65% of weight on the rear axle, and a relatively low operator's station (compared to the 10K version).

Table 11. 6K RTFLT - Ride Analysis Model Variations

Spring - Damper Connectors Active

Model Variation	Axle Suspension		Tires		Lift Linkage Cylinder Suspension
	Front	Rear	Front	Rear	
Unsuspended Baseline	NO	NO	YES	YES	NO
Front Axle Suspended	YES	NO	YES	YES	NO
Both Axles Suspended	YES	YES	YES	YES	NO
Suspended Linkage (Lift)	NO	NO	YES	YES	YES

Table 12. 6K RTFLT Model Spring/Damper Parameters

Nominal Values

Connector	Spring Rate Range lb/in	Damping Range lb-sec/in
Tires Front and Rear	4235 / tire 8470 / axle	32 / tire 64 / axle
Axle Suspension Front and Rear	5000 / axle	212 / axle
Lift Linkage Cylinder	20,000	450

Table 13. 6K RTFLT Model Rigid Body Component Parameters

Rigid Body Component	Weight lb	Mass lb-sec^2/in	Inertia* about Z in-lb-sec^2	Center of Gravity** X inches	Y inches
Chassis	18,000	46.63	110,000	10	43
Front Axle and Tires	2,550	6.61	2,500	110	24
Rear Axle and Tires	2,550	6.61	2,500	0	24
Lift Assembly	6,870	17.79	113,000	102	95
Operator's Station	-	-	-	45	55
Total Empty Vehicle	30,000	77.64	427,800	39	52

* Inertia about the component center of gravity
** Measured from the ground line at the rear tire
Wheelbase - 110 inches
Empty weight axle load split front/rear - 35/65%

2.2.1.2.2 10K RTFLT model. The model components used in the 10K RTFLT ride analysis are shown in Fig. 23. The model is similar to the 6K RTFLT model, but contains different parameter data. Table 14 lists the model variations studied in the 10K RTFLT work. The unsuspended baseline, front axle suspension, and both axles suspended variations are similar to those studied in the 6K RTFLT model. However, a suspended rear counterweight used as a harmonic damper was also studied. 10K RTFLT model spring / damper parameter data is listed in Table 15. The tire spring and damping rates are the same as those used in the 6K RTFLT model. The axle suspension spring and damping rates are somewhat lower than those used in the 6K RTFLT model. Rigid body component parameters for the 10K RTFLT are listed in Table 16. Characteristics of the 10K RTFLT include; a larger vehicle weight than the 6K RTFLT, 60% of the weight on the rear axle, and higher operator's station than the 6K RTFLT.

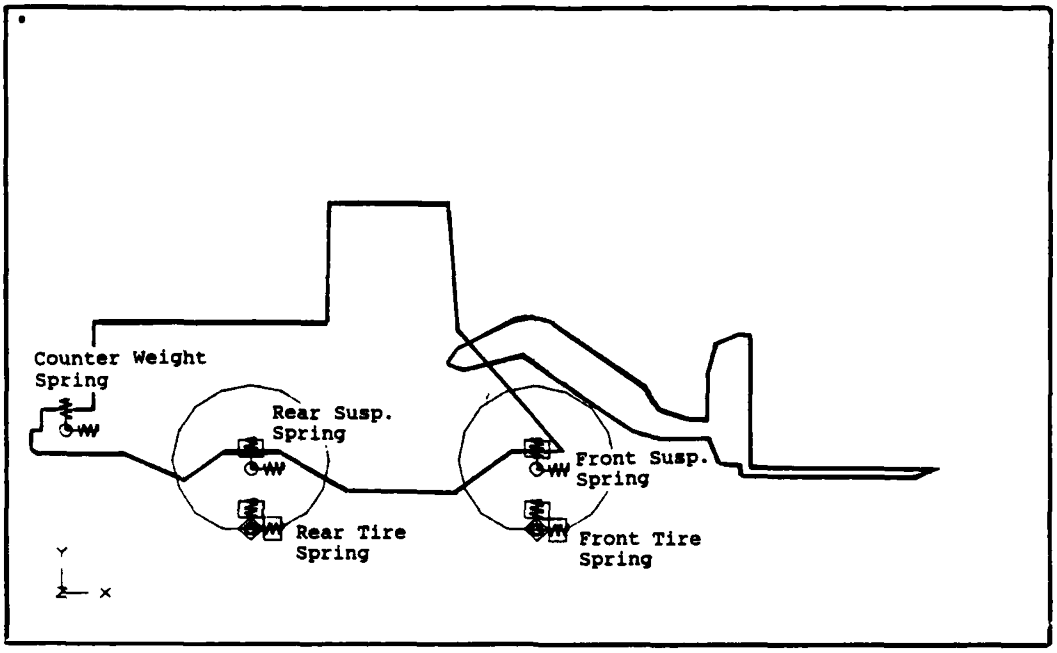


Figure 23. 10K RTFLT Ride Analysis Model

Table 14. 10K RTFLT - Ride Analysis Model Variations

Spring - Damper Connectors Active

Model Variation	Axle Suspension		Tires		Rear Counter Weight Suspension
	Front	Rear	Front	Rear	
Unsuspended Baseline	NO	NO	YES	YES	NO
Front Axle Suspended	YES	NO	YES	YES	NO
Both Axles Suspended	YES	YES	YES	YES	NO
Suspended Rear Counterweight	NO	NO	YES	YES	YES

Table 15. 10K RTFLT Model Spring/Damper Parameters

Nominal Values

Connector	Spring Rate Range lb/in	Damping Range lb-sec/in
Tires Front and Rear	4235 / tire 8470 / axle	32 / tire 64 / axle
Axle Suspension Front Only	2000 / axle	250 / axle
Axle Suspension Front and Rear	4000 / axle	355 / axle
Suspended Rear Counterweight	1,530	88

Table 16. 10K RTFLT Model Rigid Body Component Parameters

Rigid Body Component	Weight lb	Mass lb-sec^2/in	Inertia* about Z in-lb-sec^2	Center of Gravity** X inches	Y inches
Chassis + Lift Link.	20,690	53.60	123,700	62	46
Front Axle and Tires	6,600	17.10	5,340	108	24
Rear Axle and Tires	6,600	17.10	5,340	0	24
Rear Cntrweight	5,000	12.95	838	-69	39
Operator's Station	-	-	-	51	78
Total Vehicle	38,890	100.75	431,192	42	38

* Inertia about the component center of gravity
** Measured from the ground line at the rear tire
Wheelbase - 108 inches
Empty weight axle load split front/rear - 40/60%

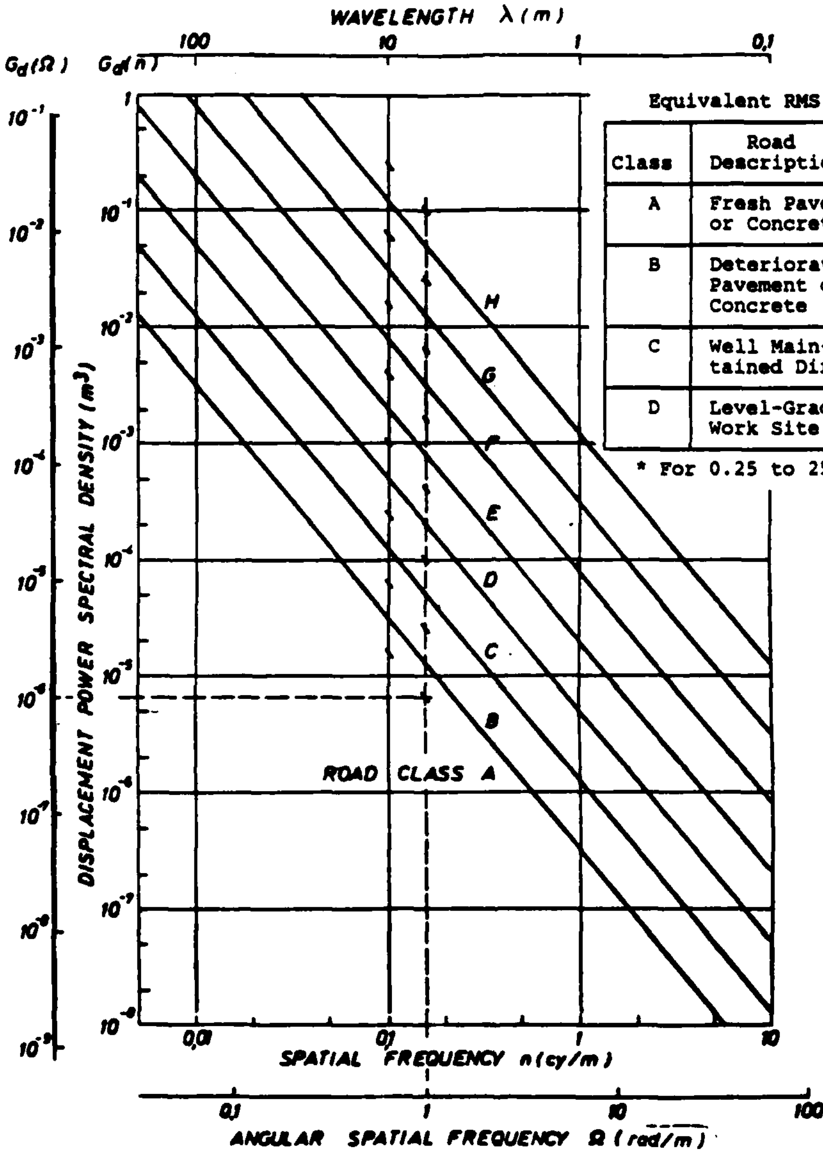
2.2.1.2.3 Road excitation functions. The ISO Road Classification Chart from ISO/DP 8606 is shown in Fig. 24. The axes of the plot are displacement Power Spectral Density (PSD) in meters cubed versus spatial frequency in cycles/meter (1/wavelength). The bands are recommended for classifying measured road surfaces and vary from very smooth to very rough surfaces. By definition, the area under the displacement PSD curve between any two spatial frequencies is the mean squared displacement of the road surface in that frequency range. Therefore, the RMS (Root Mean Square) displacement of a surface for a frequency range is the square root of the area under the displacement PSD curve. Overall equivalent RMS values and descriptions for four road surfaces are also included on Fig. 24.

The vehicle models require road excitation functions of the form shown in Fig. 25 for the front tire and Fig. 26 for the rear tire. The RMS amplitude for four different road and vehicle speed combinations are plotted versus time frequency in Hertz (cycles/sec). The RMS amplitude at each time frequency is calculated from the area under the PSD curve for that particular road classification band and for a small frequency bandwidth of spatial frequency (road wavelength). The time frequency is calculated from the road surface wavelength (1 / spatial frequency) and the vehicle velocity, viz.

Frequency = Velocity / Wavelength

where:
Frequency = Time frequency (cycles/second)
Velocity = Vehicle Velocity (meters/second)
Wavelength = Road Surface Wavelength (meters)

The phase relationship between the front and rear tires is calculated from the road surface wavelength and the wheelbase of the vehicle. Therefore, the road excitation functions are dependent upon the road surface classification from Fig. 24, the vehicle velocity, and the vehicle wheelbase.



Equivalent RMS by Road Class

Class	Road Description	Overall RMS * meters	inches
A	Fresh Paved or Concrete	0.002	0.08
B	Deteriorated Pavement or Concrete	0.004	0.16
C	Well Maintained Dirt	0.008	0.31
D	Level-Graded Work Site	0.016	0.63

* For 0.25 to 25 meter wavelength

Figure 24. ISO Road Classification Chart

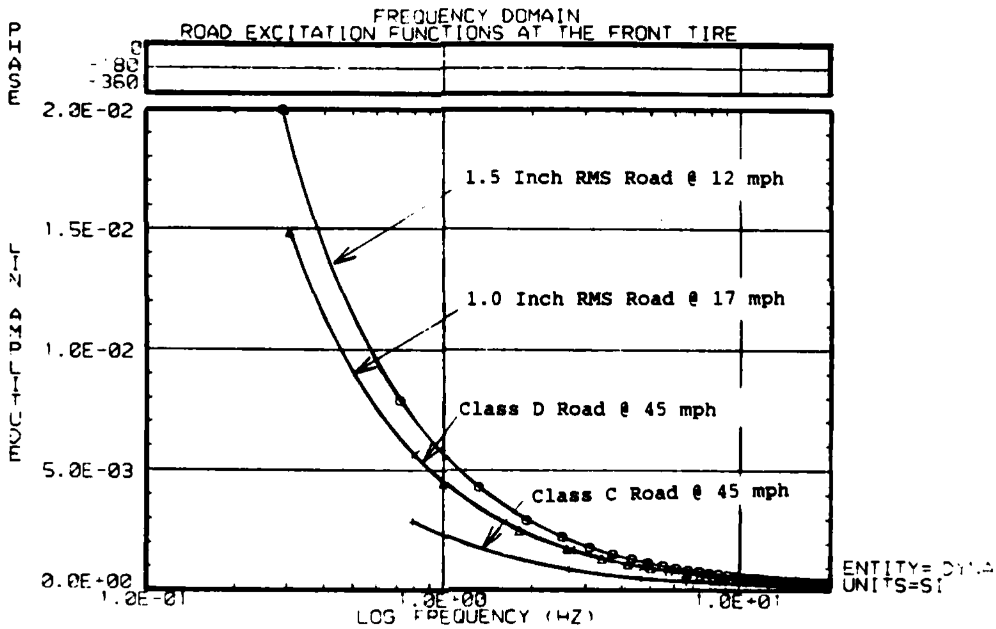


Figure 25. Road Excitation Functions at Front Tire

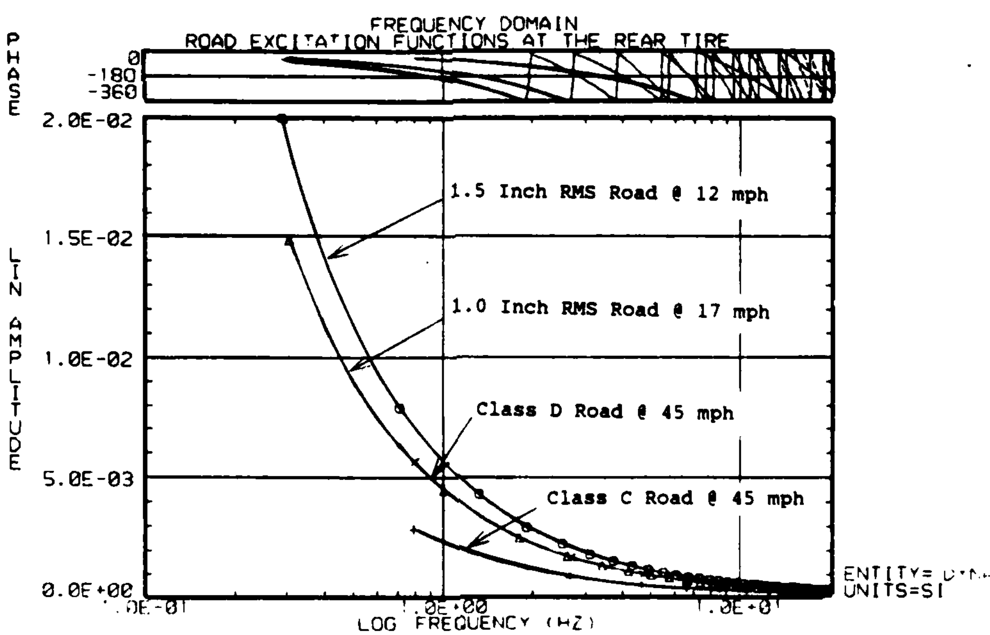


Figure 26. Road Excitation Functions at Rear Tire

2.2.1.2.4 Operator absorbed power. Operator absorbed power is a commonly used measure of ride comfort. Power is actually force times velocity, but can be calculated from RMS acceleration, frequency, and operator mass through basic relationships and differential equations. These relationships have been evaluated and combined with weighting factors to account for the sensitivity of the human body to vibration at various frequencies. A formula has been determined for calculating operator absorbed power from RMS acceleration versus frequency response functions. The formula takes the form;

$$\text{Absorbed Power} = \Sigma\ (\ K * \text{RMS Acceleration}\hat{\ }2)$$

where K is the frequency dependent weighting factor. The summation is performed at each frequency in the frequency spectrum. These weighting factors are shown on Fig. 27 for various directions of motion. Note that human sensitivity is very dependent upon frequency. Humans are very sensitive to fore and aft vibration between 1 and 2 Hertz, which as will be shown later is a very common frequency range for RTFLTs. Humans are less sensitive to vibration in the vertical direction, but this sensitivity covers a wider frequency range from about 2 to 7 Hertz. This frequency range is again common in RTFLTs. The feet contribute little to total absorbed power as shown by the low value of the foot factor curve on Fig. 27. Humans are very sensitive to side to side vibration. However, side to side motion was not included in the two dimensional ride models. In addition, side to side motion should not be commonly encountered for extended periods when operating on road surfaces.

A more complete discussion of the procedure and the K versus frequency relationship can be found in SAE paper 680091, "Analytical Analysis of Human Vibration", R.A. Lee and F. Pradko.

2.2.1.2.5 Wheel Hop (Controllability). The percent of time that a tire is off the ground is a measure of vehicle control (wheel hop) characteristics on randomly rough surfaces. The RTFLT ride analysis models predict tire RMS displacement versus frequency in response to the road excitation functions. The net RMS tire deflections over the entire frequency range of the road excitation was used to calculate the probability that a tire is off the ground. A standard statistical table of areas under standardized normal density function was used with the ratio of static tire deflection to tire RMS deflection to calculate percent time the tires were off the ground for a given road excitation.

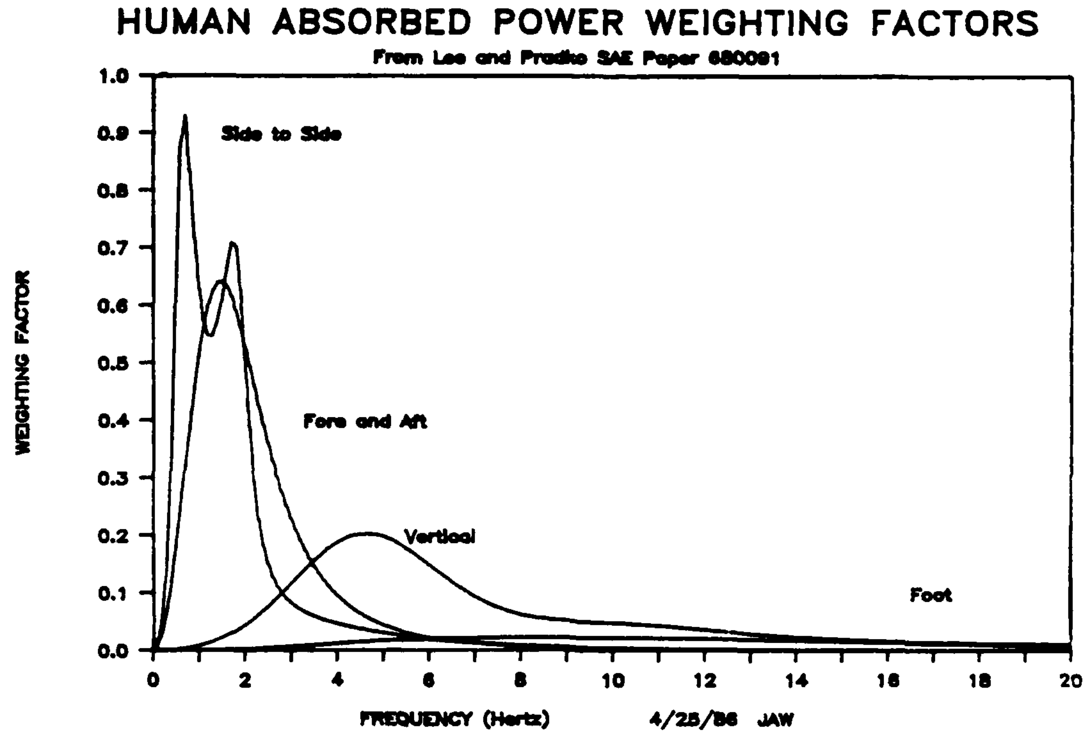


Figure 27. Human Absorbed Power Weighting Factors

2.2.1.3 Time domain models. A 3-D linkage analysis program was used to assemble models of the 6K and 10K rough terrain fork lift trucks. This model was required to examine the large motion dynamics of the vehicle that could not be simulated by the 2-dimensional ride analysis model. The 3-D models were created to examine the response of the vehicle and suspension systems to large obstacle encounters, side slope stability, material handling stability, and high speed controllability.

The 3-D analysis computer program used allows the analyst to describe the geometry and mass properties for a system of linkages and several types of joints. These features were used to describe the vehicle geometry, suspension system, and steering linkage. Subroutines were written and added to the ADAMS program to model the tires used on these vehicles. The tire subroutine was needed to calculate tire vertical, lateral, and traction forces.

The 3-D model was set up to handle a variety of suspension and steering systems. The suspension configurations that can be used for any axle include rigid mounted axle, oscillating axle, suspended axle with vertical motion only, suspended axle with vertical and roll motion, and independent left and right suspension arms. These options are shown in Fig. 28.

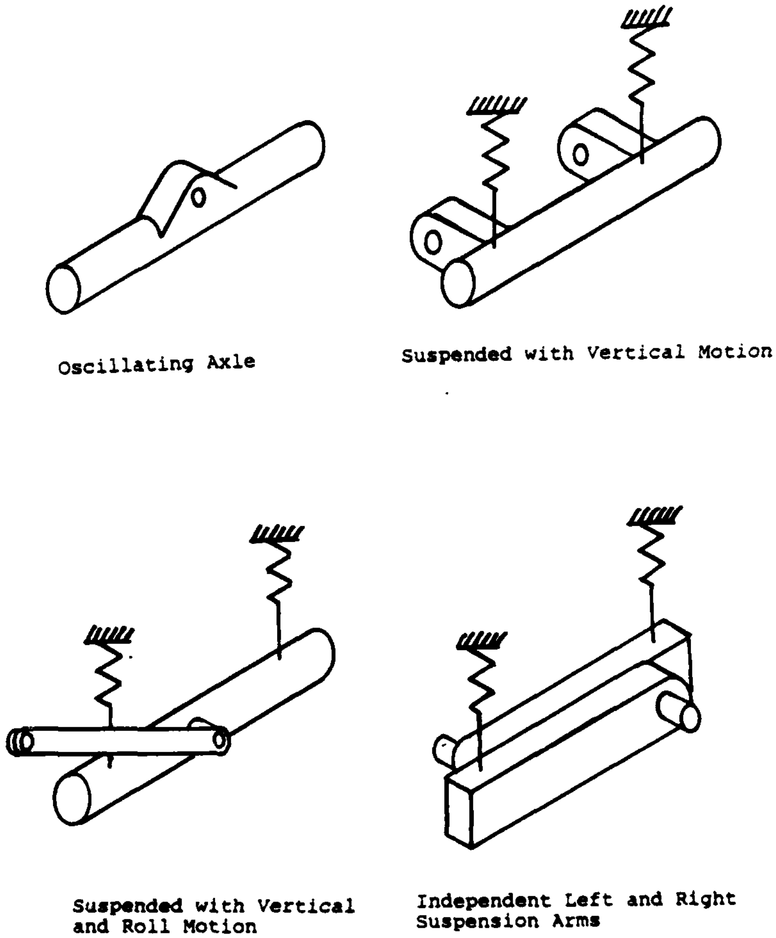


Figure 28. 3-D Dynamic Model Suspension Options

The steering system configurations that can be used include articulated frame, front axle Ackermann, and/or rear axle Ackermann. These options are shown in Fig. 29. A vehicle can be studied as it responds to any desired ground profile, slope, external load, and steering wheel input.

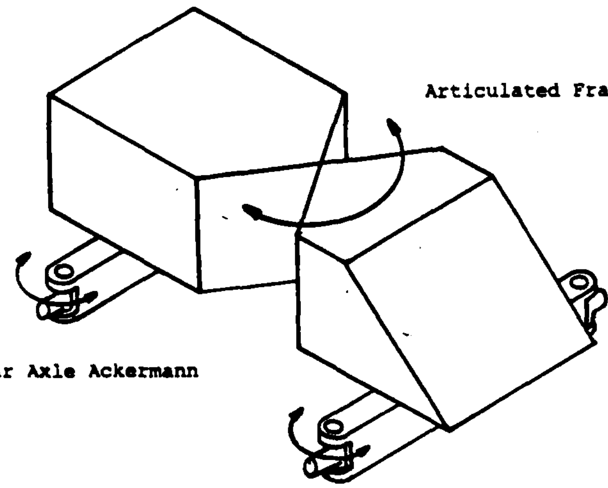


Figure 29. 3-D Dynamic Model Steering Linkage Options

The tire model is required to predict the forces produced by the tire/ground interaction. These forces are the basic items that determine vehicle dynamic response. There are three components to the tire forces that must be predicted. Tire vertical force is determined by the tire deflection, tire stiffness, and tire damping. The tire stiffness is usually modeled by a single spring between the rim and the ground which is adequate for studying response on fairly flat terrain. This study included analysis of response to 8" radius speed bumps and 12" deep potholes which are very sudden terrain changes and could not be handled accurately by a single spring tire model. The tire model was therefore developed using many evenly spaced springs that extend radially outward from the center of the wheel. Tire lateral force is determined by tire vertical force, tire slip angle, and tire cornering stiffness. Tire longitudinal force is determined by tire vertical force, fore-aft slip, and the tire pull-slip curve. Fig. 28 gives a representation of the tire model and the relationships for lateral and traction forces. The cornering stiffness and pull-slip data comes from tests of a specific tire on a specific surface. Data on the specific tires to be used on these vehicles was not available and would require an extensive test program. The information used in the simulations came from existing test data on the most similar tire that could be found. This was for a Goodyear 10.00-20 SHM tested on a paved surface.

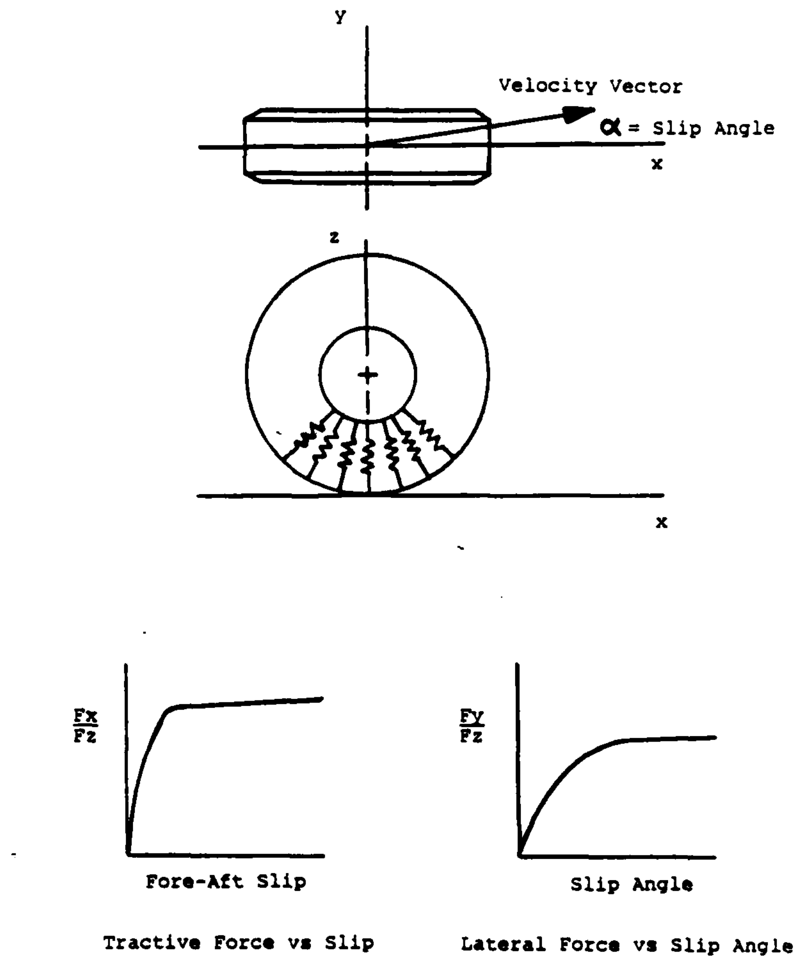


Figure 30. Vehicle Tire Model

Definitions:

- Tire stiffness or spring rate is the change in vertical force divided by the change in tire deflection (Lb/in).
- Tire damping coefficient is the change in vertical force divided by the rate of change in tire deflection (Lb per in/sec).
- Tire slip angle is the difference between the tire longitudinal or fore-aft axis direction and the direction of tire motion (degrees).
- Tire cornering stiffness is the change in lateral force divided by the change in slip angle (Lb/degree).
- Tire fore-aft slip is defined as 1 - the ratio of the actual forward speed of the tire to the ideal forward speed due to tire rotation.
- The pull-slip curve is the relationship of tire fore-aft force to tire fore-aft slip (Lb/percent slip).

2.2.2 Operating Conditions and Requirements

A set of operating conditions and acceptance criteria were established to evaluate the feasibility of traveling at 45 miles per hour. Some of these were presented in the RFP and some resulted from discussions with the Fort Belvoir project engineer. Any specifications that are in the current military specification which might be affected by the addition of 45 mile per hour capability were also examined in the study. Following is a summary of those that relate to the analysis of the suspension options for the vehicles.

The ride comfort related requirements used in this feasibility study for the RTFLTs are listed in Table 17. These requirements reflect the desire to travel at 45 mph on improved road surfaces. Total operator absorbed power should not exceed 6 watts when traveling at 45 miles per hour on a class C road. Class C is a road roughness category defined in ISO/DP 8608 which has a surface roughness of .31 inches rms. This is the approximate roughness of a well maintained dirt road. The 6 watt limit on the class D road is not required, but is desirable.

The vehicle should be controllable on a class D surface (.62 inch rms) at 45 miles per hour. Since no specific guidelines were given on the measure of controllability, a measure using percent time the tires are off the ground (wheel hop) was used. Measured and predicted data for wheel loaders, which are very similar to the 10K RTFLT are shown in Table 18. The wheel hop guidelines used were:

- Acceptable - wheels off the ground less than 5% of the time
- Marginal - wheels off the ground between 5 and 10% of the time
- Unacceptable - wheels off the ground more than 10% of the time.

Peak acceleration at the operator station should not exceed 2.5 g's when traversing an 8" radius speed bump at 7 miles per hour.

Current static and dynamic stability specifications (Table 19) should apply to the vehicle when modified to travel 45 miles per hour. These specifications include lifting 1.5 times the rated load, braking down a grade with rated load, and performing a maximum steer angle turn on a side slope with and without rated load.

Table 17. 6K and 10K RTFLT Ride Requirements

Requirement	Current RTFLT Requirements	Feasibility Study
Operator Total Absorbed Max Power Limit = 6 Watts Road (RMS inches) Class C (0.3) - Required Class D (0.6) - Desired	 - -	 45 mph 45 mph
Controllable on: Road RMS (inches) 0.3 (Class C) 0.6 (Class D)	 - -	 45 mph 45 mph
Operator Max Vertical Acceleration = 2.5 G's Hitting 8 Inch Radius Bump	-	7 mph
Rear Axle Oscillation 10K Vertical Wheel Travel 6K Vertical Wheel Travel	 +/- 7 inches +/- 6 inches	Dependent upon Suspension Option

Table 18. Wheel Loader Ride and Controllability

Measured and Predicted Data

Wheel Loader	Total Operator Absorbed Power Watts	Wheel Hop % Time Off the Ground Front	Rear
Small - Measured Class C Road 20 mph	2.0	-	-
Medium - Predicted Class D Road 20 mph Class C Road 20 mph	 10 2.5	 3 0.1	 6 0.1
Large - Measured Class C Road 20 mph	3.4	-	-

Table 19. 6K and 10K RTFLT Stability Requirements

Requirement	Current RTFLT Requirements	Feasibility Study
All Wheels Remain on the Ground Under the Following Conditions:		
1A. 10K Static F-A Stability		No Change
o Surface	Level	
o Fork Position	Max Forward	
o Steer Angle	Max	
o Lift Load	1.5*Rated	
1B. 6K Static F-A Stability		No Change
o Surface	Level	
o Fork Position	21.5 Feet	
o Lift Load	4400 lb.	
o Surface	Level	
o Fork Position	MLRS Config	
o Lift Load	9000 lb.	
2. Dynamic Fore-Aft		No Change
o Emergency Stop From	2 mph	
o Downhill Grade	30 %	
o Load	Rated	
3. Side Slope Operation		No Change
o Turn	Full Circle	
o Steer Angle	Max	
o Load	Rated & None	
o Slope (10K)	15 %	
o Slope (6K)	30 %	

2.2.3 Analysis Study

2.2.3.1 General. The analysis of the various suspension alternatives was done in several phases. The first phase was to determine the overall feasibility of suspended axles and harmonic dampers to meet the operator absorbed power requirement and to reduce wheel hop to a level that would keep the machine controllable at high speeds. These components were evaluated with the 2-D ride analysis (frequency response) model. The second phase was to use the optimum suspension characteristics that were found using the 2-D model and analyze the system from a 3-D, large motion standpoint. The 3-D vehicle dynamics model was used to examine the ability of the machines to traverse large obstacles (8" speed bump and 12" potholes), handle their rated load, and operate on side slopes. Following is a discussion of these analysis phases.

2.2.3.2 Natural modes of vehicle vibration. The interconnections, spring rates, masses, and inertias of the components of the RTFLT vehicle systems influence its natural frequencies. Ride analysis model variations included changing the suspension spring rates. Therefore, the natural frequencies and modes of vibration were different for each model variation. Table 20 lists the frequencies of the natural modes of vibration for the 6K RTFLT and gives a brief description of the mode for each model variation. Fig. 31 is an animation of the pitch mode (rocking chair motion) of the unsuspended baseline 6K RTFLT. Fig. 32 shows the predominantly vertical motion of the bounce mode of the 6K RTFLT. A higher frequency wheel hop mode is shown in Fig. 33. Fig. 34 shows both the vehicle and the linkage pitching in a "scissor" fashion in the 6K RTFLT suspended linkage model variation.

Table 20. 6K RTFLT Modes of Vehicle Vibration

Model Variation	Frequency (Hertz)	Description of the Mode
Unsuspended Baseline	1.7 2.5	Vehicle Pitch Vehicle Bounce
Front Axle Suspended	1.4 2.0 7.2	Vehicle Pitch Vehicle Bounce Front Wheel Hop
Both Axles Suspended	1.1 1.7 7.2 7.2	Vehicle Pitch Vehicle Bounce Wheel Hop, Wheels in Phase Wheel Hop, Wheels out of Phase
Suspended Linkage Assembly	1.4 1.9 3.8	Vehicle - Linkage Pitch Scissor Effect Vehicle Pitch about Front Axle with Linkage Scissor Effect Vehicle Pitch about Rear Axle Little Linkage Movement

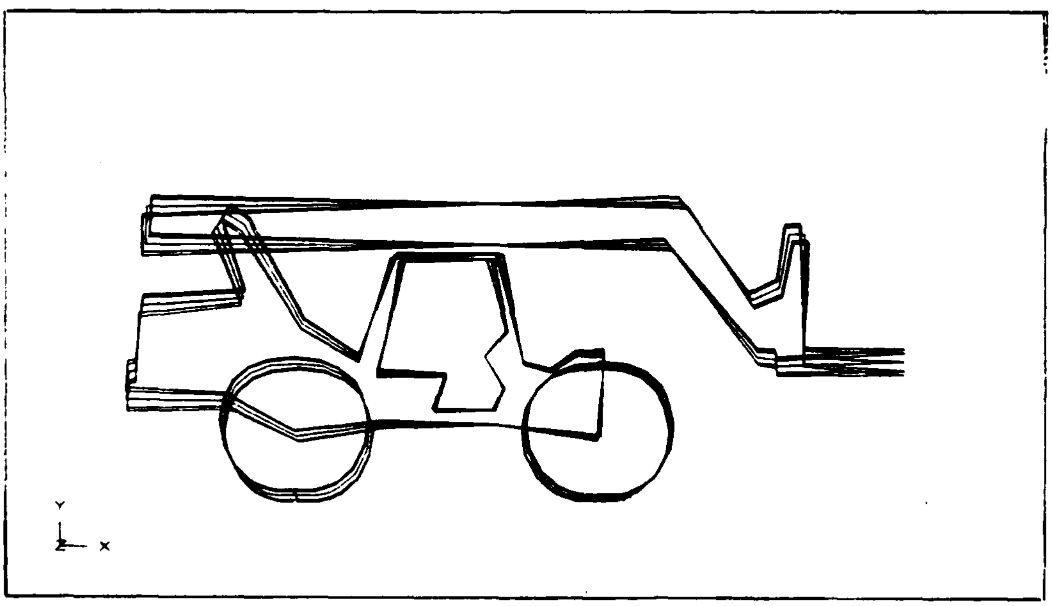


Figure 31. 6K RTFLT Pitch Mode Animation (Unsuspended - 1.7 Hertz)

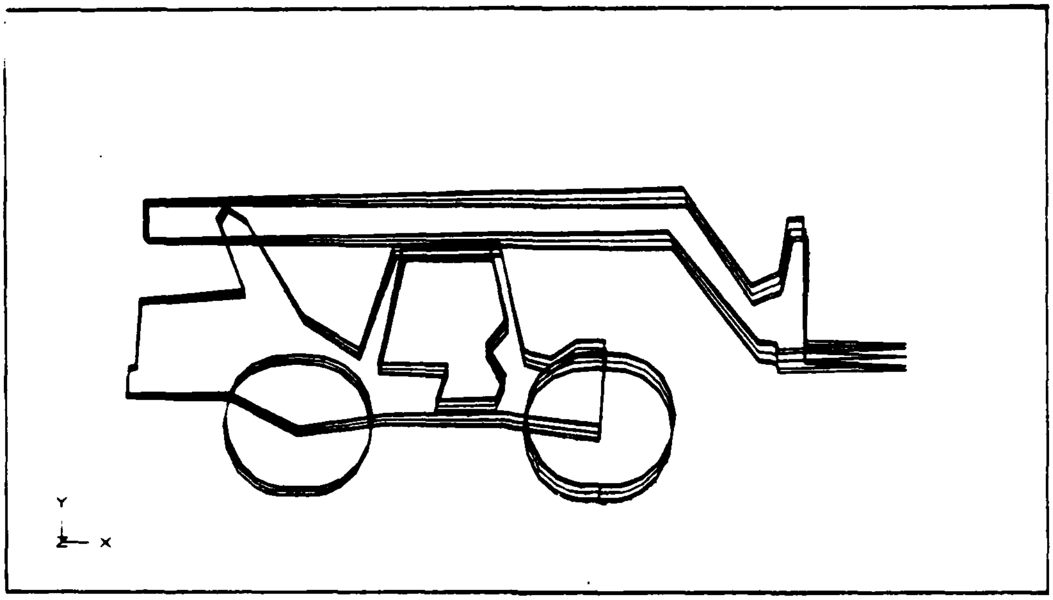


Figure 32. 6K RTFLT Bounce Mode Animation (Unsuspended - 2.5 Hertz)

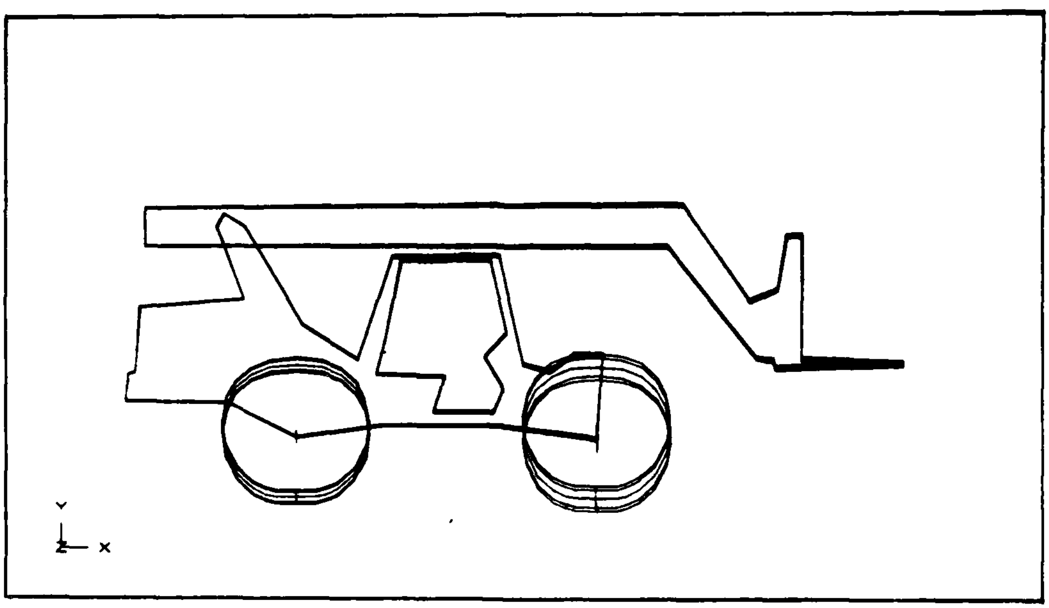


Figure 33. 6K RTFLT Wheel Hop Mode Animation (Both Suspended - 7.2 Hertz)

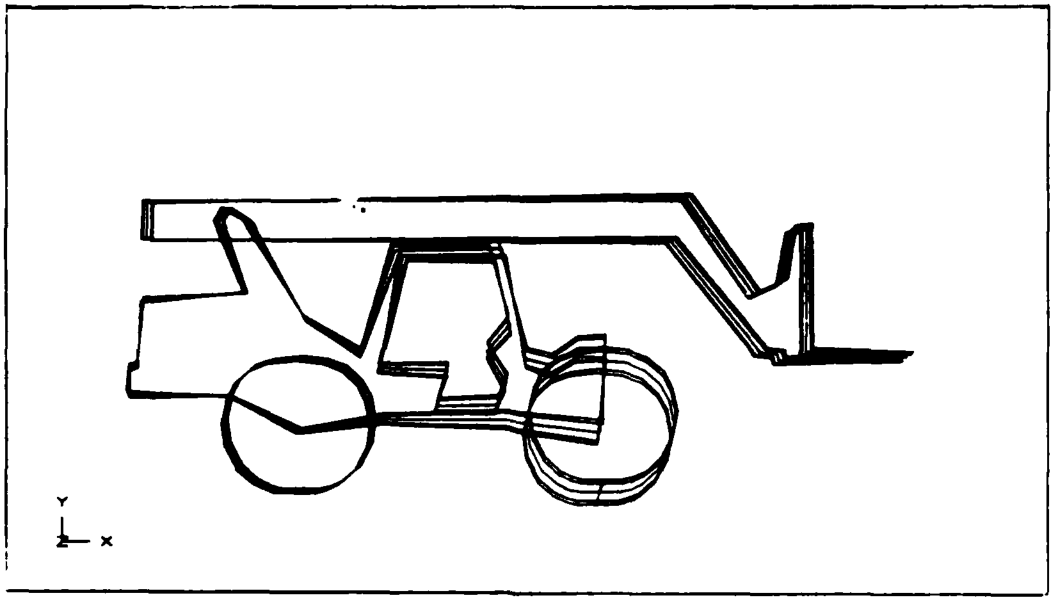


Figure 34. 6K RTFLT Vehicle/Linkage Pitch Mode Animation Suspended Linkage (3.8 Hertz)

Descriptions of the modes of vibration for the 10K RTFLT model variations are listed in Table 21. The modes are similar to those of the 6K RTFLT, but occur at different frequencies. Fig. 35 animates the pitch mode of the 10K RTFLT. A combination bounce and pitch mode of the 10K RTFLT is shown in Fig. 36. A wheel hop and vehicle pitch mode of the 10K RTFLT is animated on Fig. 37. A wheel hop mode for the 10K RTFLT with both axles suspended is shown in Fig. 38.

Table 21. 10K RTFLT Modes of Vehicle Vibration

Model Variation	Frequency (Hertz)	Description of the Mode
Unsuspended Baseline	1.7 2.2	Vehicle Pitch Vehicle Bounce with some Pitch About the Rear Bumper
Front Axle Suspended	1.2 1.8 3.7	Vehicle Pitch about Rear Axle Vehicle Bounce Front Wheel Hop
Both Axles Suspended	1.1 1.6 4.1 4.2	Vehicle Pitch Vehicle Bounce/Pitch about the Rear Bumper Wheel Hop, Wheels in Phase Wheel Hop, Wheels out of Phase
Suspended Counterweight	1.3 2.1 3.1	Vehicle Pitch about Front Axle Counterweight out of phase Vehicle Pitch about Rear Axle Little Counterweight movement Vehicle Pitch about Front Axle Counterweight out of phase

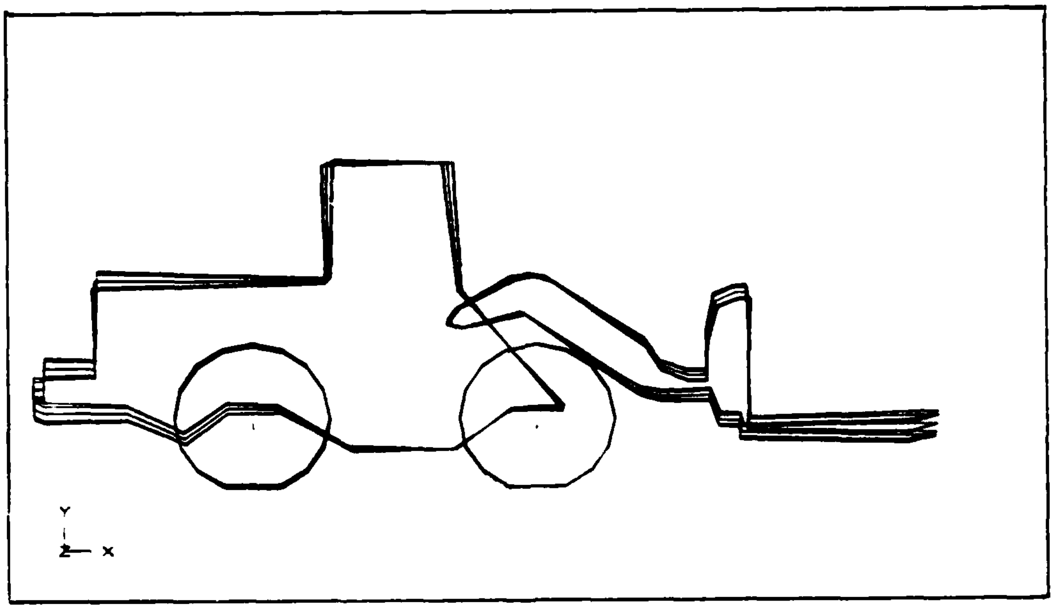


Figure 35. 10K RTFLT Pitch Mode Animation
Both Axles Suspended (1.1 Hertz)

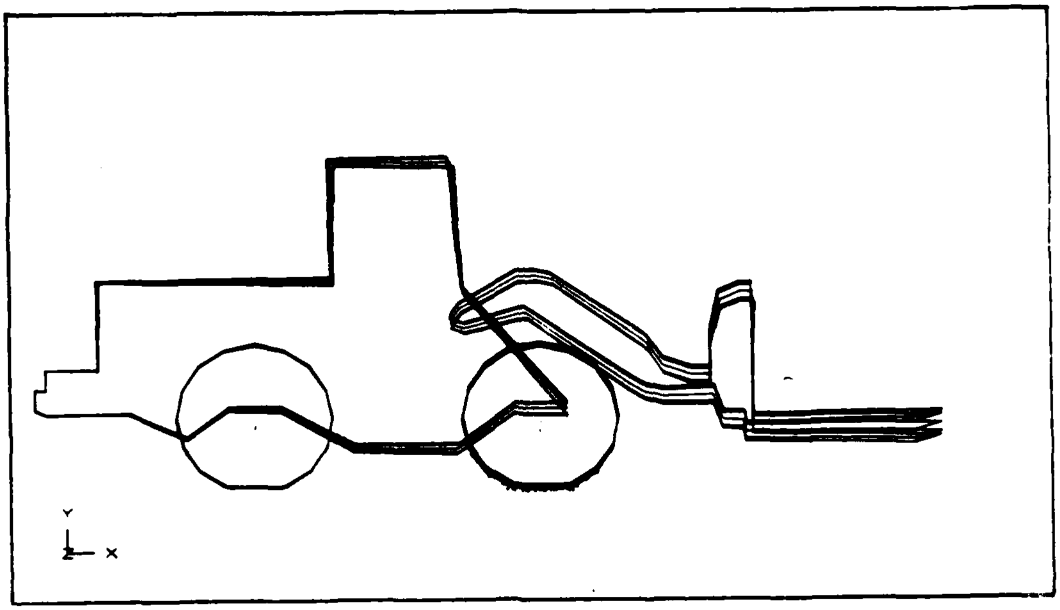


Figure 36. 10K RTFLT Bounce/Pitch Mode Animation
Both Axles Suspended (1.6 Hertz)

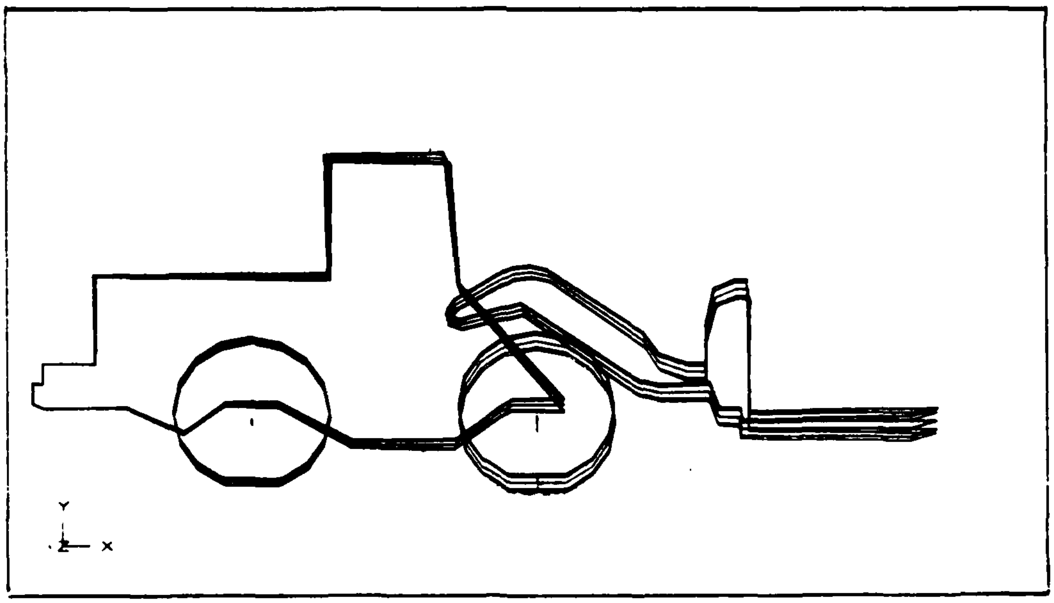


Figure 37. 10K RTFLT Wheel Hop/Pitch Mode Animation
Both Axles Suspended (4.1 Hertz)

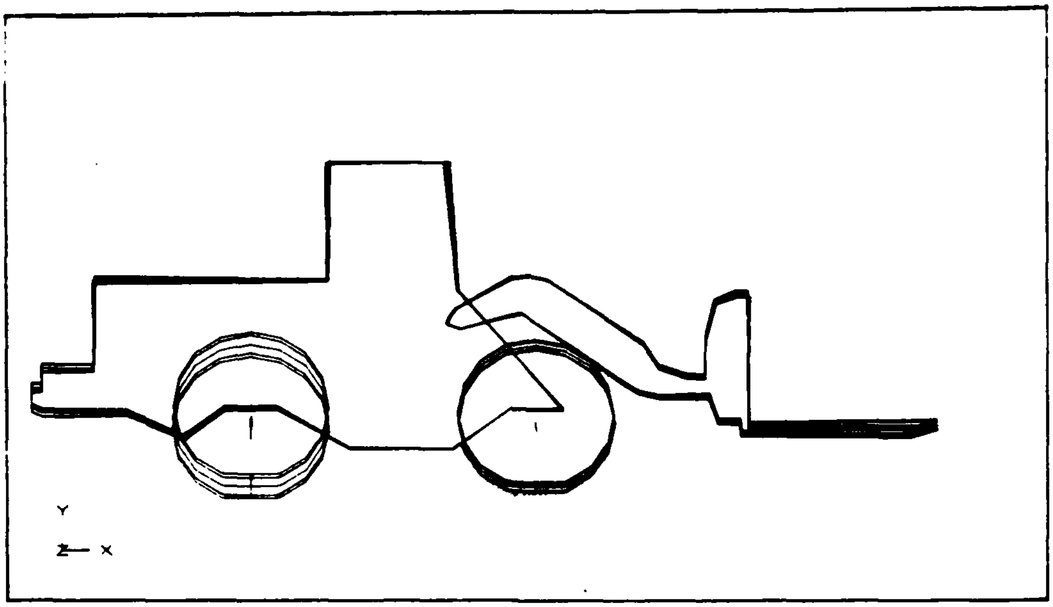


Figure 38. 10K RTFLT Wheel Hop Mode Animation
Both Axles Suspended (4.2 Hertz)

2.2.3.3 Response of the vehicle to road excitation. The natural modes of vibration discussed in 2.2.3.2 are excited when the road excitation frequency is equal to the natural mode frequency. Since a general road surface is made up of all wavelengths, the road excitation functions contain all frequencies within a frequency range.

The response to a 1.0 inch RMS road surface at 17 mph is presented as an example following of how the vehicle responds to road surface excitations. The vertical displacement of the vehicle at the operator's station to 1.0 inch RMS road surface at 17 mph is shown in Fig. 39. The horizontal displacement at the operator's station to the same road is shown in Fig. 40. These operator station response displacements (accelerations) are used with the human weighting factors of 2.2.1.2.4. to calculate total operator absorbed power. In this example, the absorbed power would be calculated for the 6K RTFLT on a 1.0 inch RMS road surface at 17 mph. Fig. 41 and Fig. 42 show the response of the front and rear tires respectively to the 1.0 inch RMS road at 17 mph. These displacements are used to calculate tire RMS deflections and subsequently, wheel hop (percent time off the ground). Similar response functions were evaluated for many combinations of road roughness and vehicle speed to obtain the results that will be presented in 2.2.3.4. and following sections.

The tires and vehicle vertical displacement at the operator's station follow the road input excitation function at low frequencies (e.g. < 1 Hertz), show a large gain over the road input excitations at the natural frequencies, but do not respond at all to the road input excitations at the higher frequencies (e.g. > 10 Hertz). The magnitude of the response at the natural frequencies is determined by the amplitude of the input road excitations at that frequency and the amount of damping in the system. The suspension system and to some extent the tires provide the damping in the system.

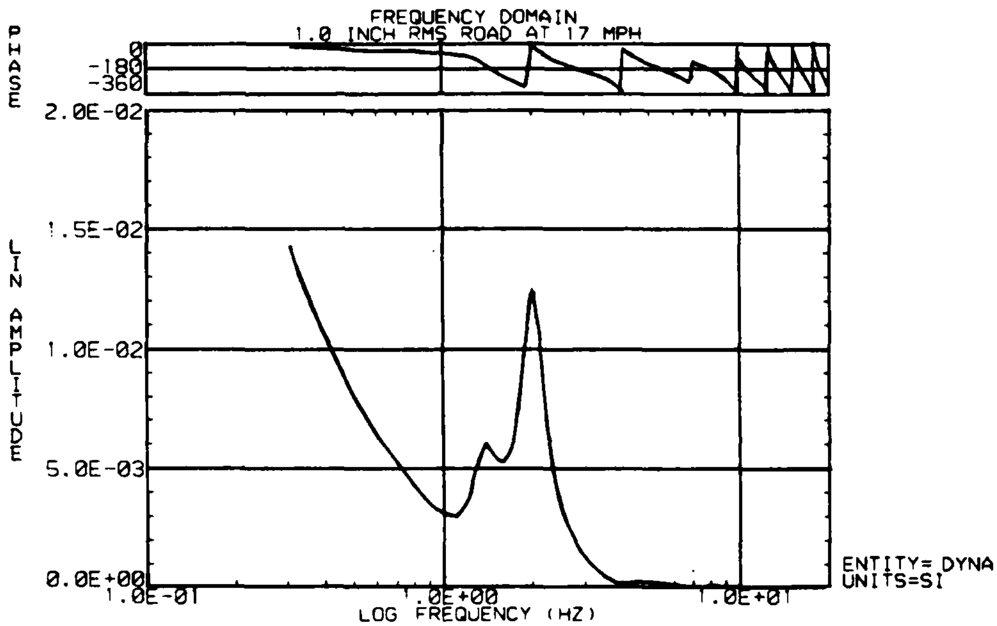


Figure 39. 6K RTFLT Vertical Displacement at the Operator's Station Due to the Road Excitation

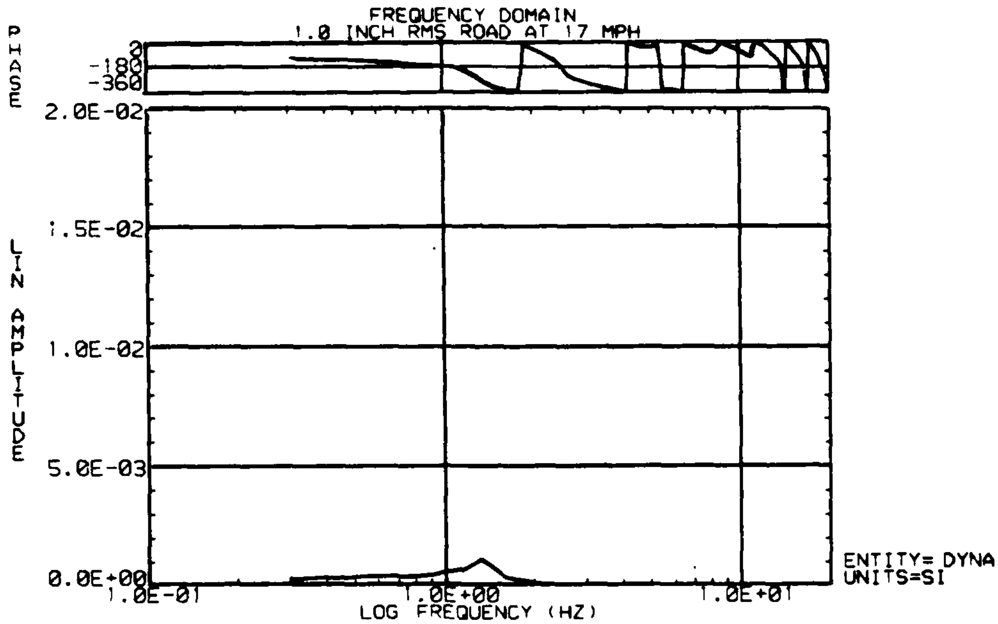


Figure 40. 6K RTFLT Fore - Aft Displacement at the Operator's Station Due to the Road Excitation

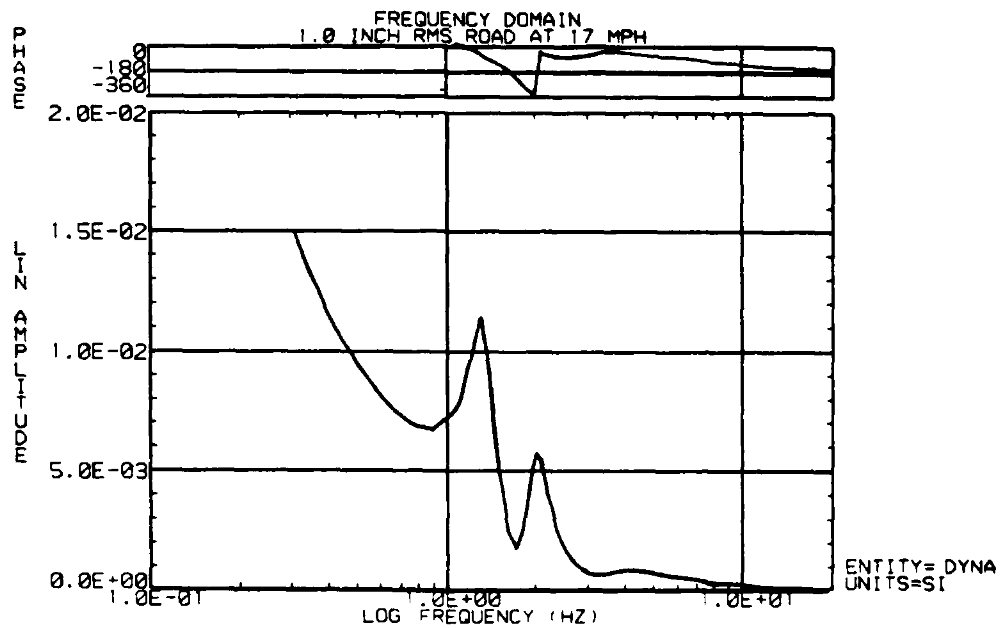


Figure 41. 6K RTFLT Front Tire Displacement Due to the Road Excitation

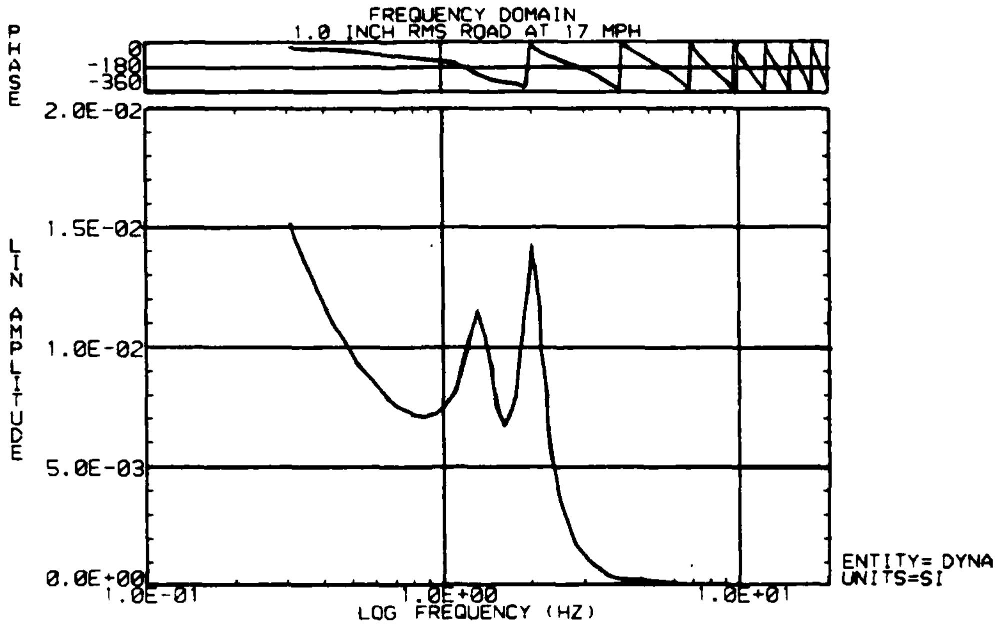


Figure 42. 6K RTFLT Rear Tire Displacement Due to the Road Excitation

2.2.3.4 Selecting suspension characteristics. Suspension spring rates and damping rates that allow the RTFLTs to operate over rough road surfaces were selected using a two step procedure. A first approximation to the correct spring rates and damping rates were determined by applying equations developed by J. P. Den Hartog, "Mechanical Vibrations", Fourth Edition, McGraw-Hill Book Co., 1956, p. 87. for dynamic vibration absorbers. The equations developed by Hartog apply to a simple 2 mass - 2 degree of freedom system. Therefore, the multi-degree of freedom system that represents the RTFLT with suspension was approximated with a 2 mass - 2 degree of freedom system. The equations for optimal spring rates and damping were then applied. These values were used as a starting value for stiffnesses used in the RTFLT ride models. The sensitivity of ride quality and wheel hop (controllability) to suspension stiffness was then determined using the RTFLT models. Those results are as follows.

The sensitivity results of the 6K RTFLT ride quality and wheel hop to front axle suspension stiffness are shown in Fig. 43, Fig. 44, and Fig. 45. The results are shown for 45 mph operation on class C and class D road surfaces. The best stiffness is selected by finding one that meets the ride quality and wheel hop criterion of 2.2.2. That is, the total operator absorbed power on the class C road at 45 mph must be less than or equal to 6 watts. It was also desirable for the 6 watt limit to be met on the class D road surface if possible. Front and rear wheel hop must be less than 10% time off the ground at 45 mph for the vehicle to be controllable. In addition, the lowest natural frequency of the vehicle should be greater than 1 Hertz to avoid operator "sea sickness". The best stiffness for the 6K RTFLT with front axle suspension was selected from Fig. 43, Fig. 44, and Fig. 45 by noting that 5000 to 10,000 lb/in stiffness gives optimum ride quality. Front wheel hop exceeds 10% at 10,000 lb/in but is acceptable at 5,000 lb/in. A check of the lowest natural frequency for 5000 lb/in (Table 20) reveals 1.4 Hertz. Therefore, 5,000 lb/in was chosen as the best suspension stiffness when the front axle only is suspended on the 6K RTFLT.

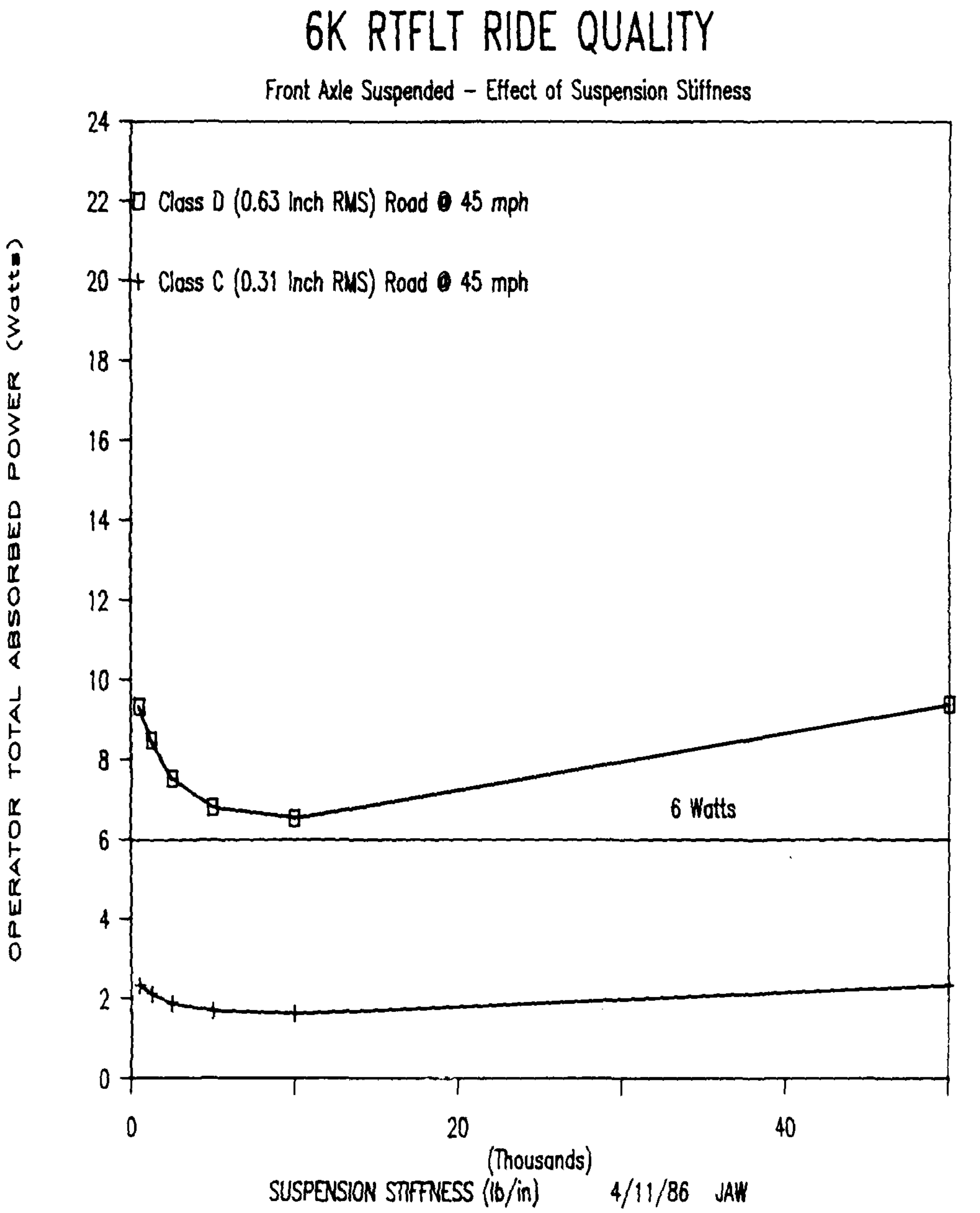


Figure 43. Effect of Front Suspension Stiffness on Ride Quality (6K RTFLT)

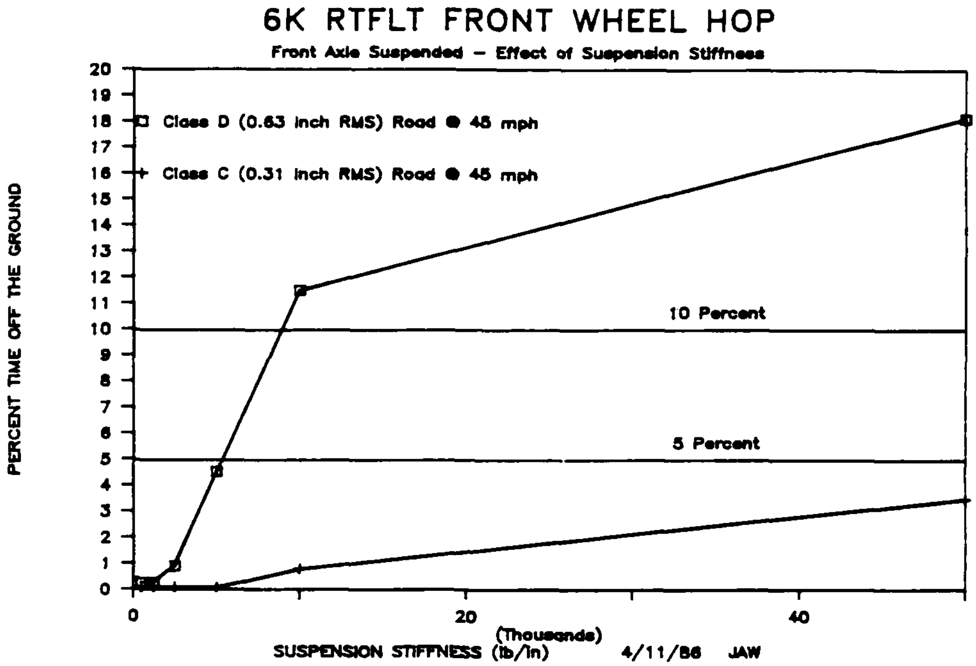


Figure 44. Effect of Front Suspension Stiffness on Front Wheel Hop (6K RTFLT)

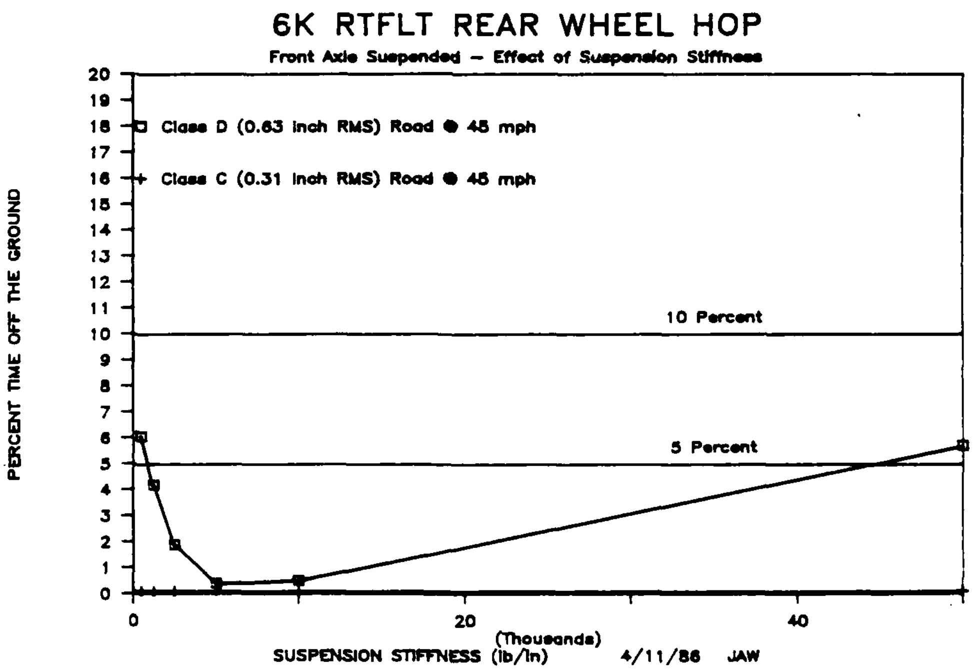


Figure 45. Effect of Front Suspension Stiffness on Rear Wheel Hop (6K RTFLT)

A similar stiffness selection process was followed for both front and rear axle suspensions (Fig. 46, Fig. 47, Fig. 48) A 5,000 lb/in stiffness on both front and rear axles met both the required 6 watt limit on the class C road surface as well as the class D road surface. Wheel hop was also acceptable for 5,000 lb/in.

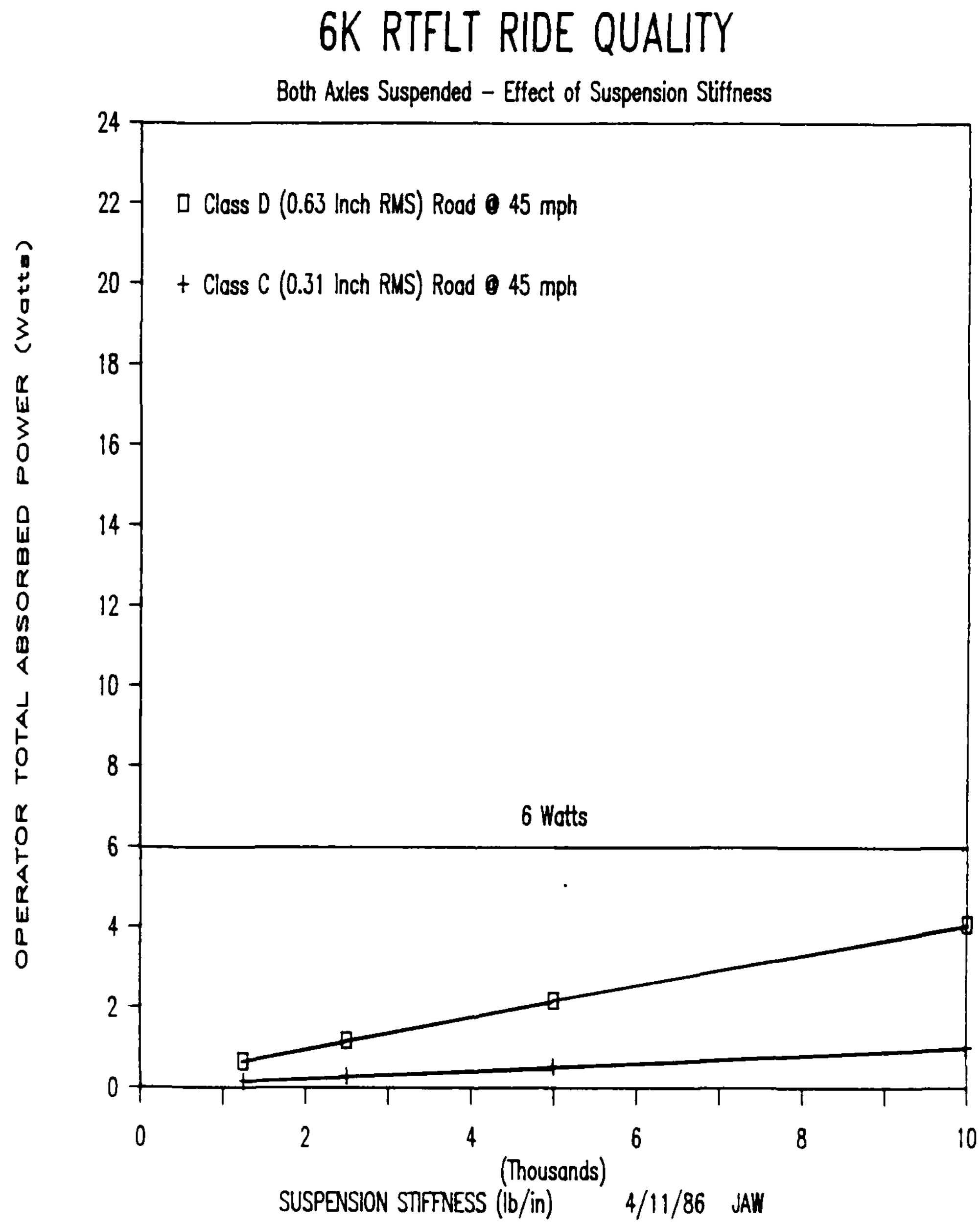


Figure 46. Effect of Front and Rear Suspension Stiffness on Ride Quality (6K RTFLT)

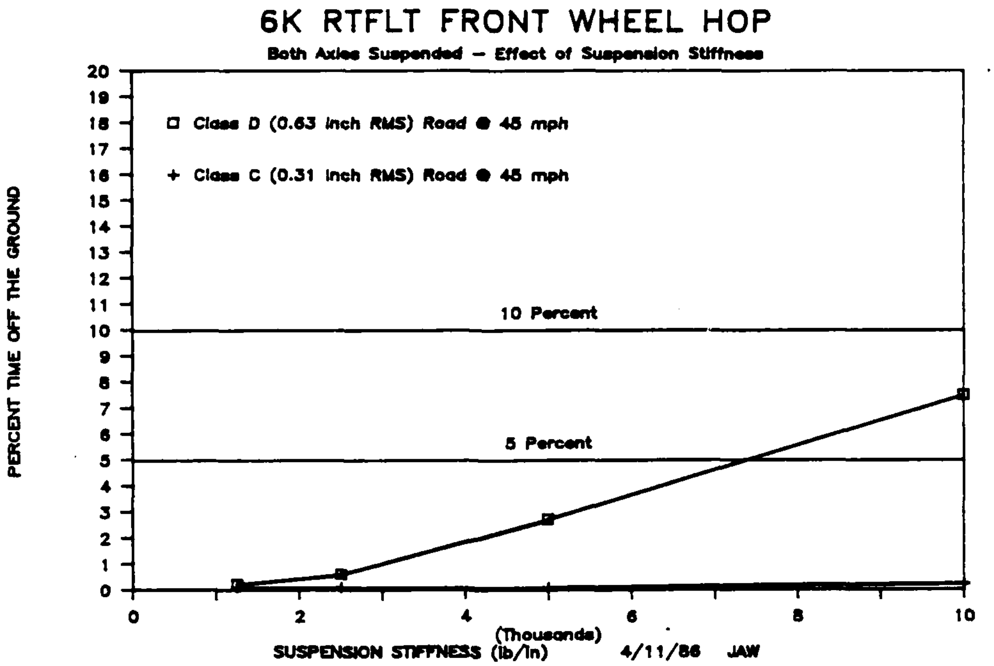


Figure 47. Effect of Front and Rear Suspension Stiffness on Front Wheel Hop (6K RTFLT)

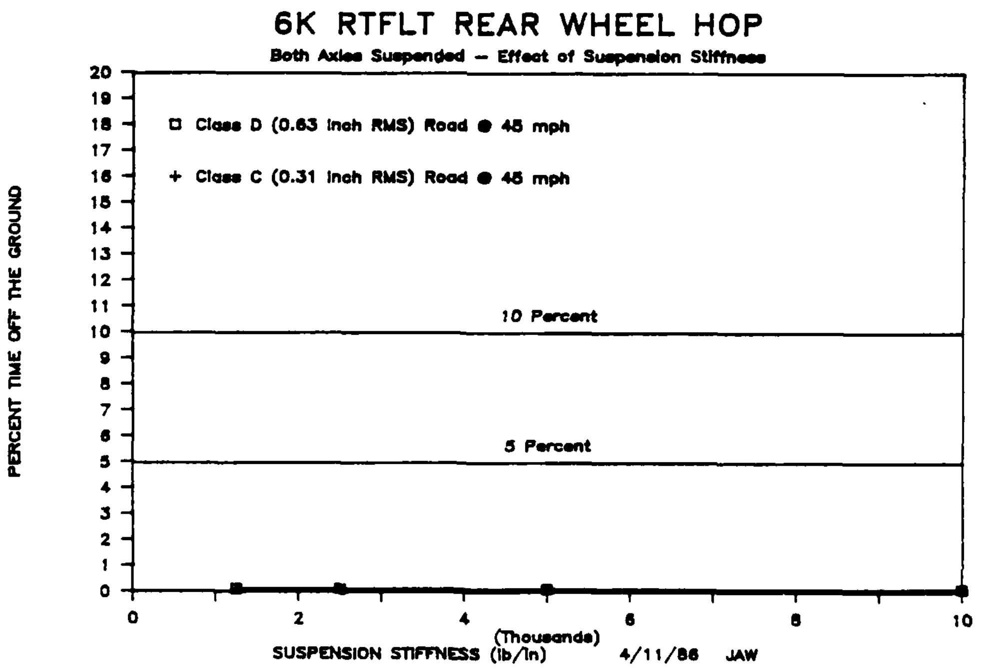


Figure 48. Effect of Front and Rear Suspension Stiffness on Rear Wheel Hop (6K RTFLT)

The linkage suspension stiffness was selected from Fig. 49, Fig. 50, and Fig. 51. A stiffness of 20,000 lb/in is barely acceptable for ride quality on the class C road and greatly exceeds the 6 watt desired limit on the class D road surface. Fig. 49 indicates that a stiffer linkage suspension would improve ride quality, but would make wheel hop unacceptable as shown in Fig. 50 and Fig. 51. Therefore, 20,000 lb/in was chosen as the best compromise stiffness.

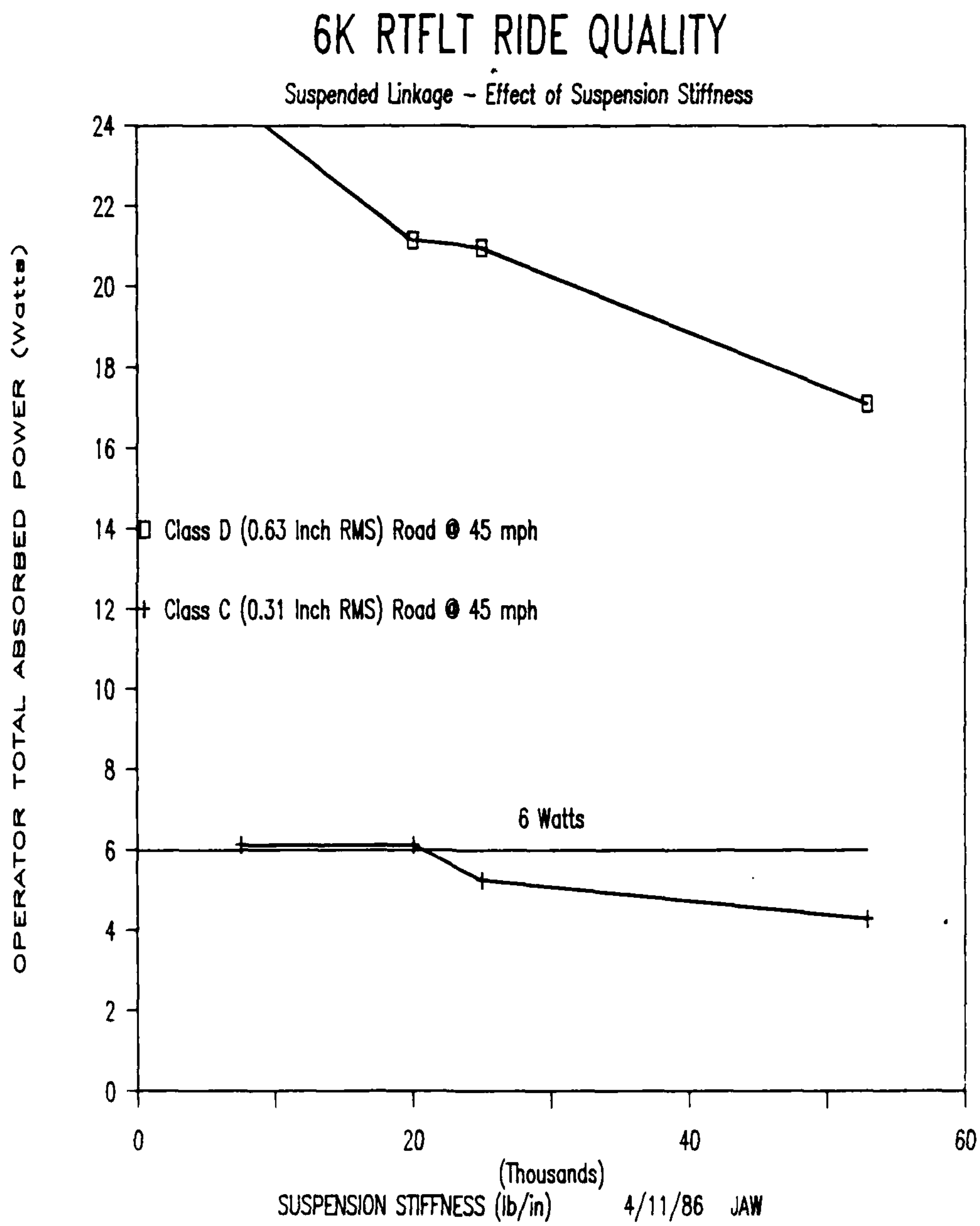


Figure 49. Effect of Suspended Linkage Stiffness on Ride Quality (6K RTFLT)

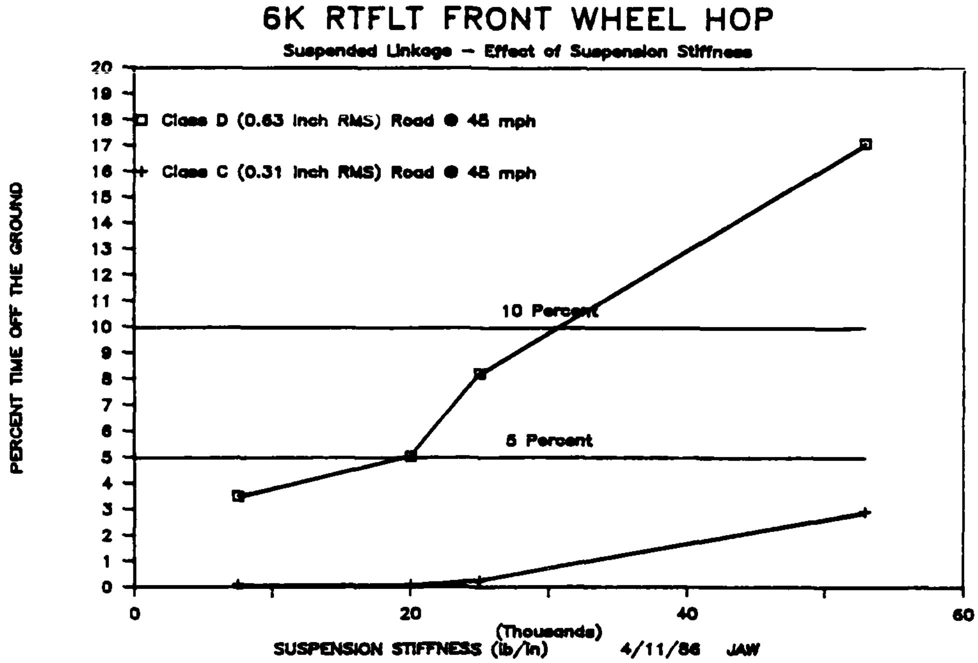


Figure 50. Effect of Suspended Linkage Stiffness on Front Wheel Hop (6K RTFLT)

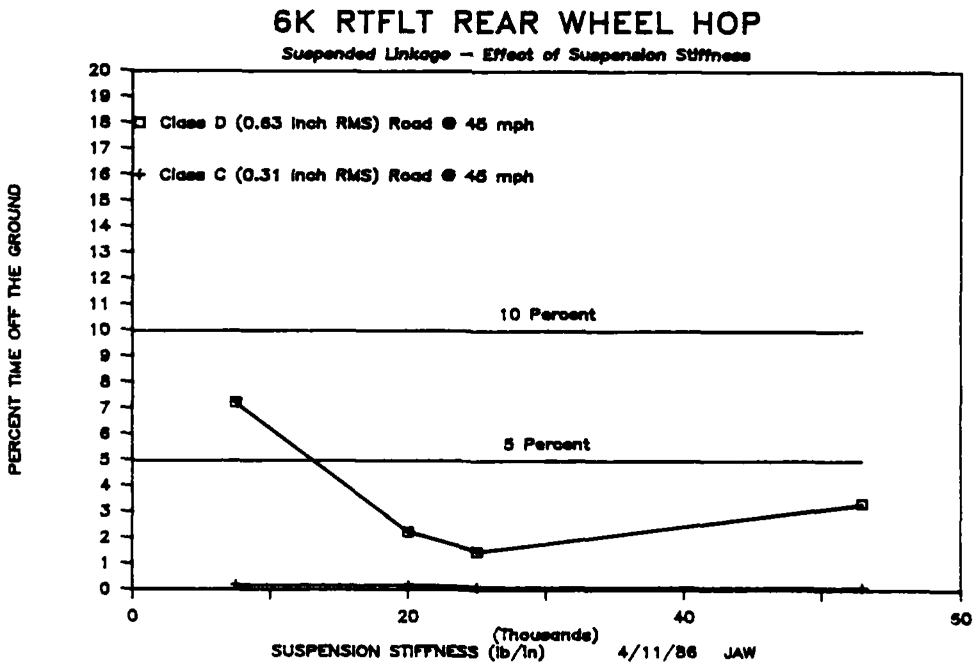


Figure 51. Effect of Suspended Linkage Stiffness on Rear Wheel Hop (6K RTFLT)

Results from the 10K RTFLT front axle suspension stiffness sensitivity study are shown on Fig. 52, Fig. 53, and Fig. 54. The same selection process was followed for the 10K RTFLT as for the 6K RTFLT. The best stiffness value for a front only stiffness was chosen as 2000 lb/in. This value meets the 6 watt requirement for the class C road surface and gives the minimum value for the class D road surface.

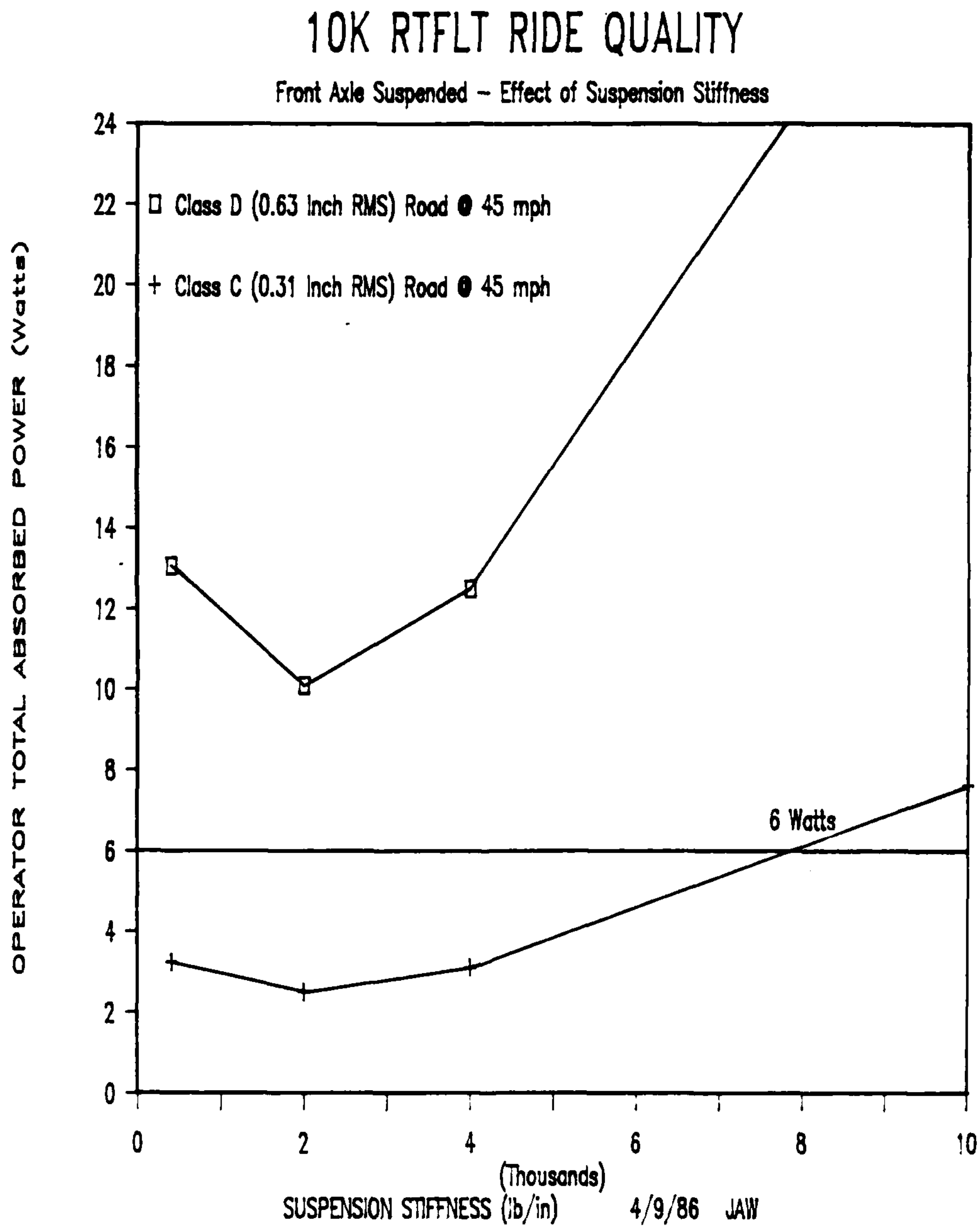


Figure 52. Effect of Front Suspension Stiffness on Ride Quality (10K RTFLT)

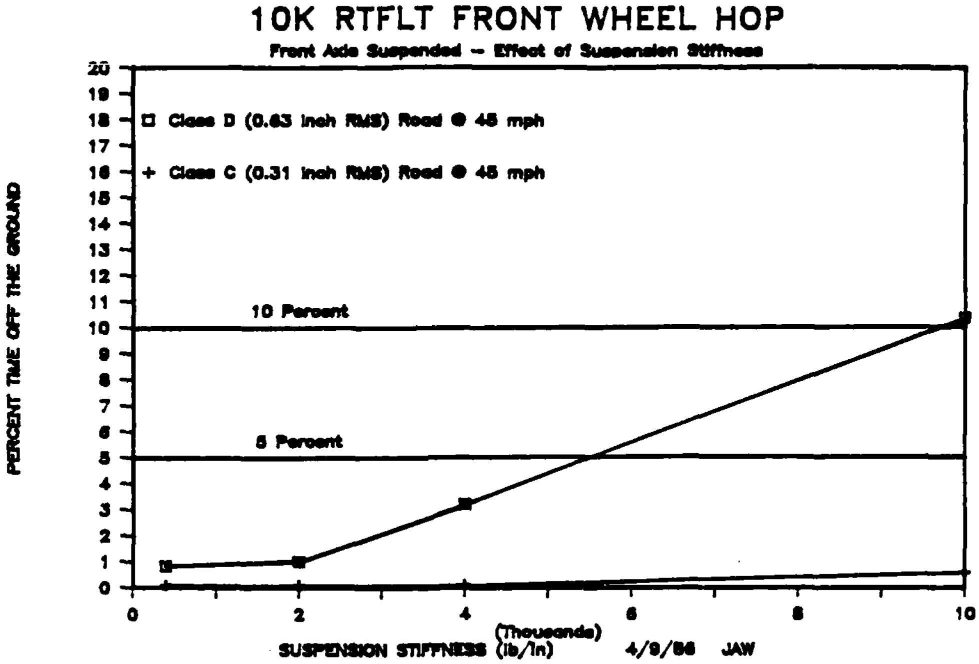


Figure 53. Effect of Front Suspension Stiffness on Front Wheel Hop (10K RTFLT)

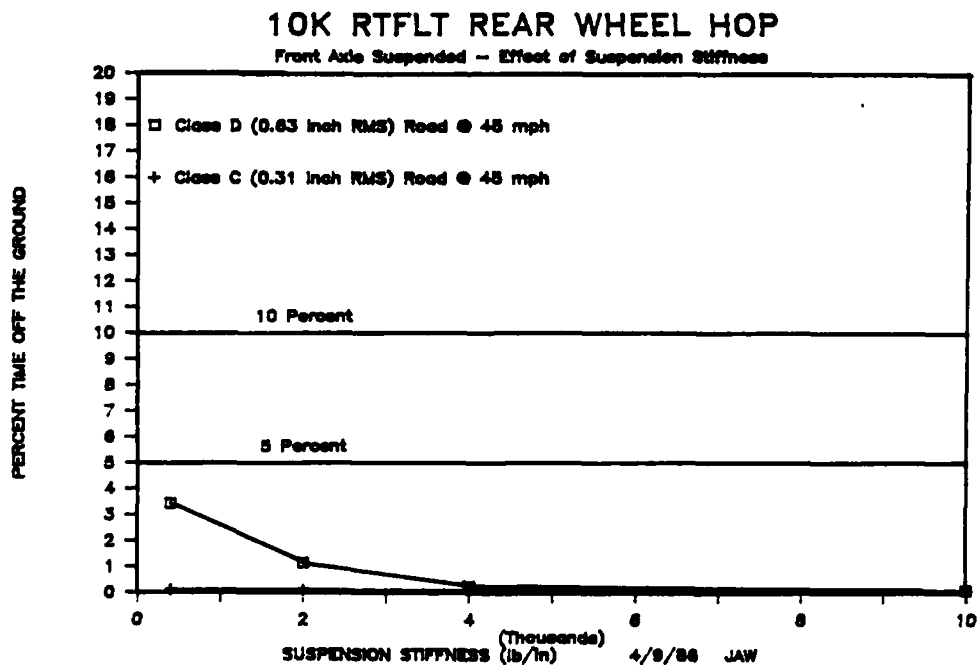


Figure 54. Effect of Front Suspension Stiffness on Rear Wheel Hop (10K RTFLT)

The results when both front and rear axles are suspended are shown in Fig. 55, Fig. 56, and Fig. 57. In this case 2000 lb/in meets both the class C road surface and the class D road surface 6 watt limit. However, a check of the vehicle natural frequencies revealed that the lowest natural frequency was less than 1 Hertz. Therefore, a 4000 lb/in stiffness was chosen since it resulted in a 1.1 Hertz natural frequency and still met the 6 watt limit on the class C road surface. However, Fig. 55 indicates than the desired 6 watt limit will be exceeded on the class D road surface with the 4,000 lb/in stiffness.

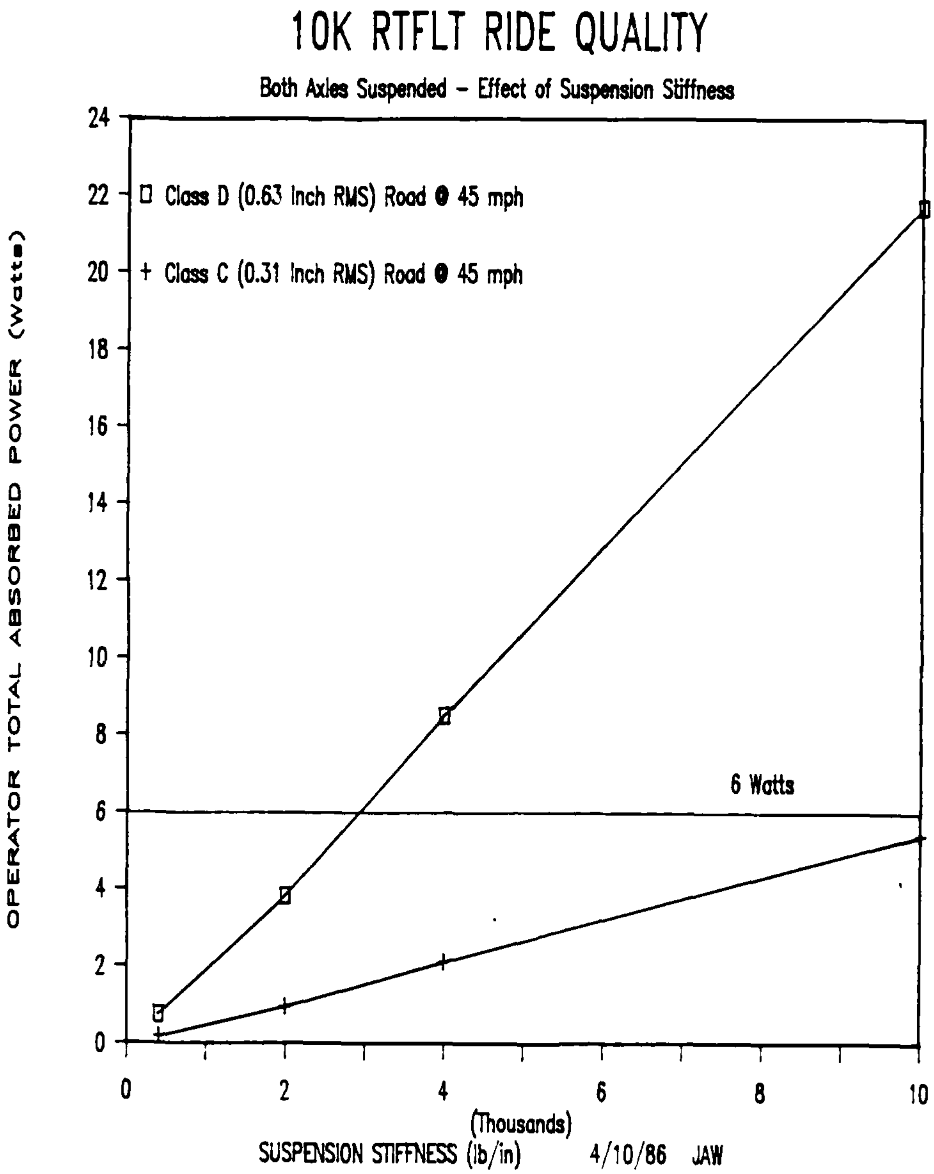


Figure 55. Effect of Front and Rear Suspension Stiffness on Ride Quality (10K RTFLT)

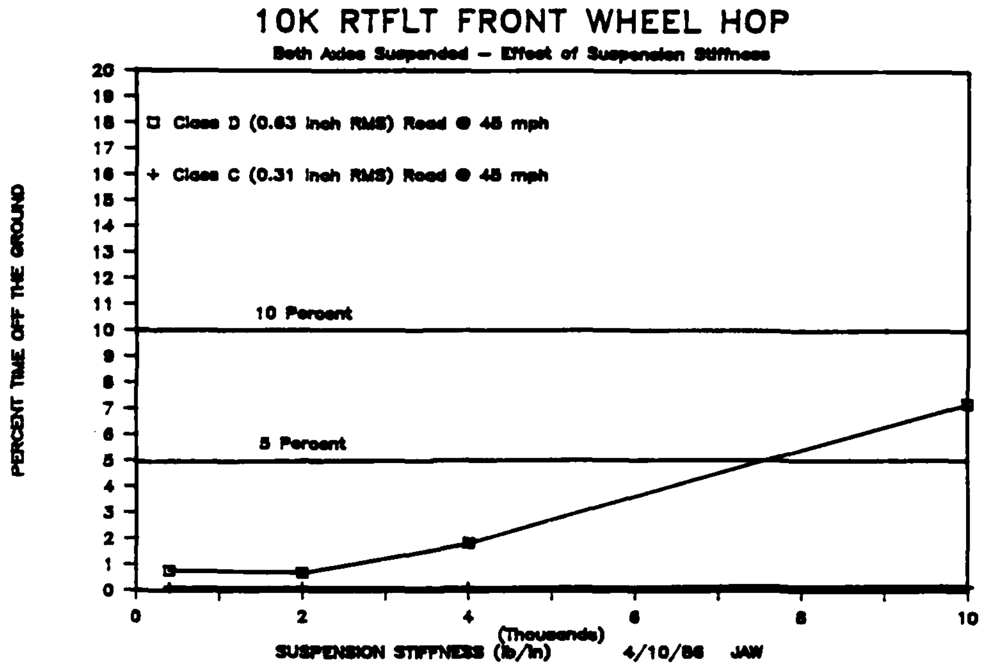


Figure 56. Effect of Front and Rear Suspension Stiffness on Front Wheel Hop (10K RTFLT)

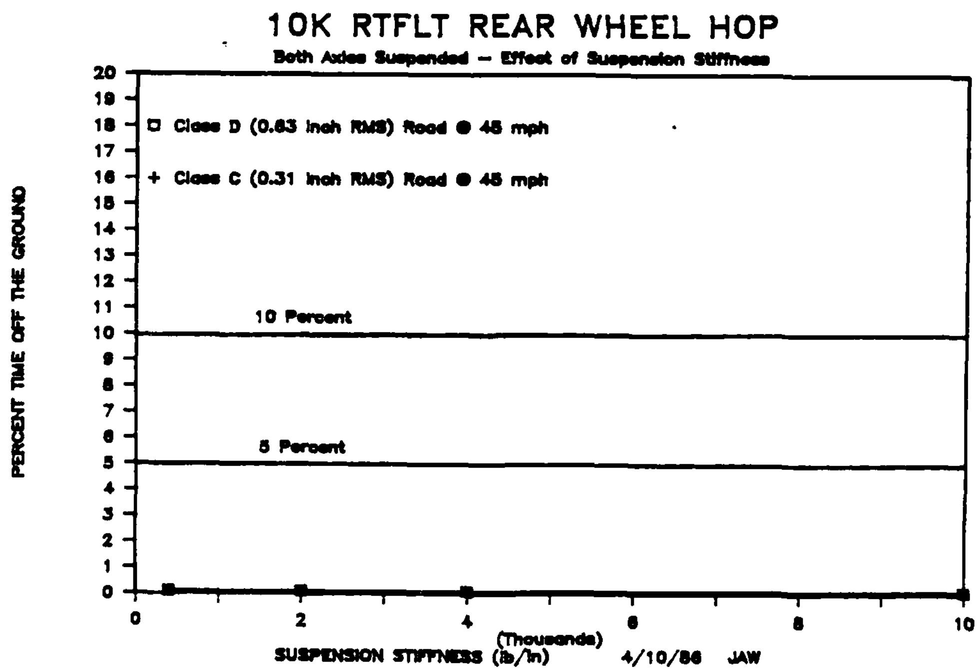


Figure 57. Effect of Front and Rear Suspension Stiffness on Rear Wheel Hop (10K RTFLT)

The effects of rear counterweight suspension stiffness on ride quality and wheel hop are shown in Fig. 58, Fig. 59, and Fig. 60. In this case, the 6 watt limit on the class C road could not be met. In addition, the operator absorbed power on the class D road surface was completely off the chart, and front wheel hop exceeded the 10% time off the ground limit. A stiffness value of 1530 lb/in was chosen as the best value for further analysis, but it will not meet the requirements.

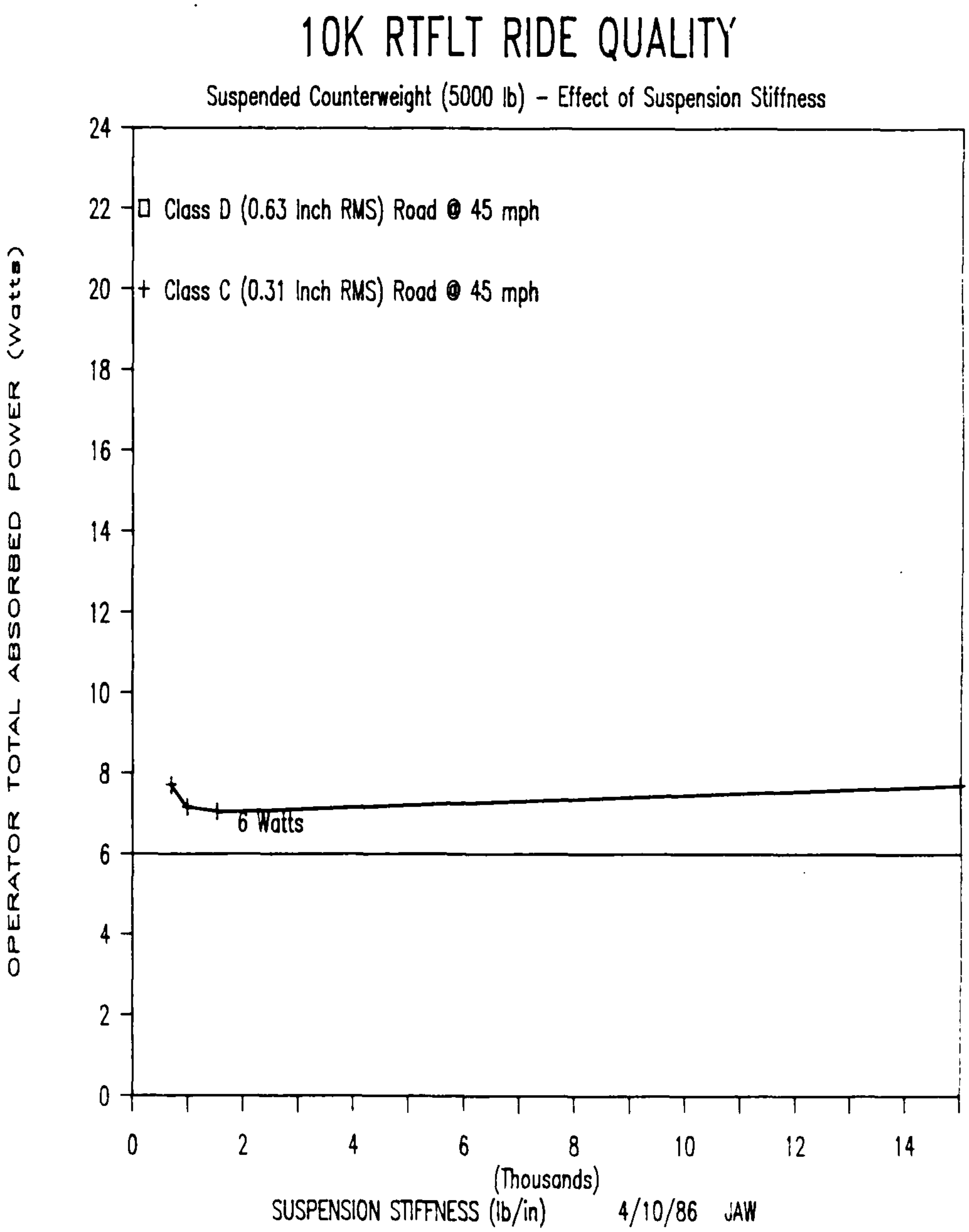


Figure 58. Effect of Suspended Counterweight Stiffness on Ride Quality (10K RTFLT)

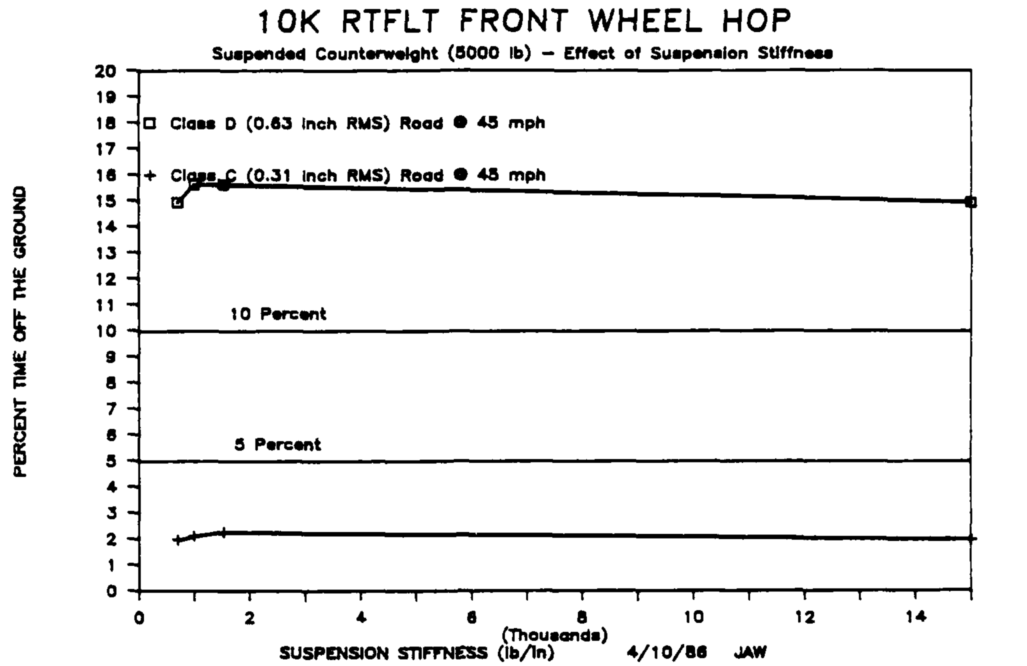


Figure 59. Effect of Suspended Counterweight Stiffness on Front Wheel Hop (10K RTFLT)

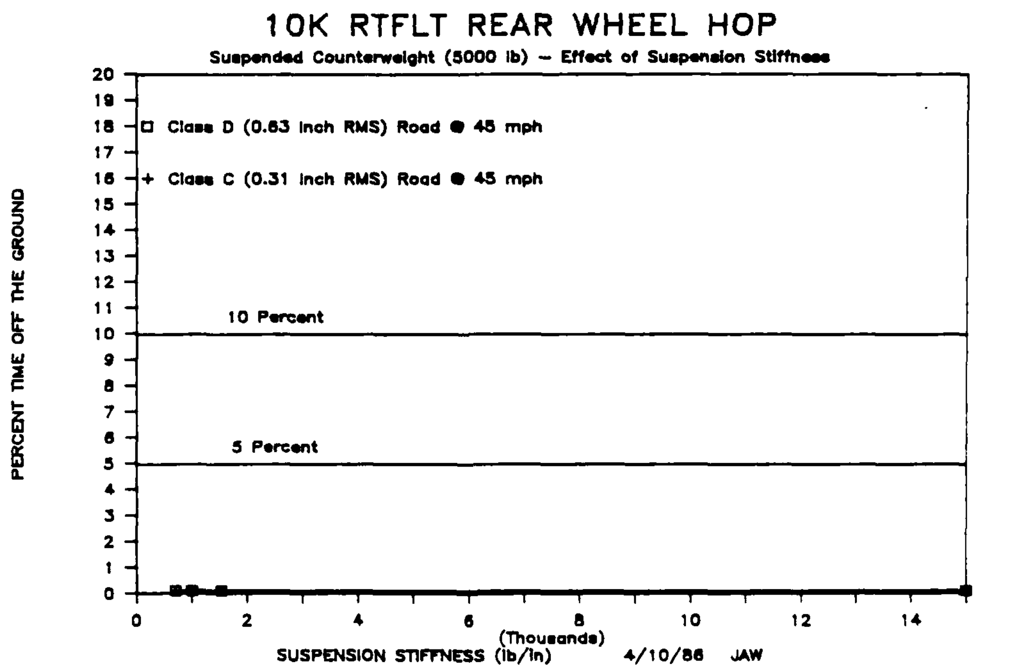


Figure 60. Effect of Suspended Counterweight Stiffness on Rear Wheel Hop (10K RTFLT)

2.2.3.5 Ride performance on four road surfaces. The ride quality and wheel hop (controllability) of the suspension variation with the stiffnesses selected in 2.2.3.4. can be predicted on several road surfaces. The road surfaces included are the class C and class D as well as two rougher road surfaces with 1.0 inch RMS and 1.5 inch RMS values. The road RMS roughness roughly doubles from the class C to the class D. An RMS road surface double the class D road would fall halfway between the 1.0 inch and the 1.5 inch RMS roads.

The ride quality and wheel hop for the unsuspended baseline 6K RTFLT traveling at various vehicle speeds on four road surfaces are shown in Fig. 61, Fig. 62, and Fig. 63, respectively. The ride quality and wheel hop are acceptable at all speeds on the class C road. However, the desired 6 watt power limit on the class D road as well as the 10% front wheel hop limit are exceeded at all speeds greater than 15 mph. Rear wheel hop is marginal at speeds greater than about 15 mph. Fig. 61, Fig. 62, and Fig. 63 also show a very large increase in the curves (particularly the total absorbed power curve) at 17-20 mph. This speed is where the road surface excites the bounce mode of vibration of the vehicle. Since there is no suspension on the vehicle to absorb this energy of vibration, the vehicle experiences large displacement gains over the road excitation input and bounces violently.

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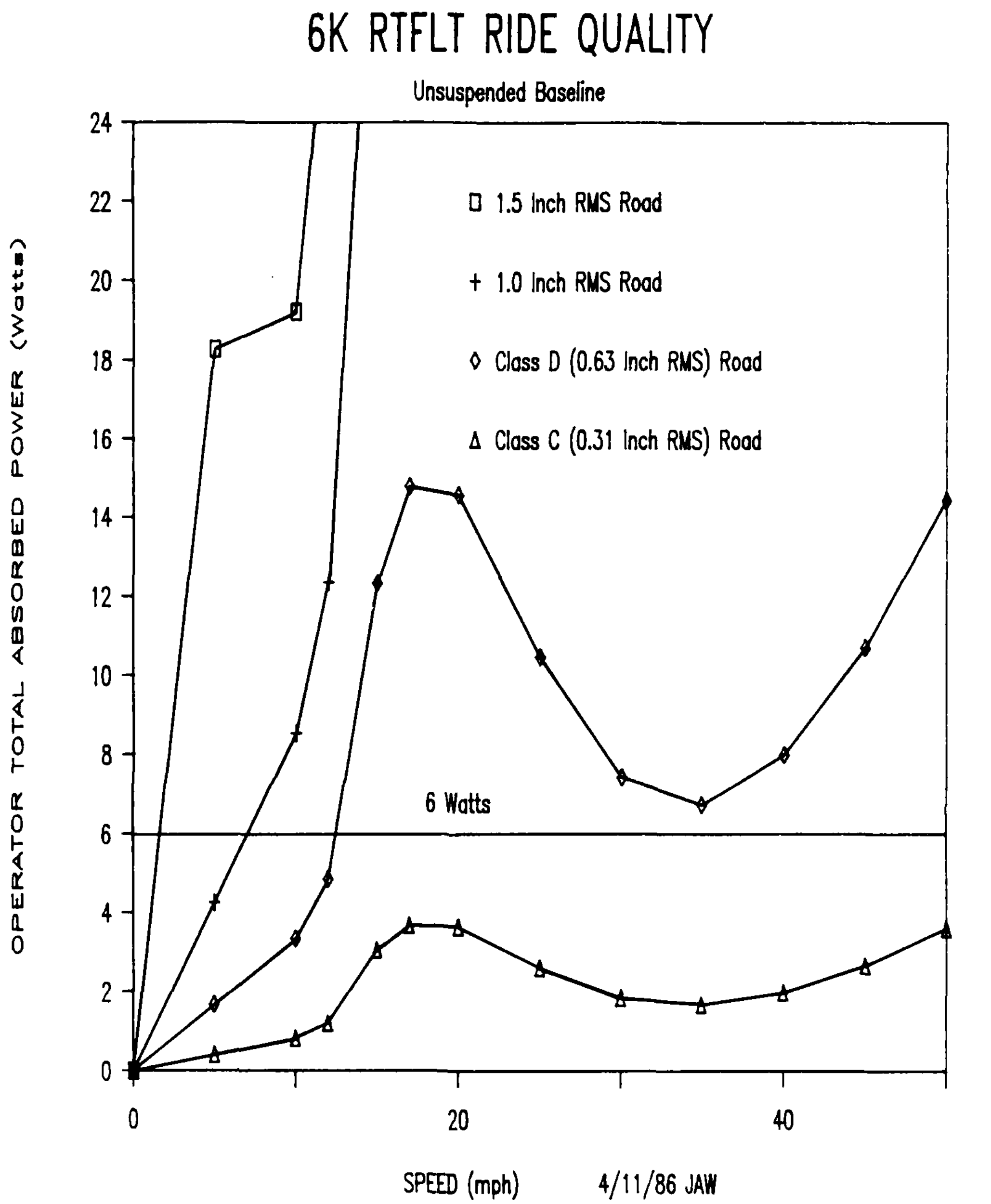


Figure 61. Ride Quality on Four Road Surfaces (6K RTFLT Unsuspended Baseline)

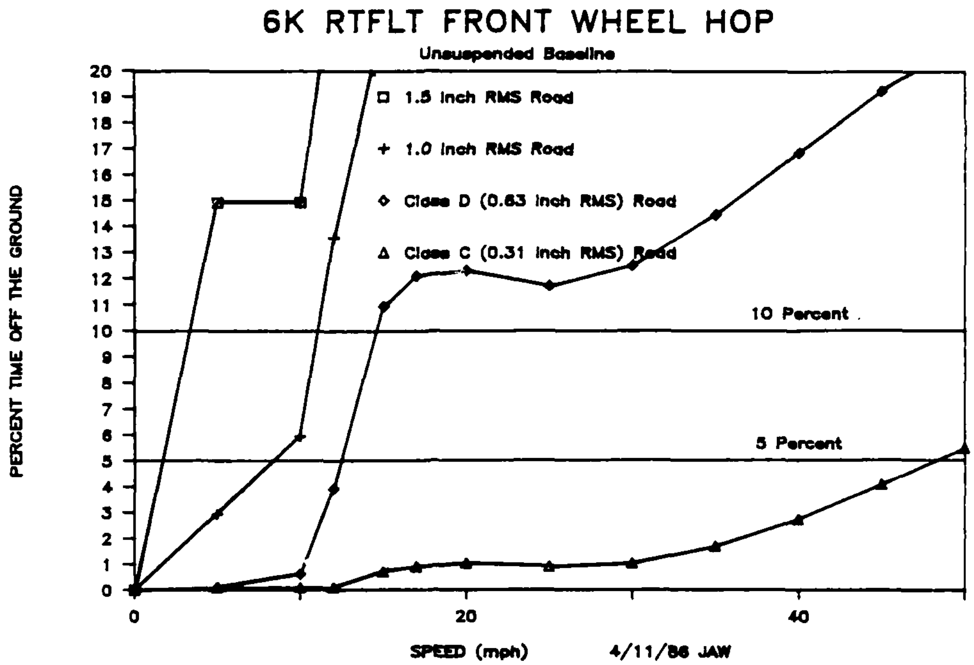


Figure 62. Front Wheel Hop on Four Road Surfaces (6K RTFLT Unsuspended Baseline)

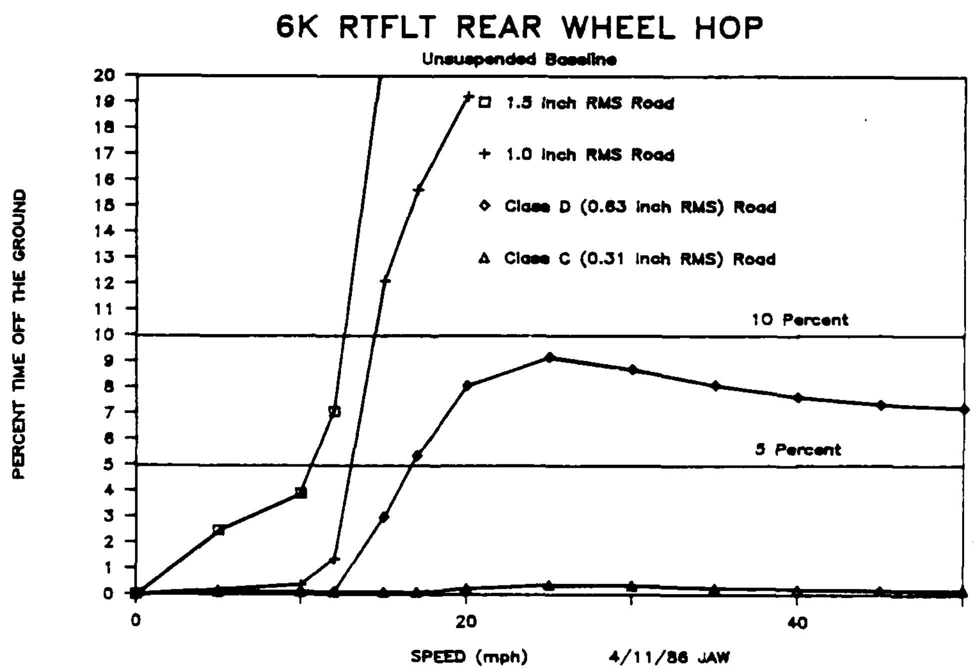
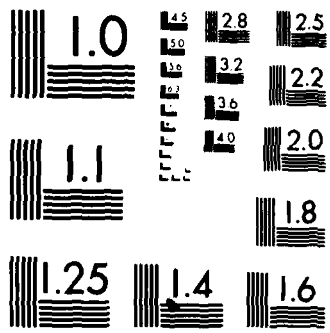


Figure 63. Rear Wheel Hop on Four Road Surfaces (6K RTFLT Unsuspended Baseline)

AD-A172 368 45 MPH 6000-POUND AND 10000-POUND ROUGH TERRAIN FORK LIFT TRUCK FEASIBILITY STUDY(U) CATERPILLAR INC PEORIA IL DEFENSE PRODUCTS DEPT G A ANDERS ET AL. 24 JUN 86 2/2

UNCLASSIFIED DAAK70-85-C-0111 F/G 13/6 NL



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The addition of the front axle suspension to the 6K RTFLT greatly improves the ride quality and wheel hop over the unsuspended baseline on all four road surfaces as shown in Fig. 64, Fig. 65, and Fig. 66. The class C road requirements are easily met. The 6 watt desired limit on the class D at 45 mph is exceeded slightly, but the wheel hop is acceptable at all speeds.

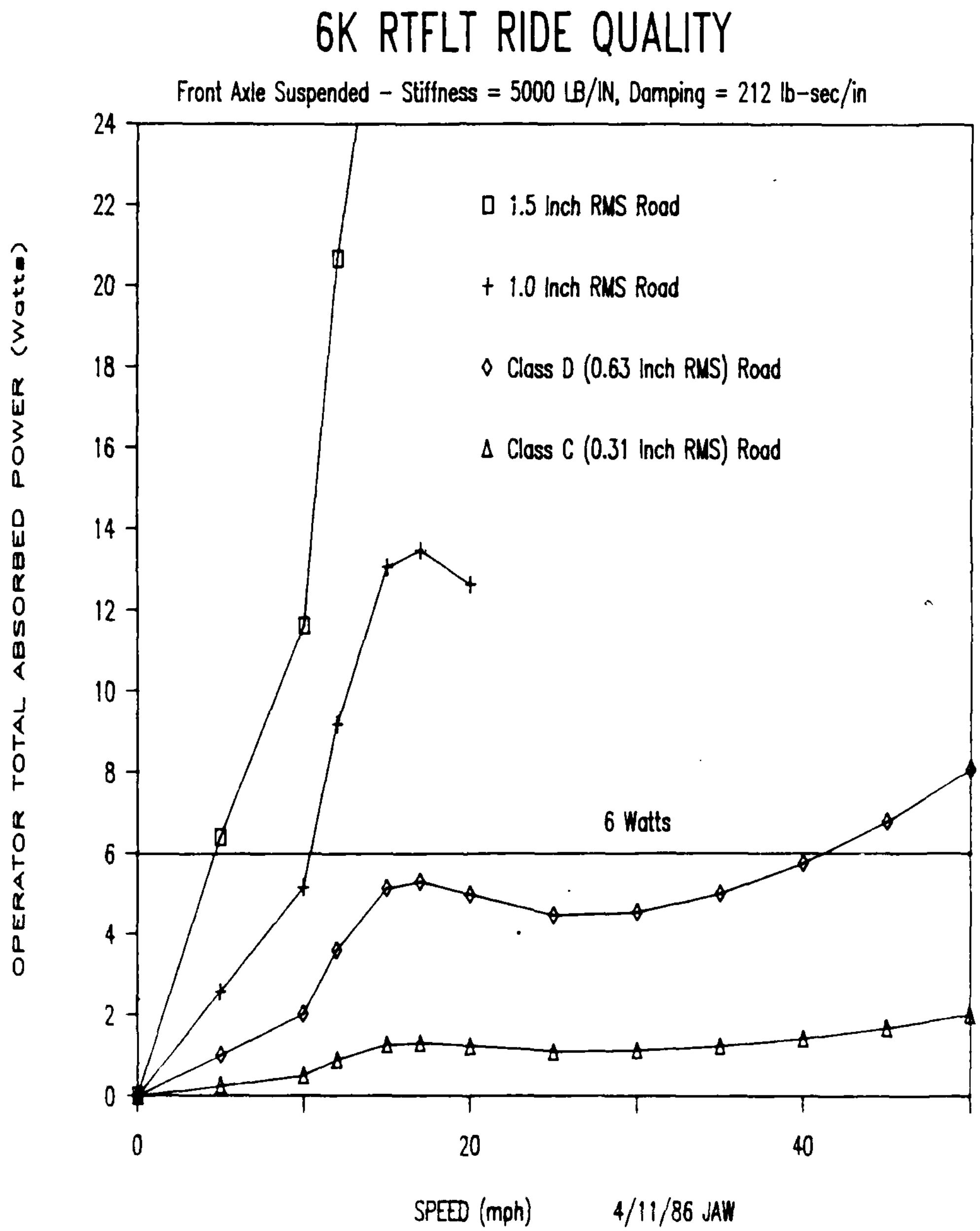

Figure 64. Ride Quality on Four Road Surfaces (6K RTFLT Front Axle Suspended)

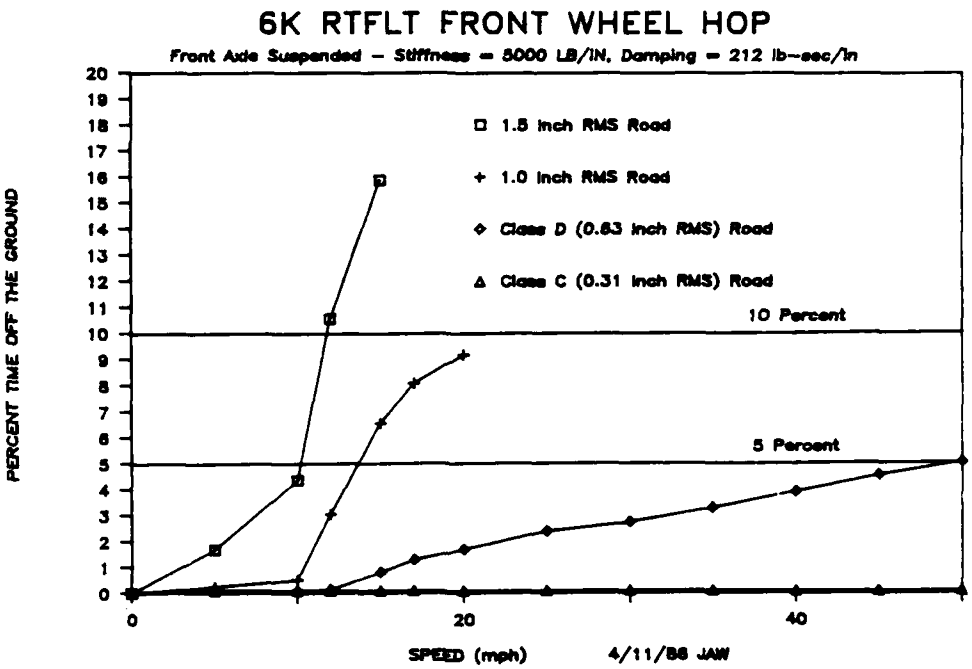

Figure 65. Front Wheel Hop on Four Road Surfaces (6K RTFLT Front Axle Suspended)

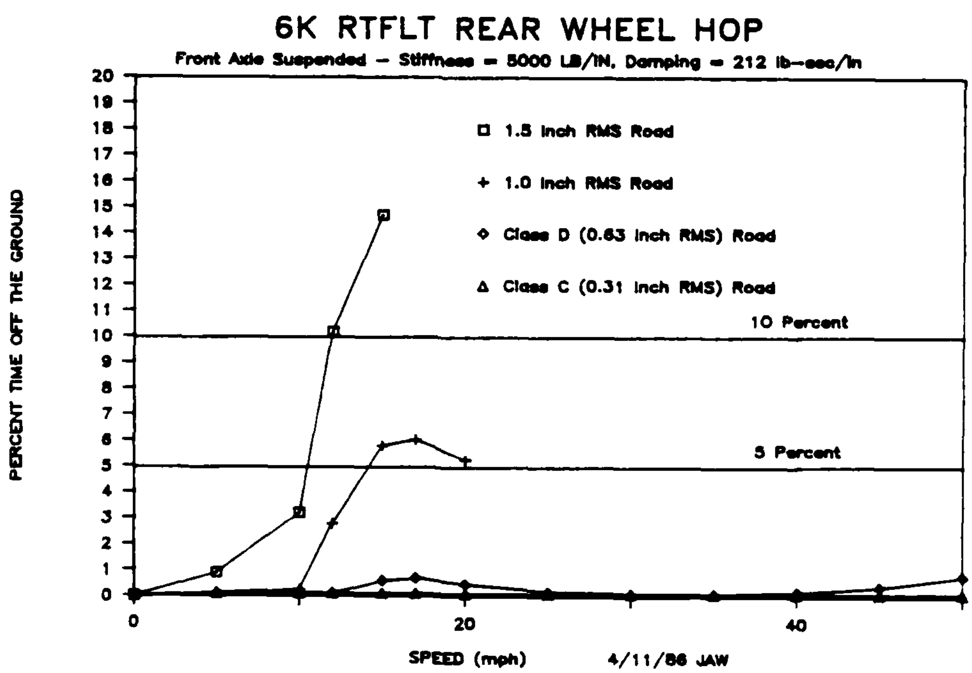

Figure 66. Rear Wheel Hop on Four Road Surfaces (6K RTFLT Front Axle Suspended)

The addition of a rear axle suspension as well as a front axle suspension improves the ride quality and wheel hop further (Fig. 67, Fig. 68, and Fig. 69). The 6 watt power limit as well as the wheel hop limit are easily met on both the class C and class D road surfaces.

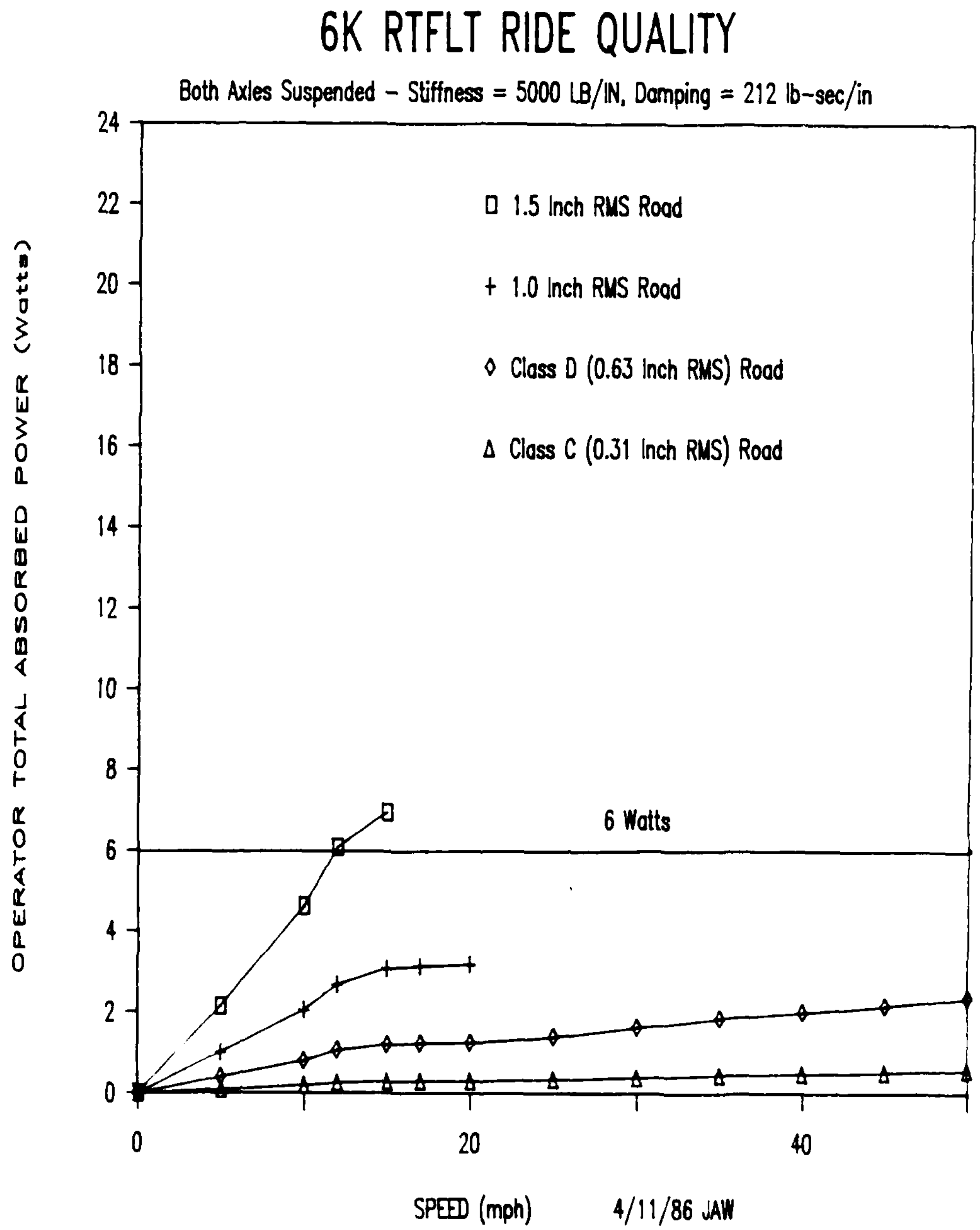

Figure 67. Ride Quality on Four Road Surfaces (6K RTFLT Both Axles Suspended)

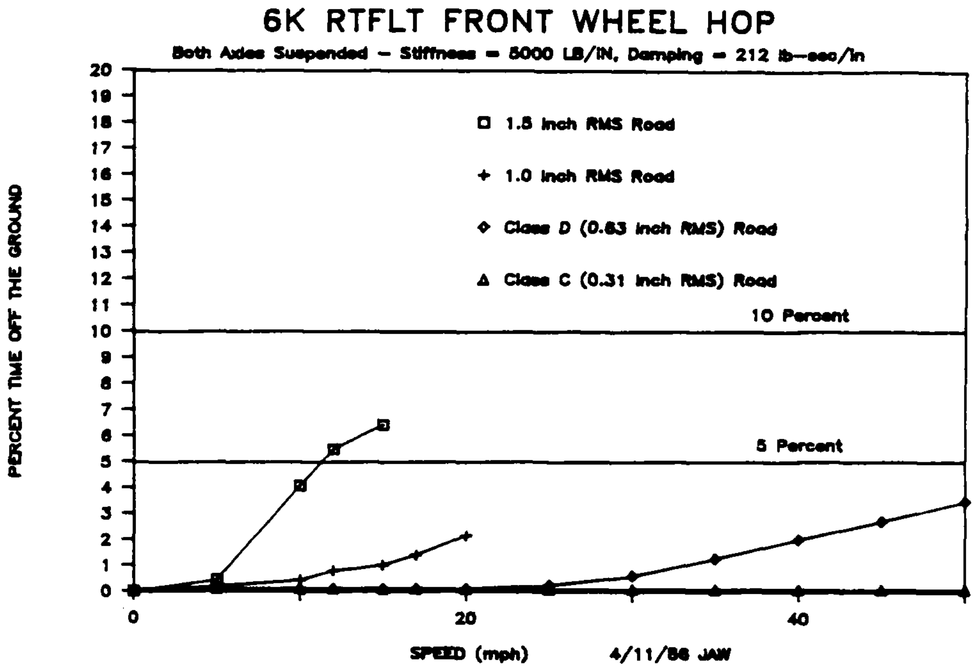

Figure 68. Front Wheel Hop on Four Road Surfaces (6K RTFLT Both Axles Suspended)

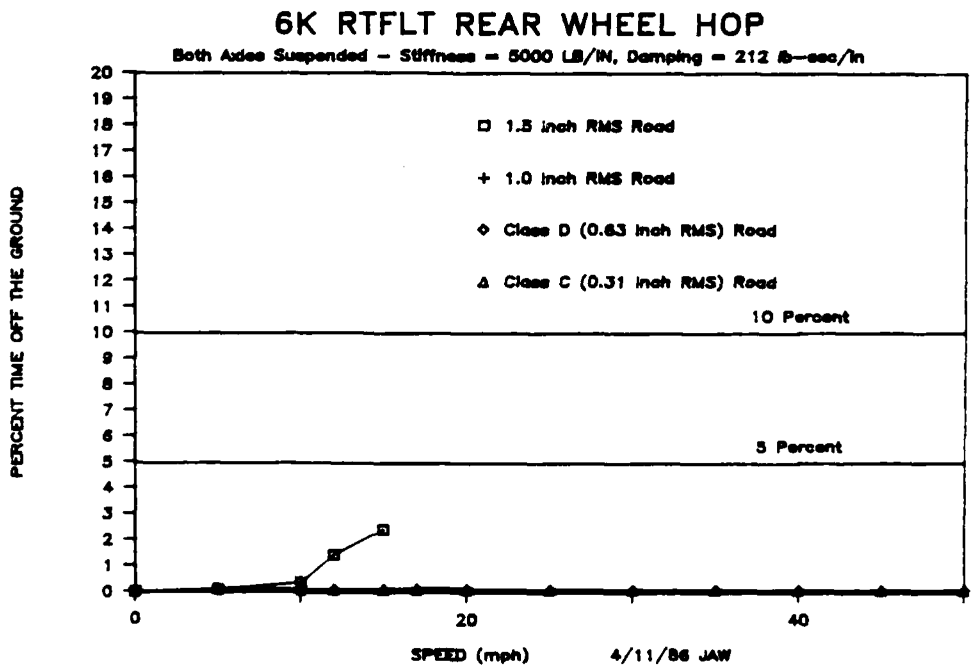

Figure 69. Rear Wheel Hop on Four Road Surfaces (6K RTFLT Both Axles Suspended)

The suspended linkage option without any axle suspension meets the 6 watt limit at 45 mph on the class C road, but slightly exceeds that limit at 35 mph (Fig. 70). This option greatly exceeds the 6 watt limit on the class D road at all speeds above 10 mph. However, the wheel hop is acceptable for all speeds on both the class C and class D roads (Fig. 71 and Fig. 72).

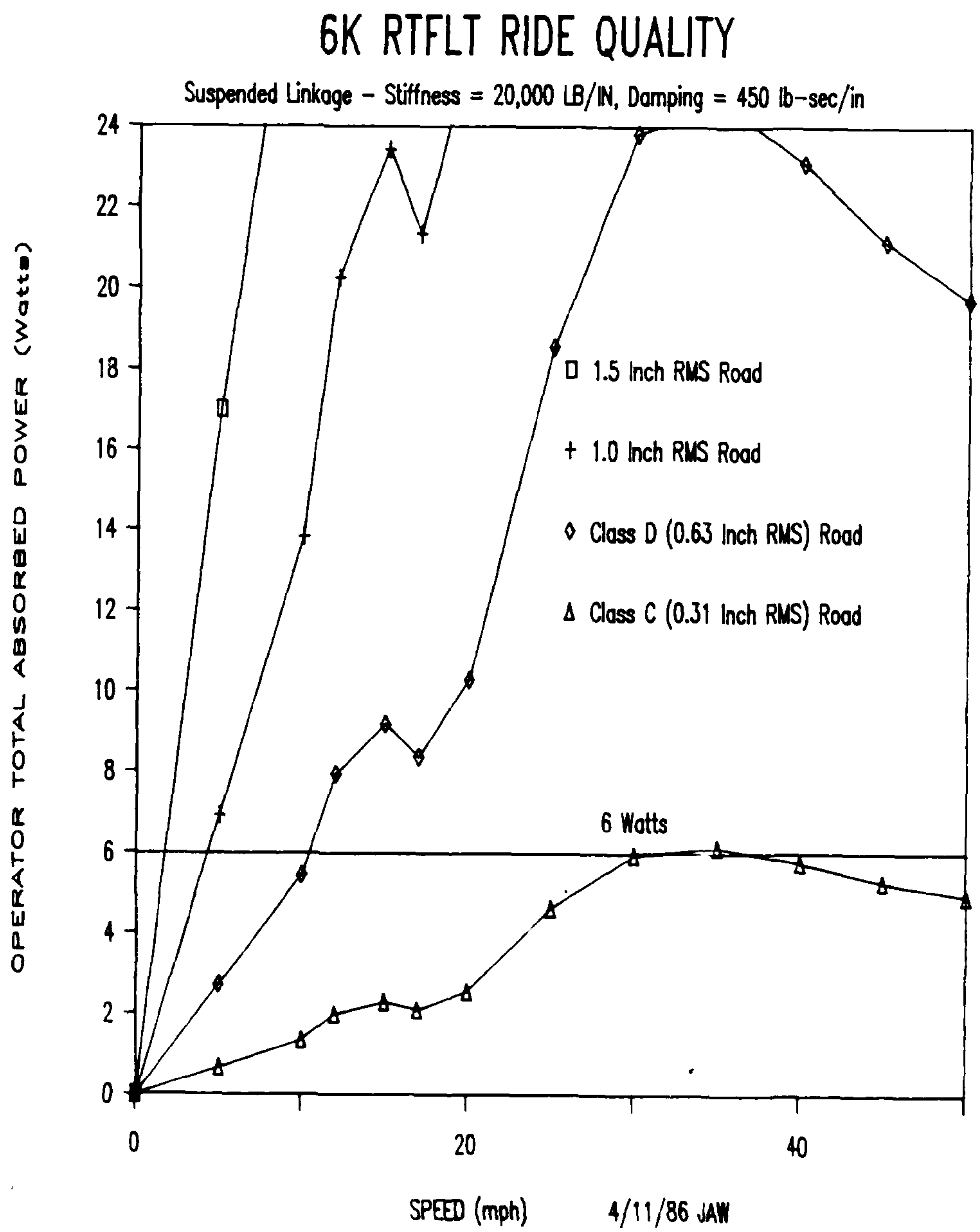

Figure 70. Ride Quality on Four Road Surfaces (6K RTFLT Suspended Linkage)

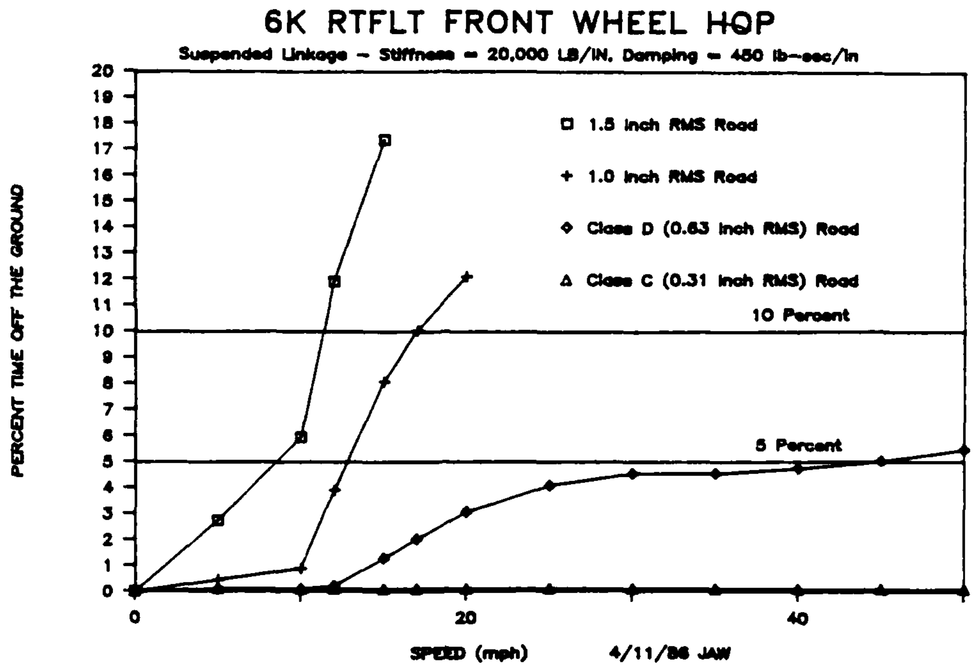

Figure 71. Front Wheel Hop on Four Road Surfaces (6K RTFLT Suspended Linkage)

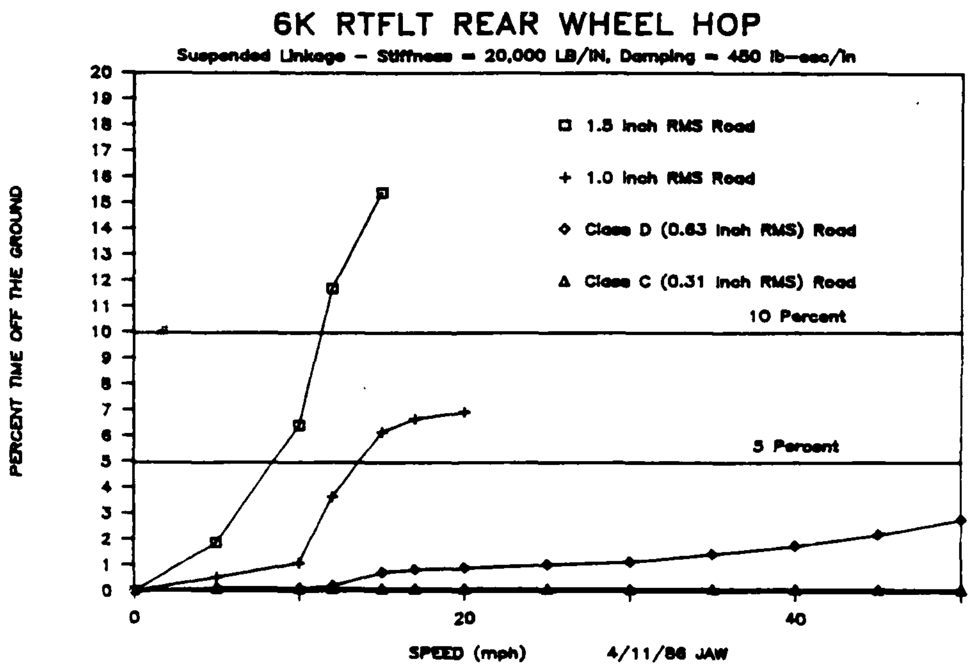

Figure 72. Rear Wheel Hop on Four Road Surfaces (6K RTFLT Suspended Linkage)

The 10K RTFLT unsuspended baseline vehicle exceeds the 6 watt limit above 15 mph on the class C road (Fig. 73). Wheel hop is acceptable on the class C road however. Front wheel hop is unacceptable on the class D road above 30 mph (Fig. 74). Rear wheel hop is acceptable at all speeds (Fig. 75).

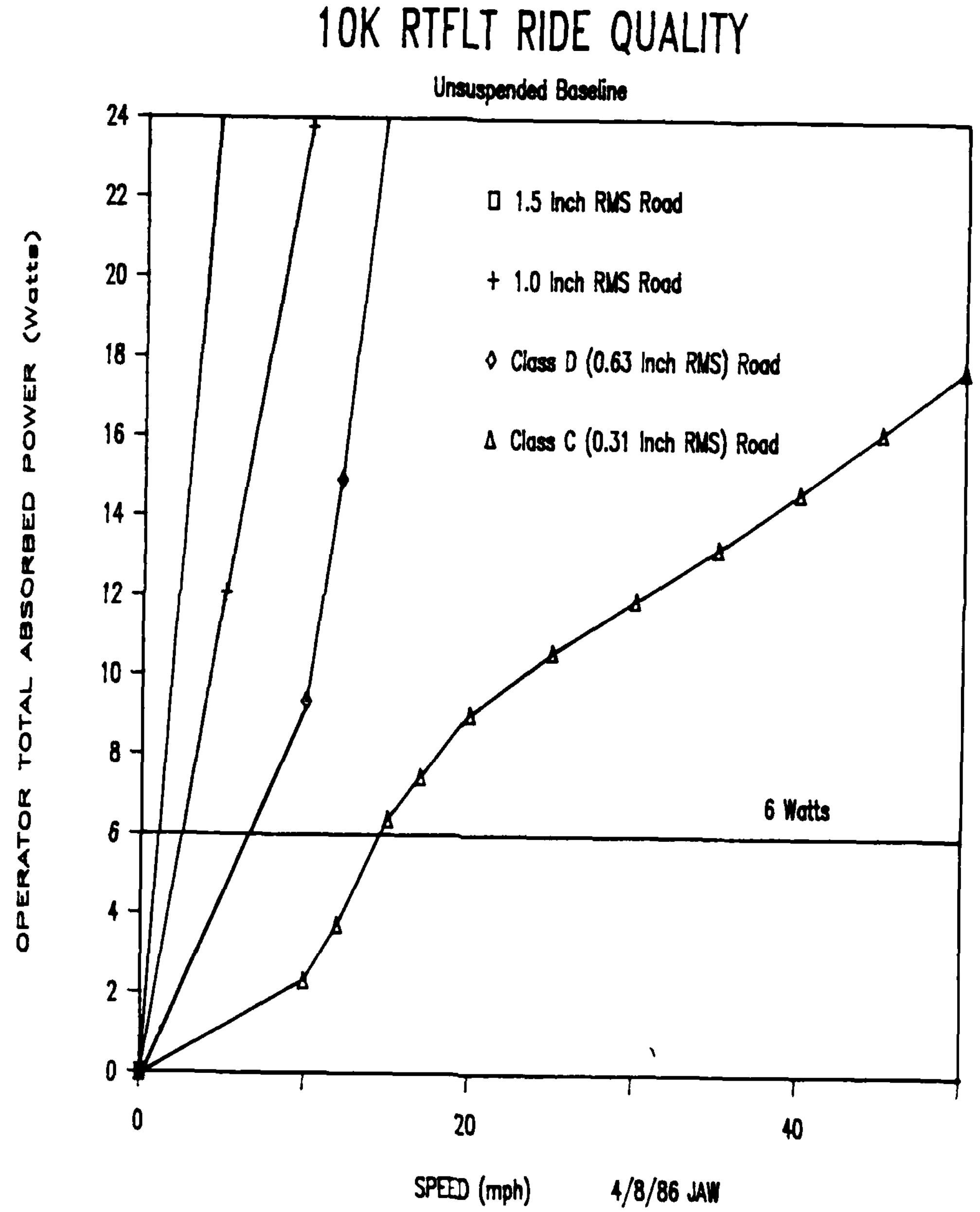

Figure 73. Ride Quality on Four Road Surfaces (10K RTFLT Unsuspended Baseline)

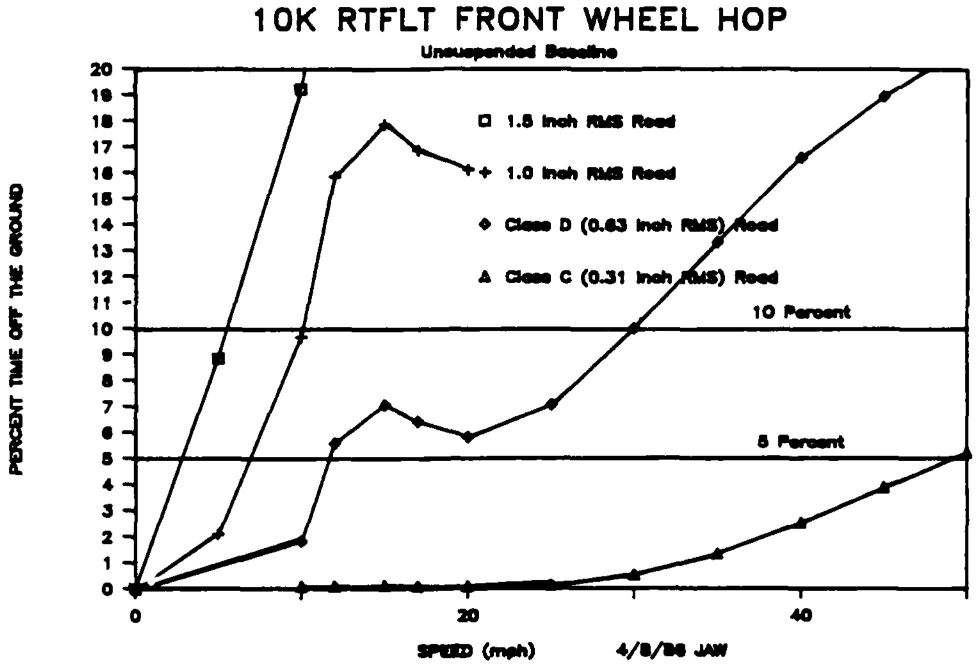

Figure 74. Front Wheel Hop on Four Road Surfaces (10K RTFLT Unsuspended Baseline)

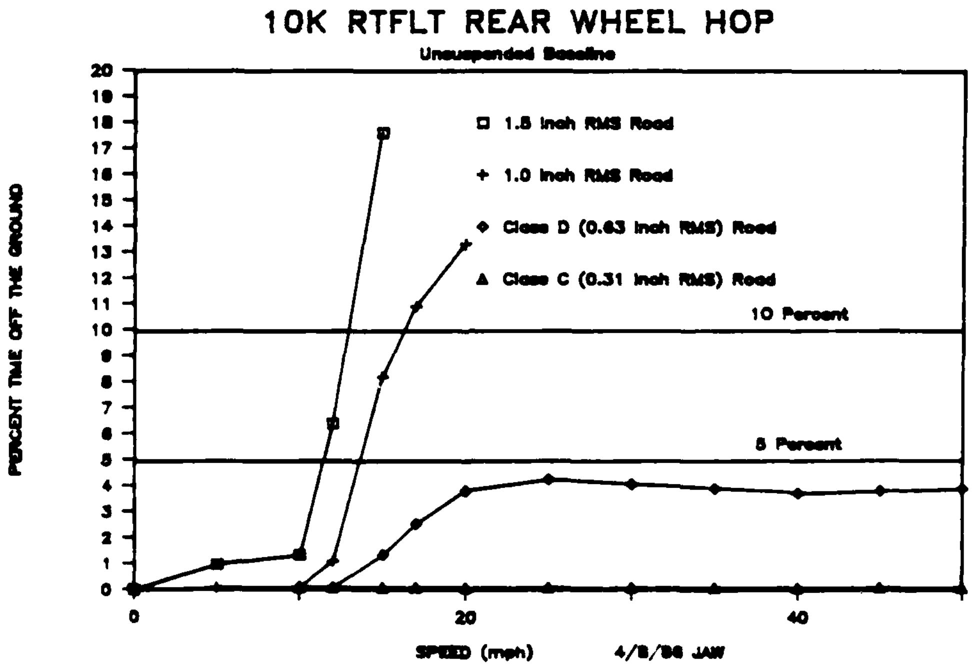

Figure 75. Rear Wheel Hop on Four Road Surfaces (10K RTFLT Unsuspended Baseline)

Adding the front axle suspension only to the 10K RTFLT provides acceptable ride (Fig. 76) and wheel hop (Fig. 77 and Fig. 78) at all speeds on the class C road. The desired 6 watt limit is exceeded above 12 mph on the class D road. However, wheel hop is acceptable at all speeds.

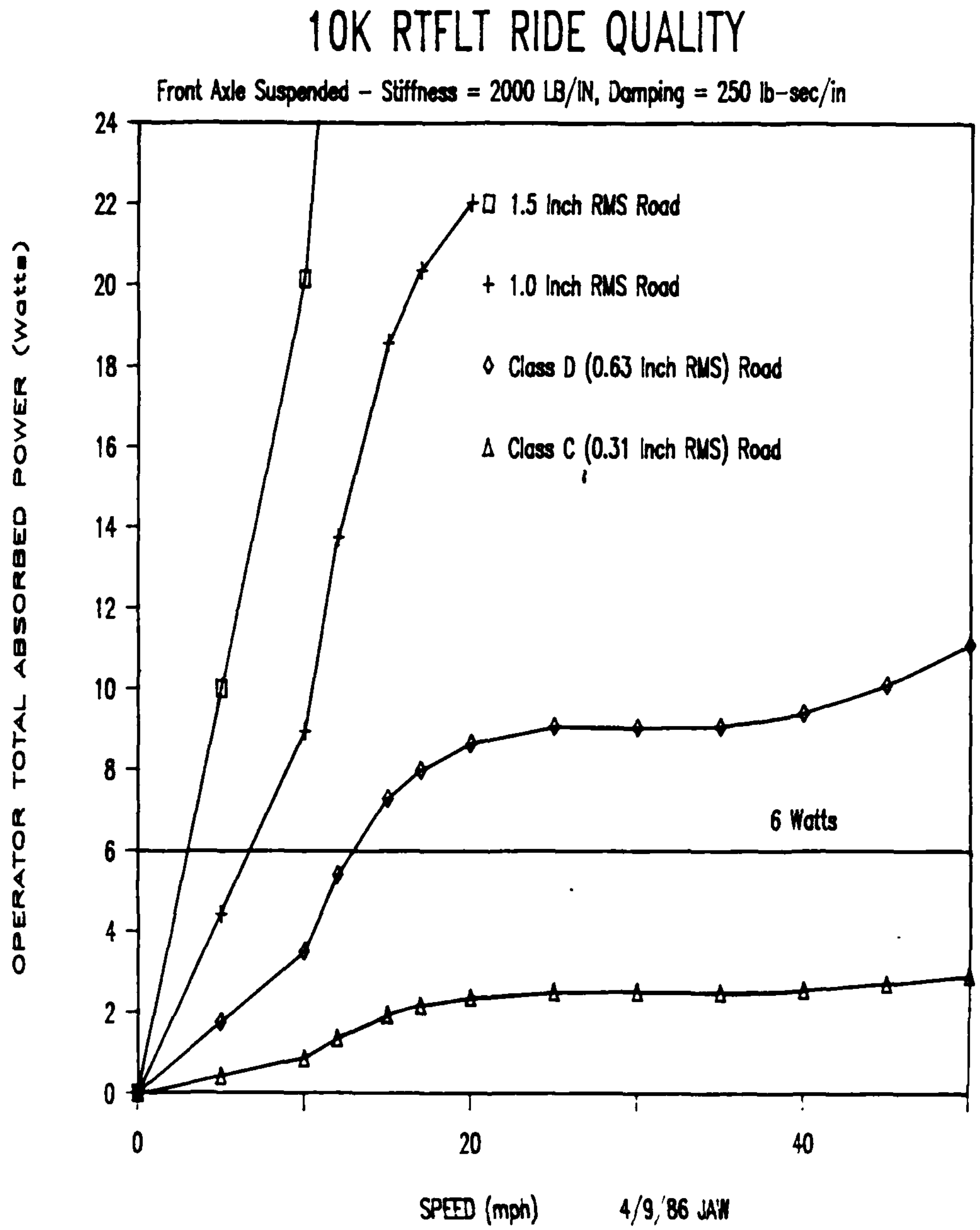

Figure 76. Ride Quality on Four Road Surfaces (10K RTFLT Front Axle Suspended)

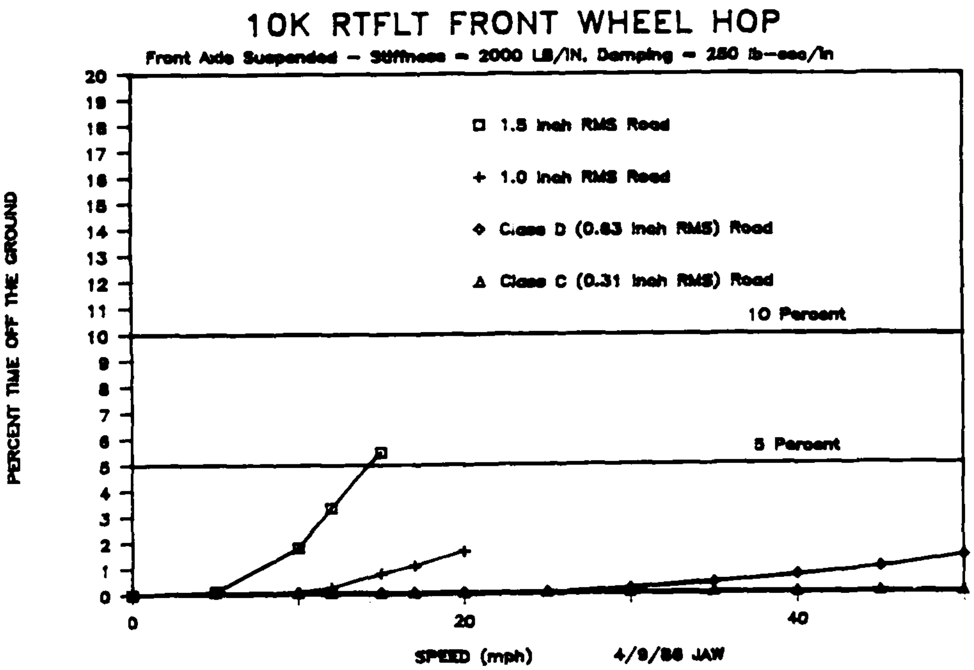

Figure 77. Front Wheel Hop on Four Road Surfaces (10K RTFLT Front Axle Suspended)

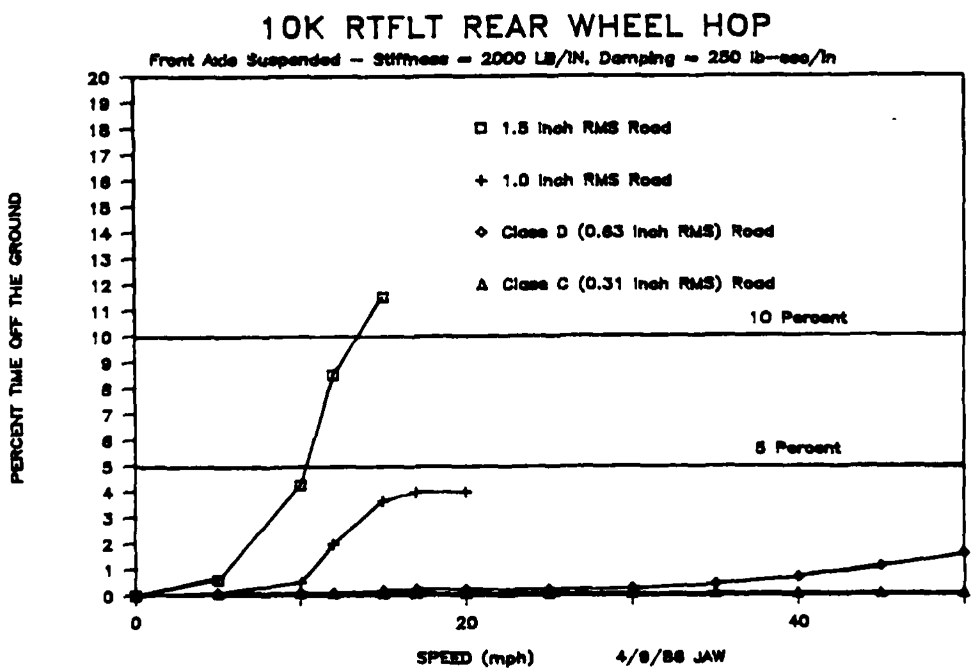

Figure 78. Rear Wheel Hop on Four Road Surfaces (10K RTFLT Front Axle Suspended)

The addition of the rear axle suspension as well as a front axle suspension to the 10K RTFLT improves the ride quality somewhat over the front axle suspension only (Fig. 79). However, the desired 6 watt limit is still exceeded on the class D road at 25 mph. Wheel hop is acceptable at all speeds on both the class C and class D roads (Fig. 80 and Fig. 81)

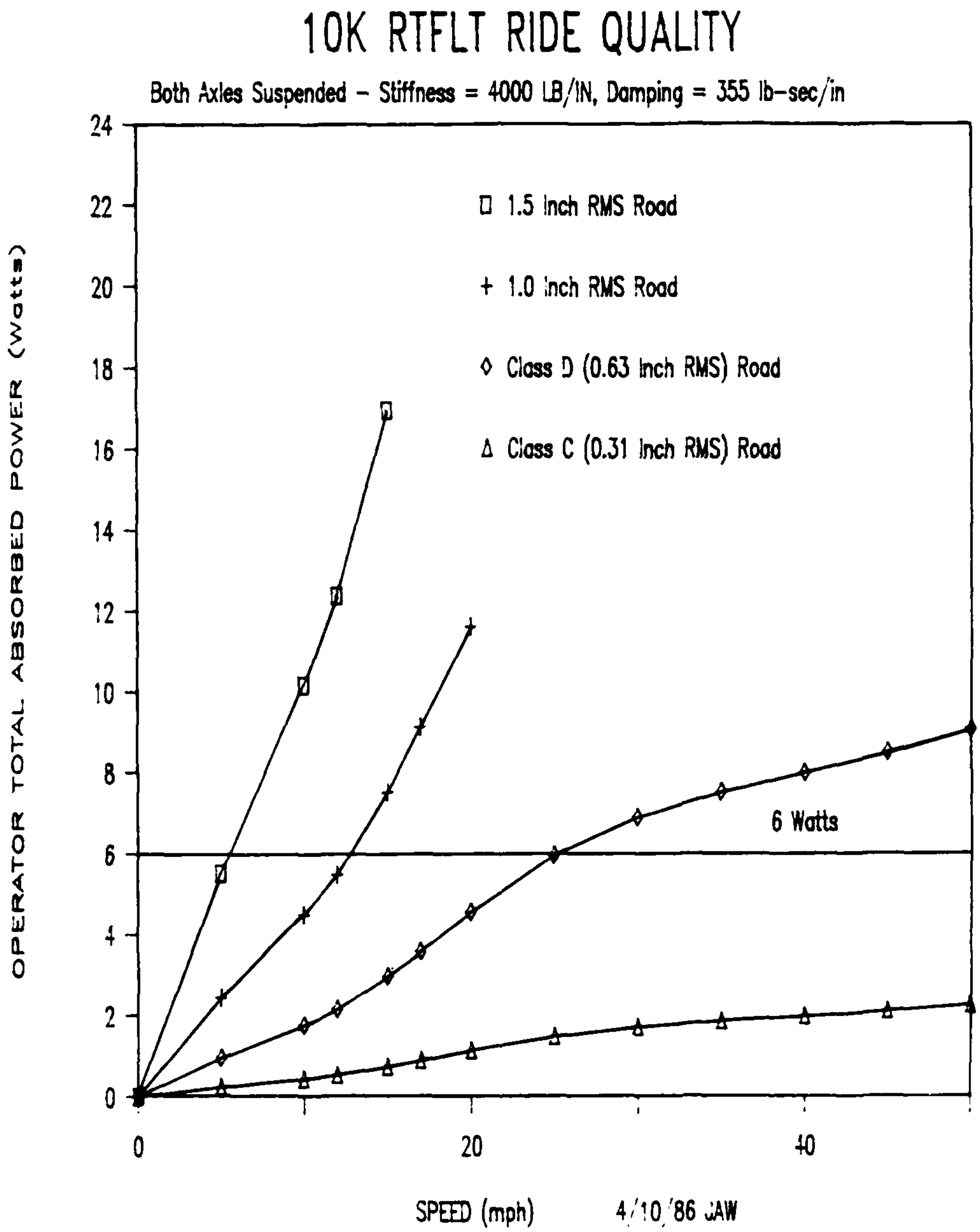

Figure 79. Ride Quality on Four Road Surfaces (10K RTFLT Both Axles Suspended)

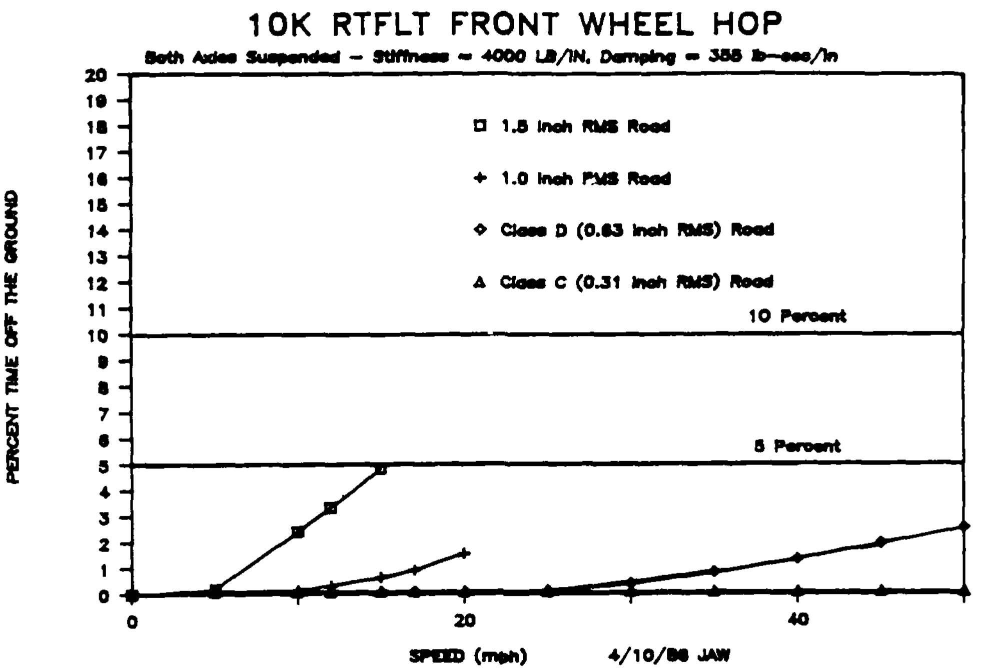

Figure 80. Front Wheel Hop on Four Road Surfaces (10K RTFLT Both Axles Suspended)

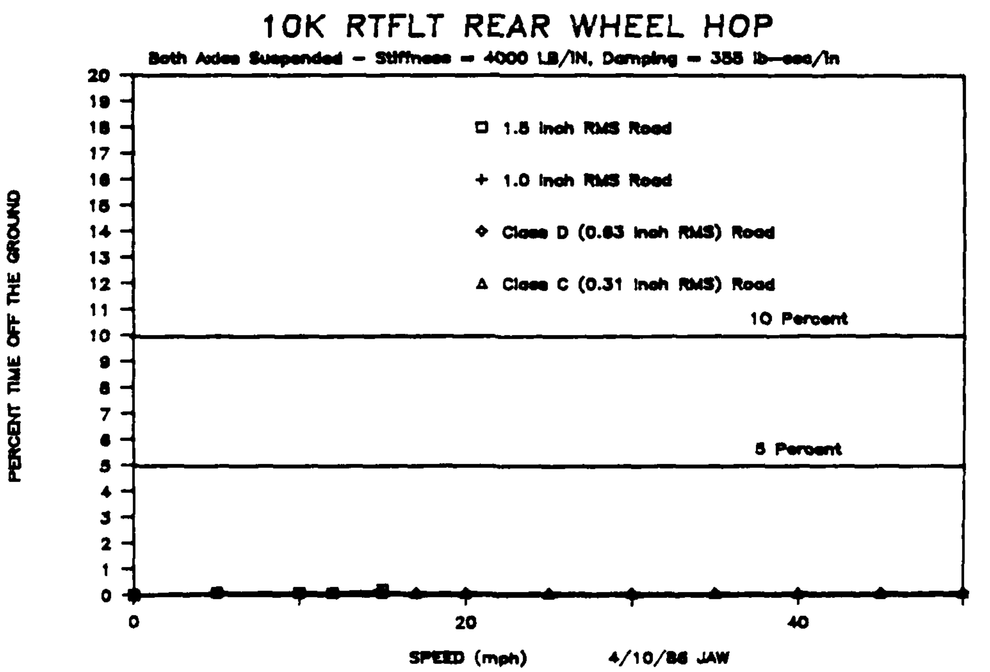

Figure 81. Rear Wheel Hop on Four Road Surfaces (10K RTFLT Both Axles Suspended)

The addition of a suspended counterweight with no axle suspensions does not meet the requirements as suggested in section 2.2.3.4. The 6 watt limit is exceeded at 40 mph on the class C road and 10 mph on the class D road (Fig. 82). Front wheel hop is exceeded at 35 mph on the class D road (Fig. 83). Rear wheel hop is acceptable however (Fig. 84).

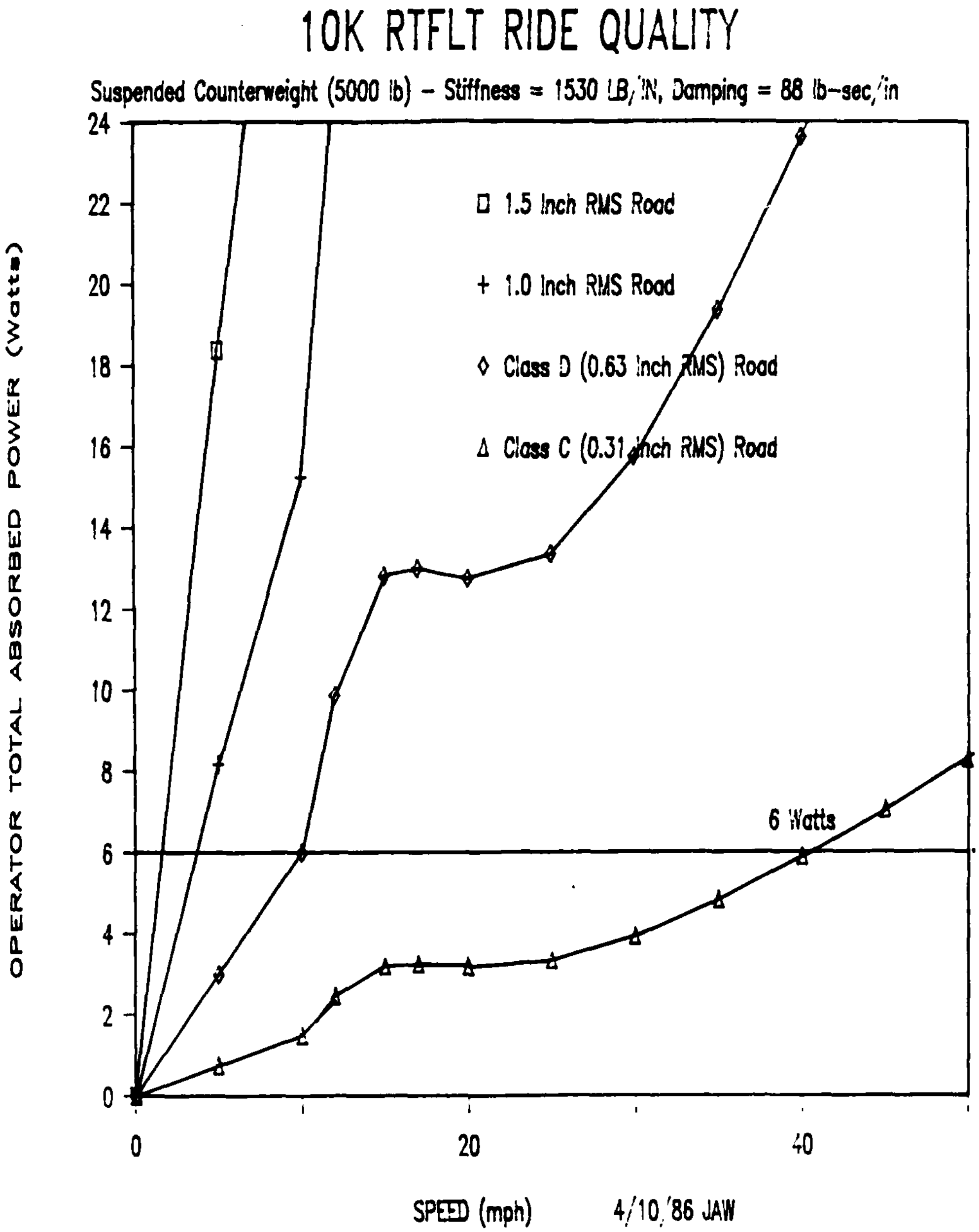

Figure 82. Ride Quality on Four Road Surfaces (10K RTFLT Suspended Rear Counterweight)

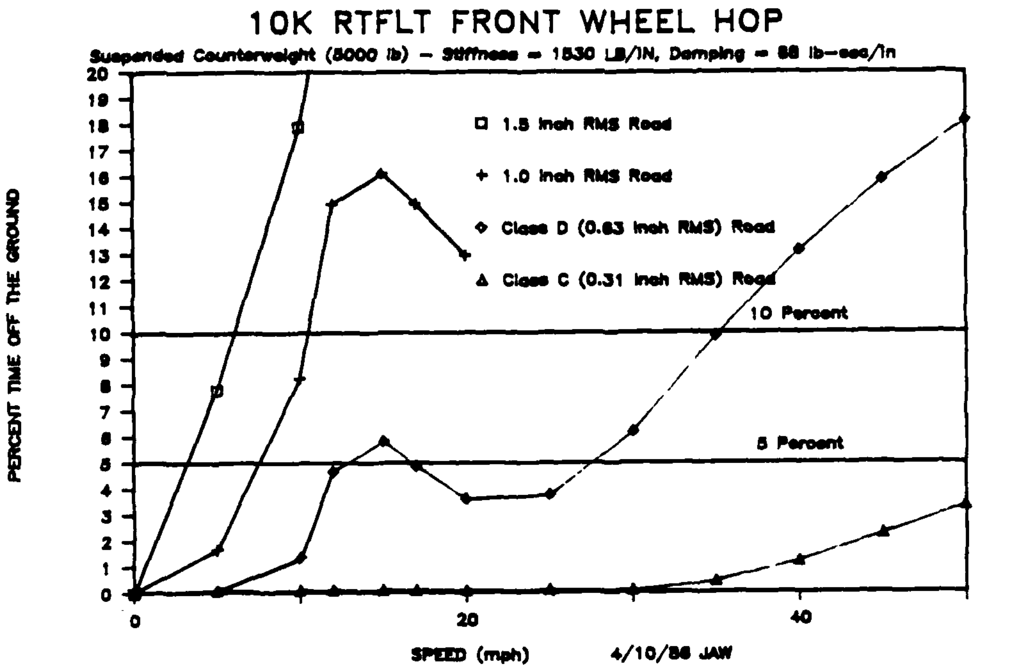

Figure 83. Front Wheel Hop on Four Road Surfaces (10K RTFLT Suspended Rear Counterweight)

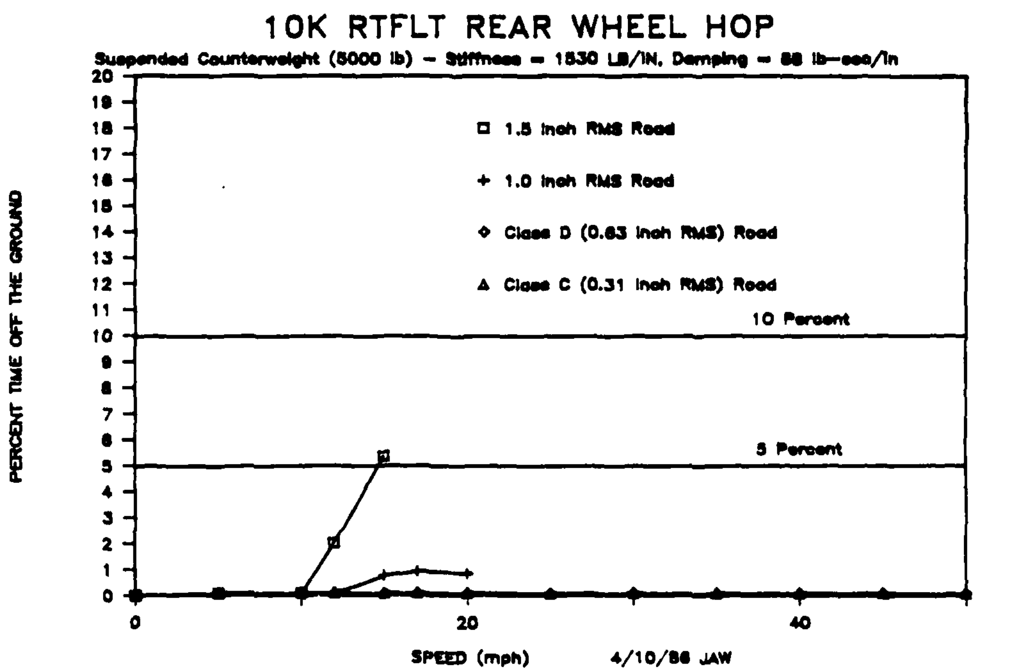

Figure 84. Rear Wheel Hop on Four Road Surfaces (10K RTFLT Suspended Rear Counterweight)

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2.2.3.6 Summary of Suspension Options with Ideally round Tires. The performance of the suspension options discussed in section 2.2.3.5 are summarized in Table 22 for the 6K RTFLT. The unsuspended baseline machine exceeds the wheel hop limit on the class D road. The suspended linkage option meets the class C 6 Watt limit at 45 mph but exceeds it slightly at lower speeds, and therefore is marginally acceptable. Both the front axle suspension only and the front and rear axle suspensions meet the class C road surface 6 Watt limit at 45 mph requirement and the wheel hop limit at 45 mph on both the class C and class D roads.

Table 22. 6K RTFLT Comparison of Suspension Alternatives With Ideally Round Tires

Suspension Option	Class C Road @ 45 mph			Class D Road @ 45 mph	
	Ride Watts	% Wheel Hop Front	% Wheel Hop Rear	% Wheel Hop Front	% Wheel Hop Rear
Unsuspended Baseline	3	4	0.2	\ / 19 / \	/ 7 /
Front Axle Suspended	2	0.1	0.1	5	0.4
Both Axles Suspended	0.5	0.1	0.1	3	0.1
Suspended Linkage	/ 5 /	0.1	0.1	5	2

KEY:
Total Absorbed Power Max Limit - 6 Watts
Wheel Hop / Controllability
- Acceptable - Less than 5% Time Off the Ground
- Marginal - 5-10% Time Off the Ground
- Unacceptable - Greater than 10% Time Off the Ground

[ ] = Acceptable [/ /] = Marginal [\ / / \ ] = Unacceptable

The 10K RTFLT suspension options are summarized in Table 23. Both the unsuspended baseline vehicle and the suspended counterweight only vehicle exceed the ride quality requirement on the class C road and the front wheel hop limit on the class D road. The front axle suspension option and the front and rear axle suspension option both meet the 45 mph requirements.

Table 23. 10K RTFLT Comparison of Suspension Alternatives With Ideally Round Tires

Suspension Option	Class C Road @ 45 mph			Class D Road @ 45 mph	
	Ride Watts	% Wheel Hop Front	% Wheel Hop Rear	% Wheel Hop Front	% Wheel Hop Rear
Unsuspended Baseline	\ / 16 / \	4	0.1	\ / 19 / \	3
Front Axle Suspended	3	0.1	0.1	1	1
Both Axles Suspended	2	0.1	0.1	2	0.1
Suspended Counter-weight	\ / 7 / \	2	0.1	\ / 16 / \	0.1

KEY:

Total Absorbed Power Max Limit - 6 Watts

Wheel Hop / Controllability

Acceptable - Less than 5% Time Off the Ground

Marginal - 5-10% Time Off the Ground

Unacceptable - Greater than 10% Time Off the Ground

[ ] = Acceptable [/ /] = Marginal [\ / / \ ] = Unacceptable

2.2.3.7 Summary of suspension options with out-of-round tires. The results discussed on 2.2.3.5. were all based on ideally round tires. Real world, out-of-round tires can impact ride quality and wheel hop. This section covers effects of out-of-round tires on the performance of some of the suspension options discussed in 2.2.3.5. Tire out-of-roundness is actually radial runout, but can be expressed in terms of an equivalent force. For example, if a out-of-round tire is rolled at constant rolling radius, the radial force on the tire will vary around the circumference of the tire. The difference between the largest and smallest radial force around the circumference of the tire is a measure of the degree of out-of-roundness of the tire.

In this study, three tire out-of-round variations were studied. They are ideally round, 175 pound equivalent out-of-round, and 500 pound equivalent out-of-round. Information from tire manufacturers indicates that 500 pound equivalent out-of-roundness may occur in tires "out of the mold" of the size (17.5 X 25 or 20.5 X 25) and type required for the RTFLT. However, the out-of-roundness can be improved to 175 pounds equivalent by grinding.

Fig. 85 shows the effects of out-of-round tires on the 6K RTFLT unsuspended baseline machine ride quality on a class C road. The acceptable 6 Watt limit is greatly exceeded if 500 lb. equivalent out-of-round tires are on the vehicle. The out-of-round tires rotate at a frequency equal to the bounce mode of the vehicle at 22 mph and result in the large spike in the curve. Wheel hop also becomes unacceptable at 22 mph (Fig. 86 and Fig. 87).

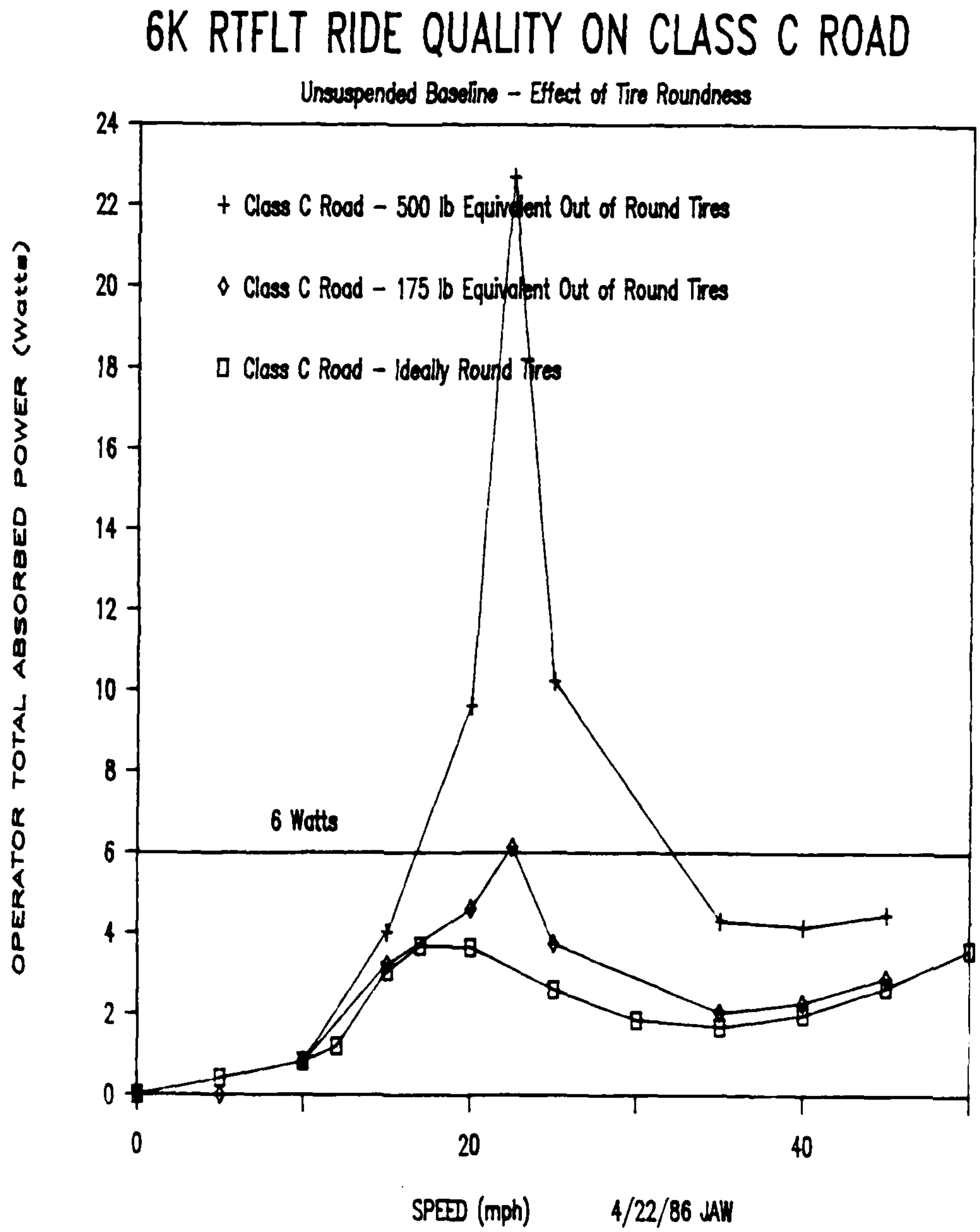

Figure 85. Effect of Tire Roundness on Ride Quality on Class C Road (6K RTFLT Unsuspended Baseline)

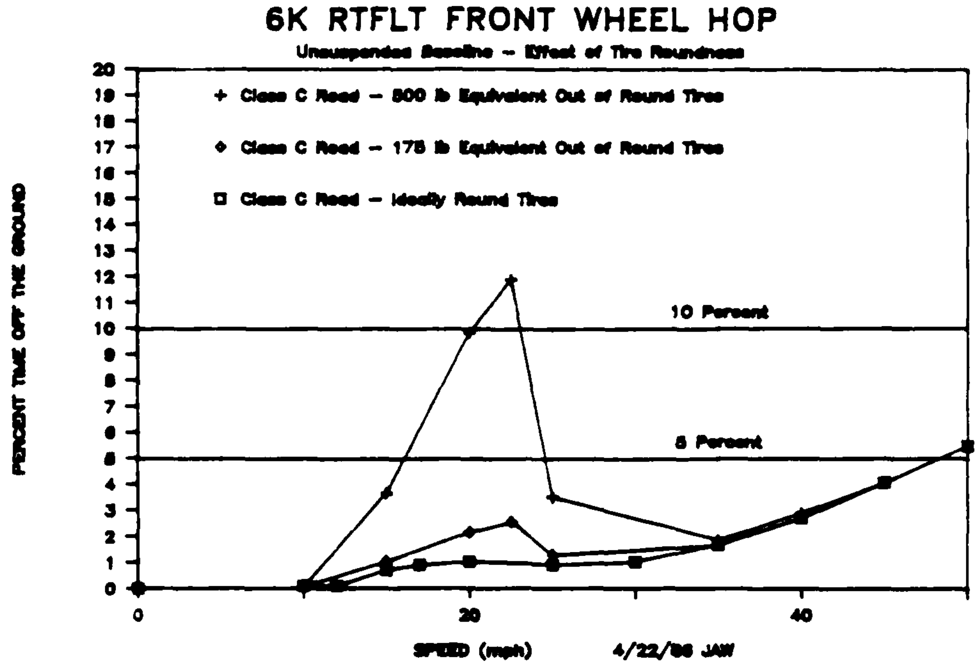

Figure 86. Effect of Tire Roundness on Front Wheel Hop on Class C Road (6K RTFLT Unsuspended Baseline)

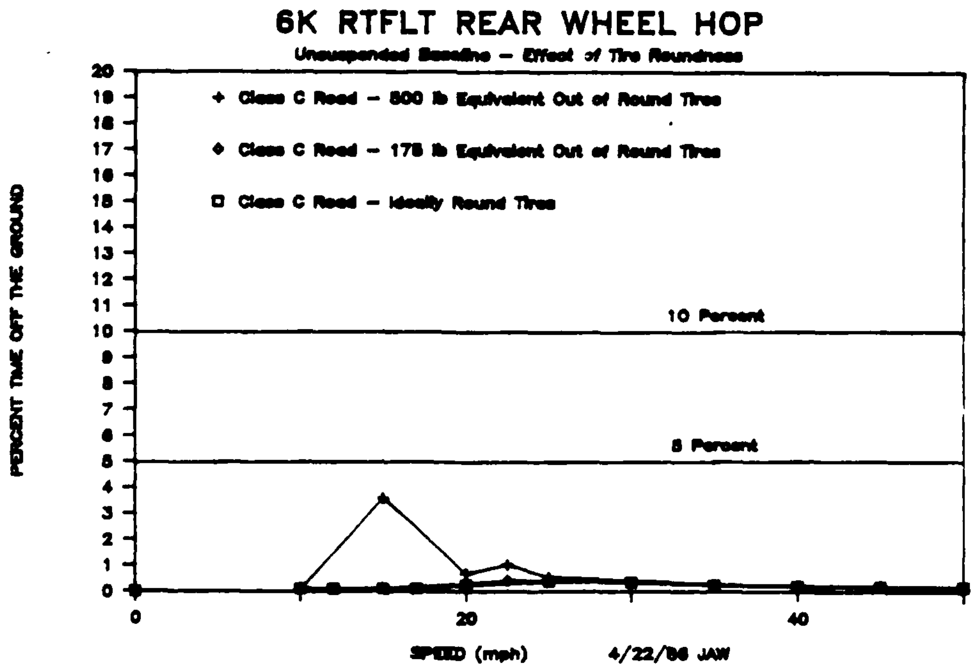

Figure 87. Effect of Tire Roundness on Rear Wheel Hop on Class C Road (6K RTFLT Unsuspended Baseline)

The 500 lb. equivalent out-of-round tires also cause the front axle suspension version to exceed the 6 Watt limit at 18 mph (Fig. 88). However, with 175 pound equivalent out-of-round tires, the absorbed power limit is not exceeded. Wheel hop is acceptable in all cases for the front suspension option (Fig. 89 and Fig. 90).

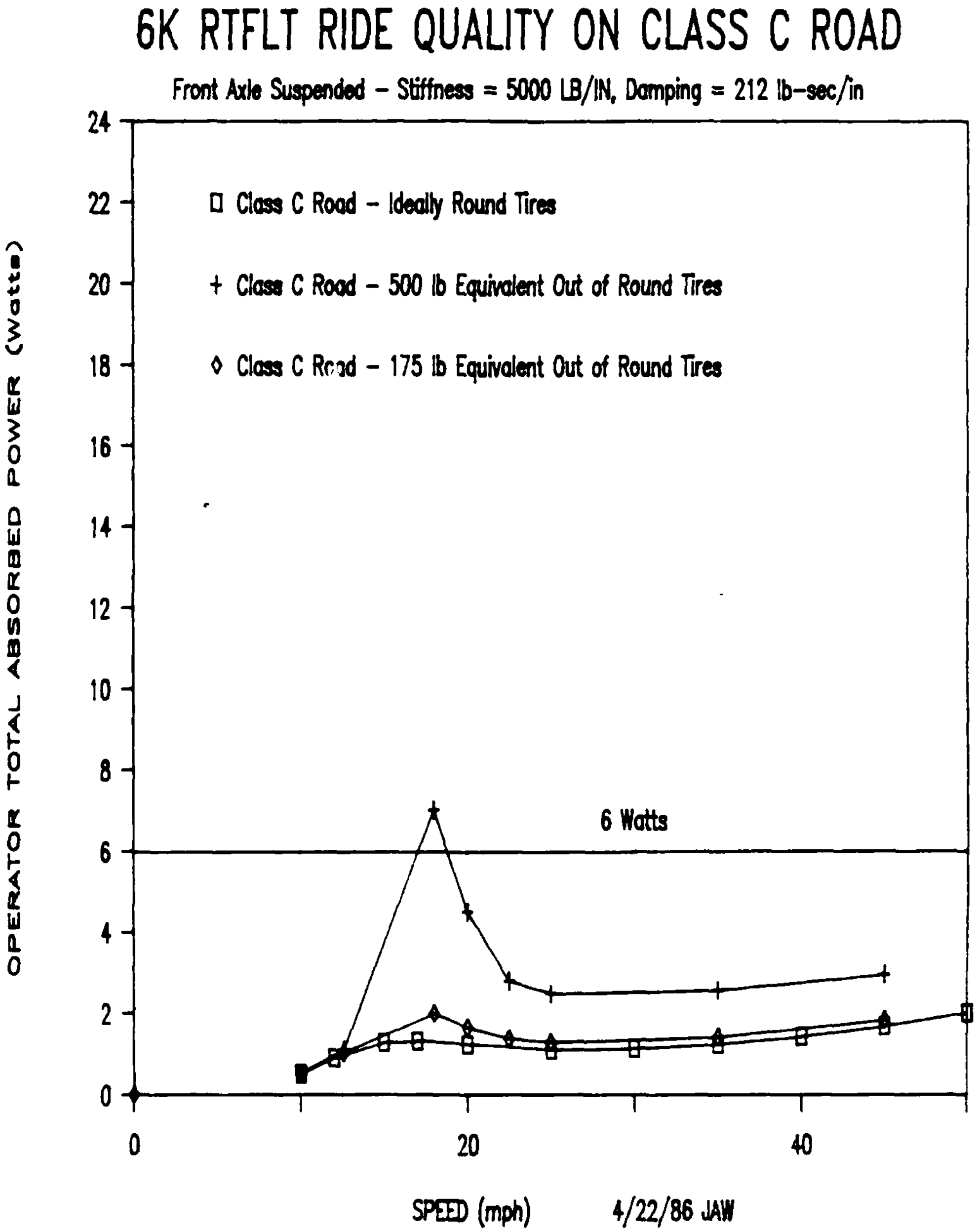

Figure 88. Effect of Tire Roundness on Ride Quality on Class C Road (6K RTFLT Front Axle Suspended)

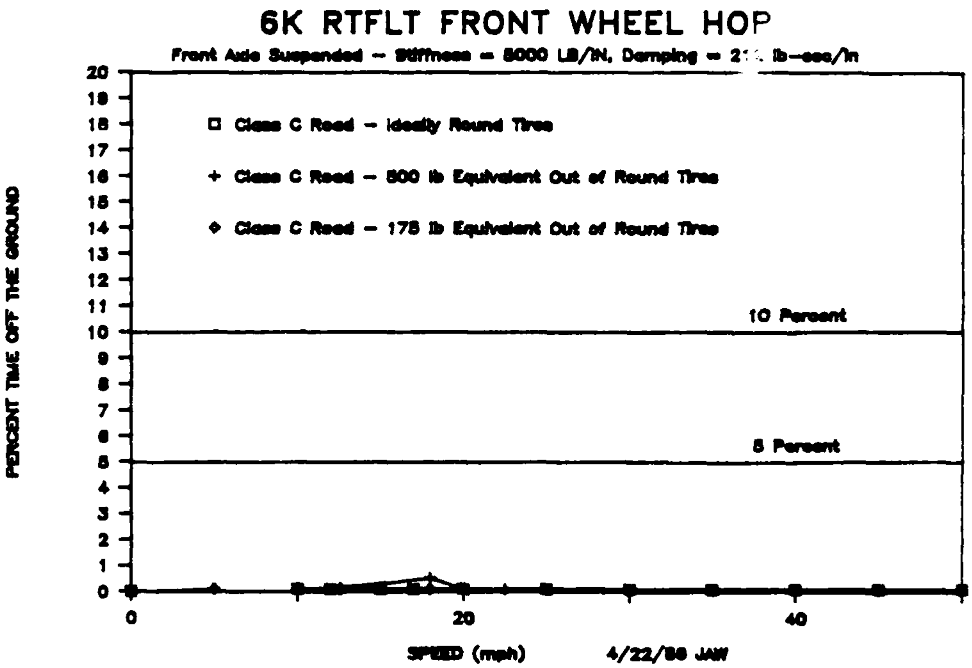

Figure 89. Effect of Tire Roundness on Front Wheel Hop on Class C Road (6K RTFLT Front Axle Suspended)

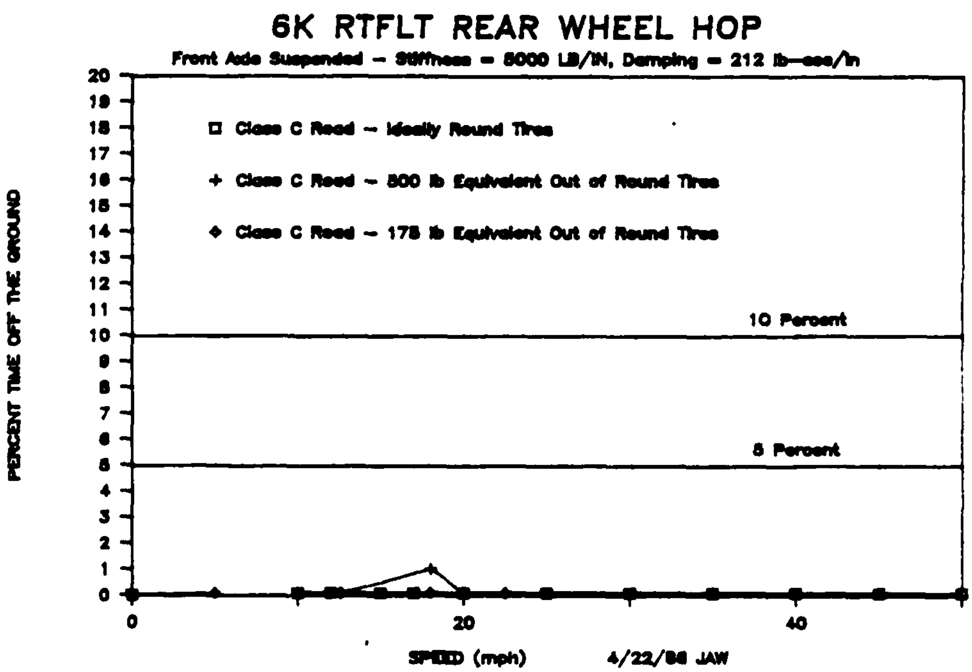

Figure 90. Effect of Tire Roundness on Rear Wheel Hop on Class C Road (6K RTFLT Front Axle Suspended)

Tire out-of-roundness has little effect on the ride quality and wheel hop when both axles are suspended (Fig. 91, Fig. 92, and Fig. 93).

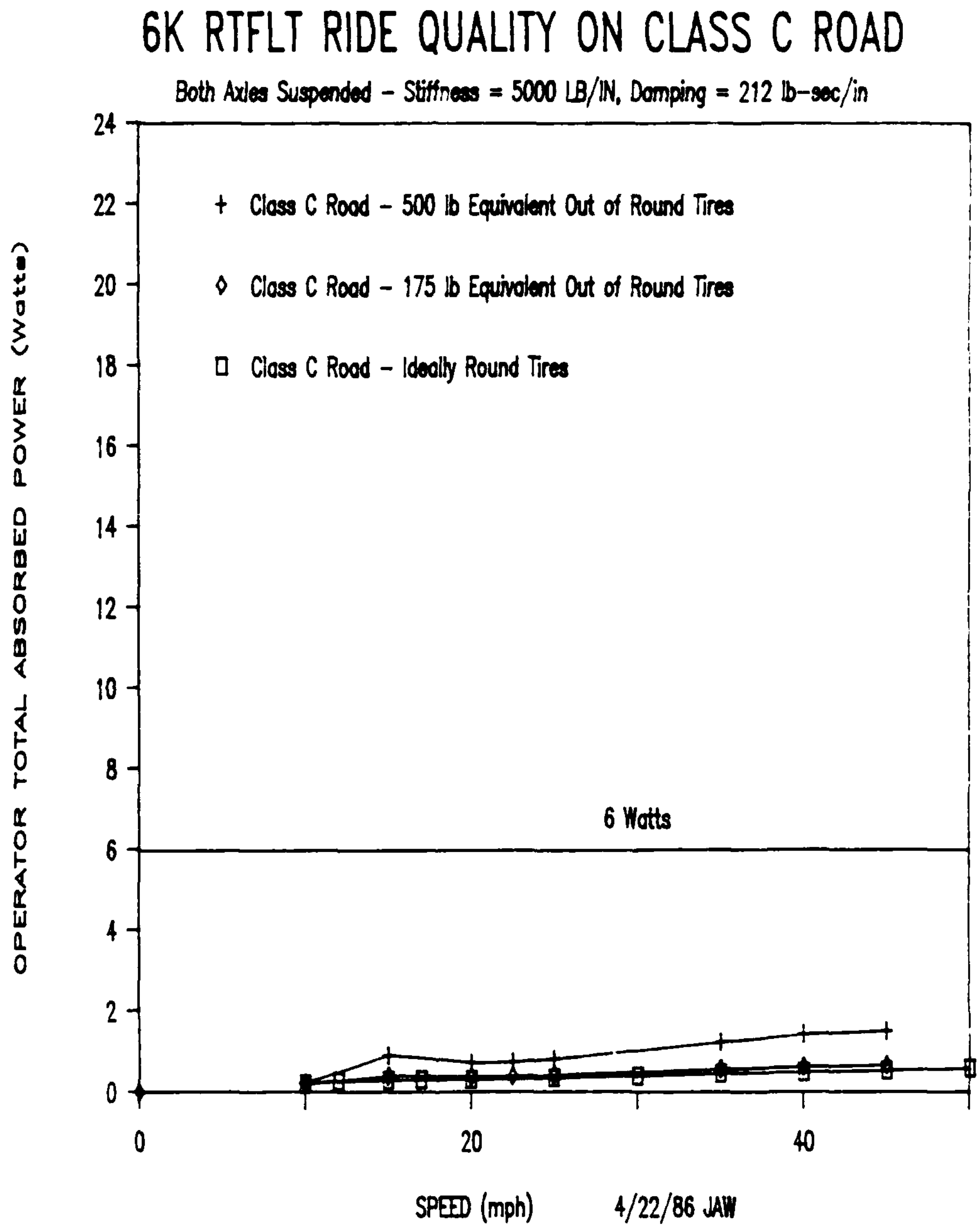

Figure 91. Effect of Tire Roundness on Ride Quality on Class C Road (6K RTFLT Both Axles Suspended)

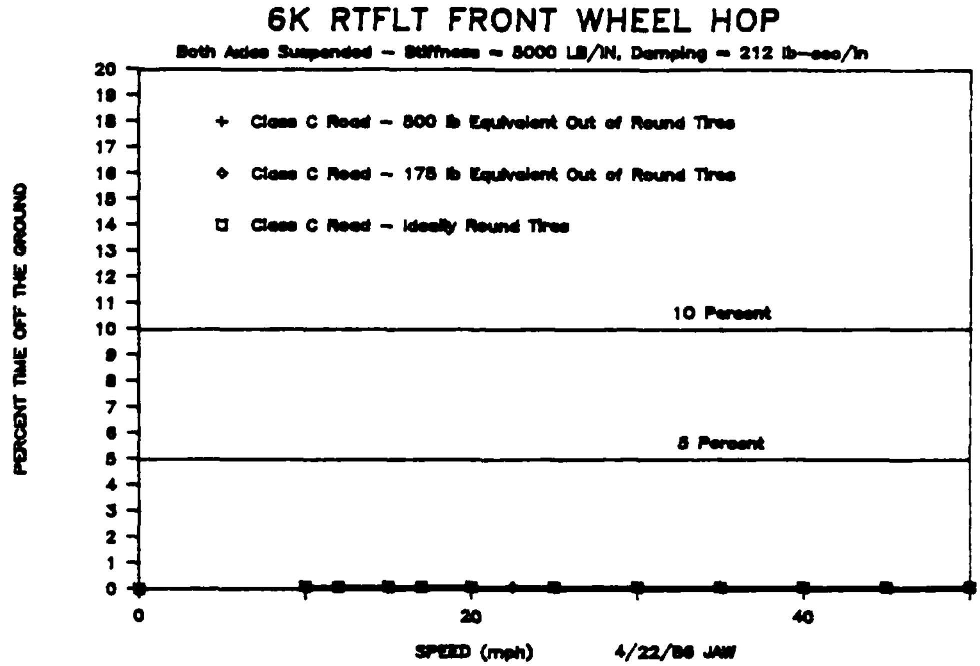

Figure 92. Effect of Tire Roundness on Front Wheel Hop on Class C Road (6K RTFLT Both Axles Suspended)

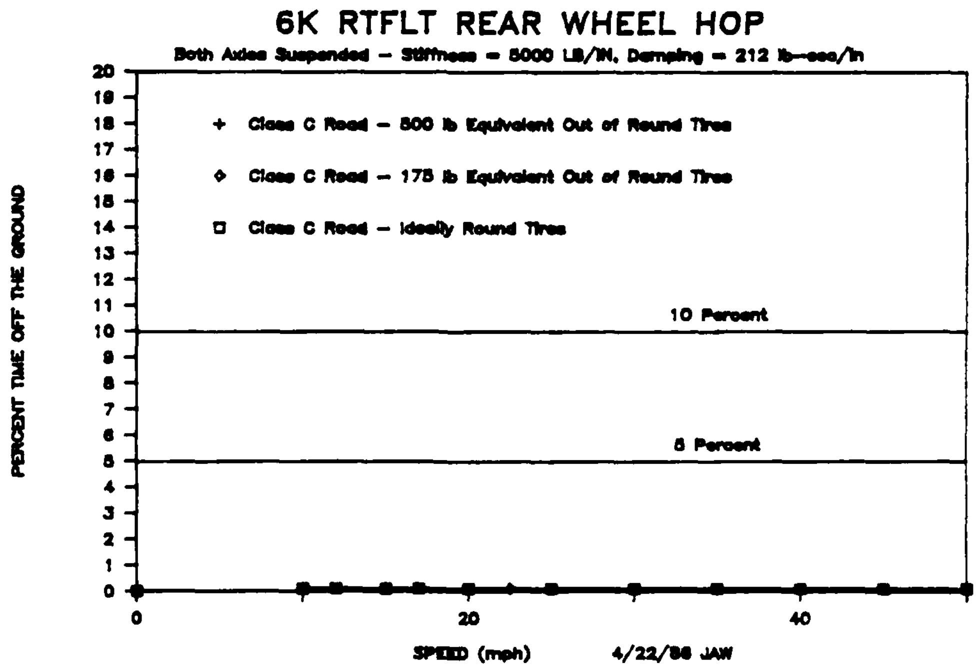

Figure 93. Effect of Tire Roundness on Rear Wheel Hop on Class C Road (6K RTFLT Both Axles Suspended)

The effects of tire out-of-roundness on the ride quality and wheel hop performance for the 6K RTFLT on the class C road is summarized in Table 24. The unsuspended baseline vehicle total operator absorbed power and front wheel hop are unacceptable with 500 pound equivalent out-of-round tires. The ride quality is marginal when the tires are improved to 175 pound equivalent out-of-roundness. The front axle suspension version exceeds the total absorbed power limit with 500 pound equivalent out-of-round tires. However, the both axles suspended option still meets the requirements on the class C road surface with 500 pound equivalent out-of-round tires.

Table 24. 6K RTFLT Summary of Suspension Alternatives with Out-of-Round Tires

Class C Road

Suspension Option	Peak Absorbed Power (Watts)			Front Wheel Hop Percent			Rear Wheel Hop Percent		
	Ideal Tires	175lb Out	500lb Out	Ideal Tires	175lb Out	500lb Out	Ideal Tires	175lb Out	500lb Out
Unsusp. Baseline @ 22.5 mph	3	/ 6 /	\ / 23 / \	1	2	\ / 12 / \	0.5	0.5	1
Front Axle Suspended @ 18 mph	1	2	\ / 7 / \	0.1	0.1	0.5	0.1	0.1	1
Both Axles Suspended @ 45 mph	0.5	0.6	2	0.1	0.1	0.1	0.1	0.1	0.1

KEY:

Tire Roundness (Applies to All Four Tires Equally)
- Ideal - Ideally Round Tires
- 175 lb - 175 lb Equivalent Out-of-Round Tires
- 500 lb - 500 lb Equivalent Out-of-Round Tires

Total Absorbed Power Max Limit - 6 Watts

Wheel Hop / Controllability
- Acceptable - Less than 5% Time Off the Ground
- Marginal - 5-10% Time Off the Ground
- Unacceptable - Greater than 10% Time Off the Ground

= Acceptable   / / = Marginal   \ / / \ = Unacceptable

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The 10K RTFLT unsuspended baseline vehicle is very adversely affected by out-of-round tires (Fig. 94, Fig. 95, and Fig. 96). The worst effects of tire out-of-roundness are seen at 20 mph.

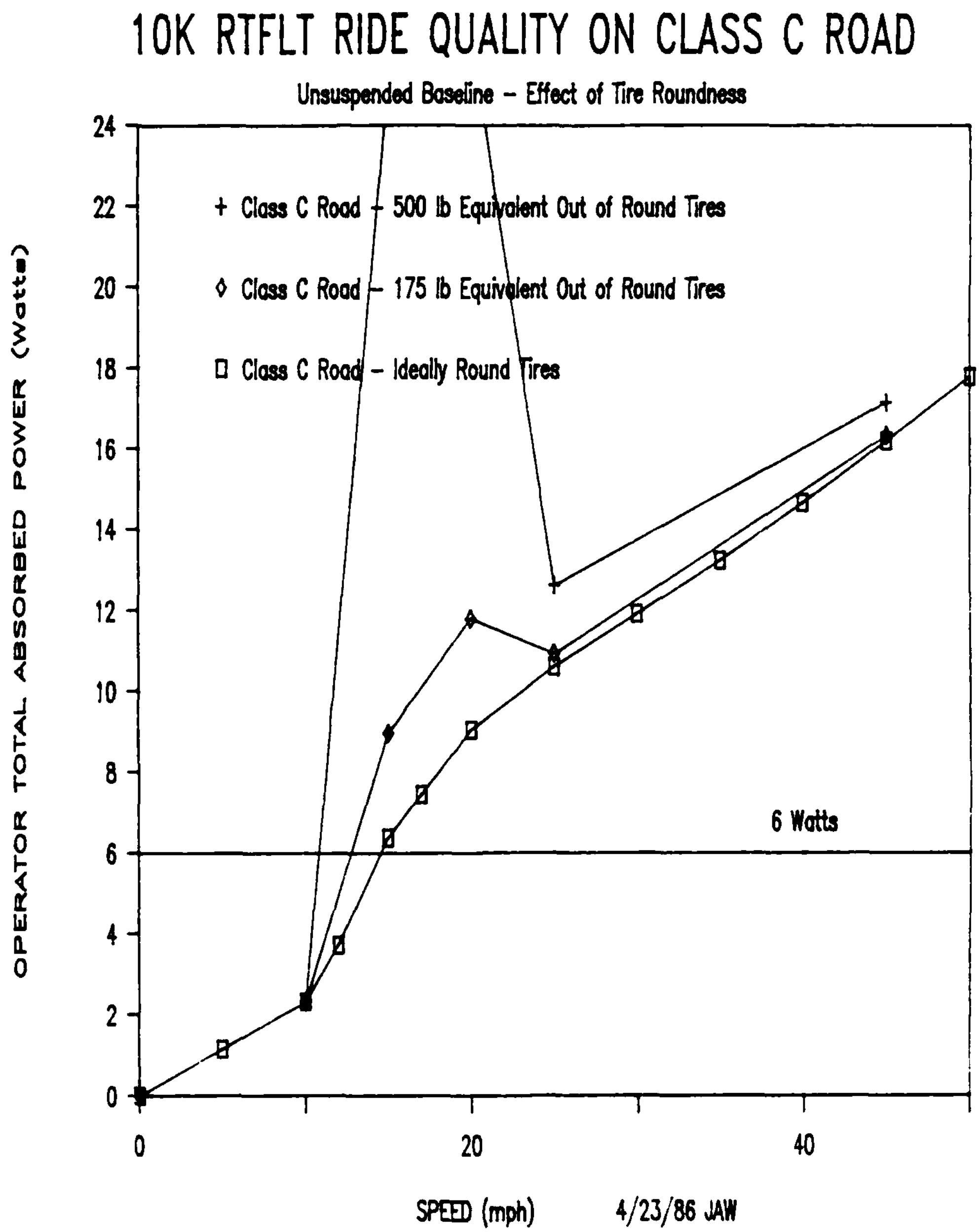

Figure 94. Effect of Tire Roundness on Ride Quality on Class C Road (10K RTFLT Unsuspended Baseline)

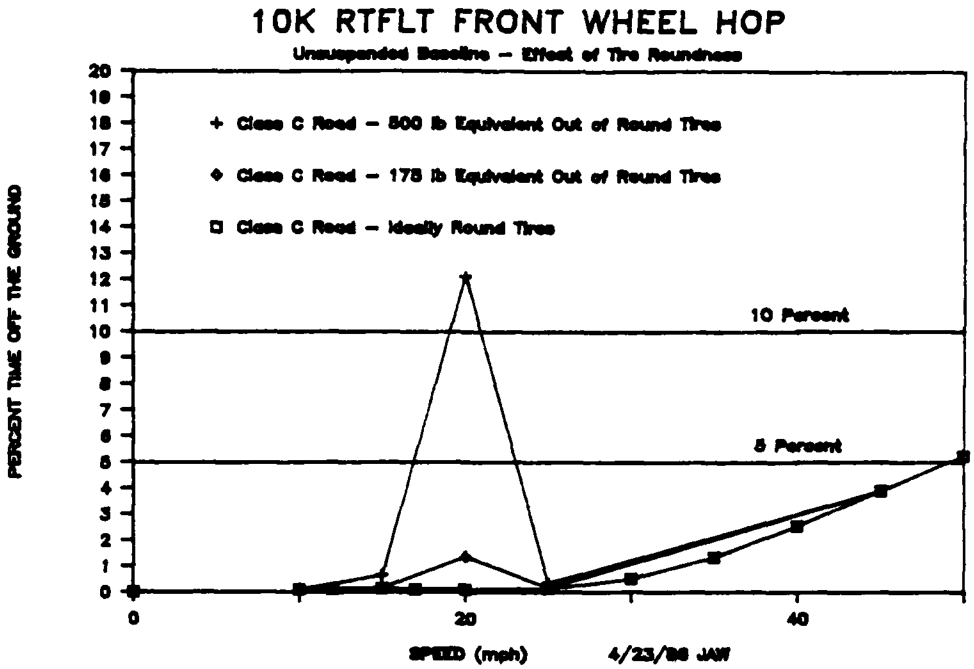

Figure 95. Effect of Tire Roundness on Front Wheel Hop on Class C Road (10K RTFLT Unsuspended Baseline)

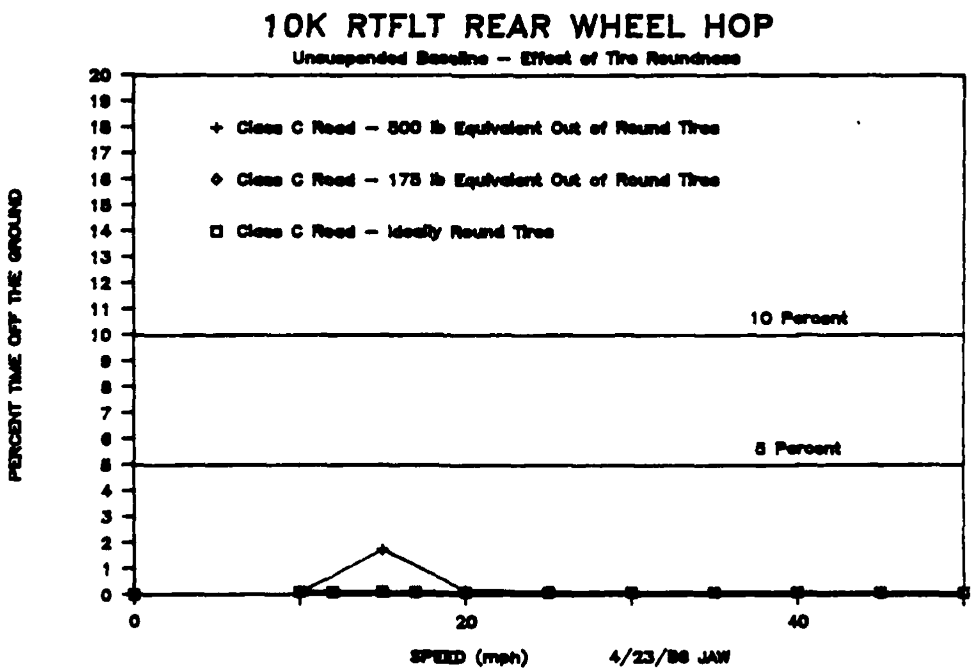

Figure 96. Effect of Tire Roundness on Rear Wheel Hop on Class C Road (10K RTFLT Unsuspended Baseline)

The front axle suspension version of the 10K RTFLT exceeds the total absorbed power limit at 15 mph with 500 pound equivalent out-of-round tires on the class C road (Fig. 97). However, wheel hop is still acceptable both front (Fig. 98) and rear (Fig. 99).

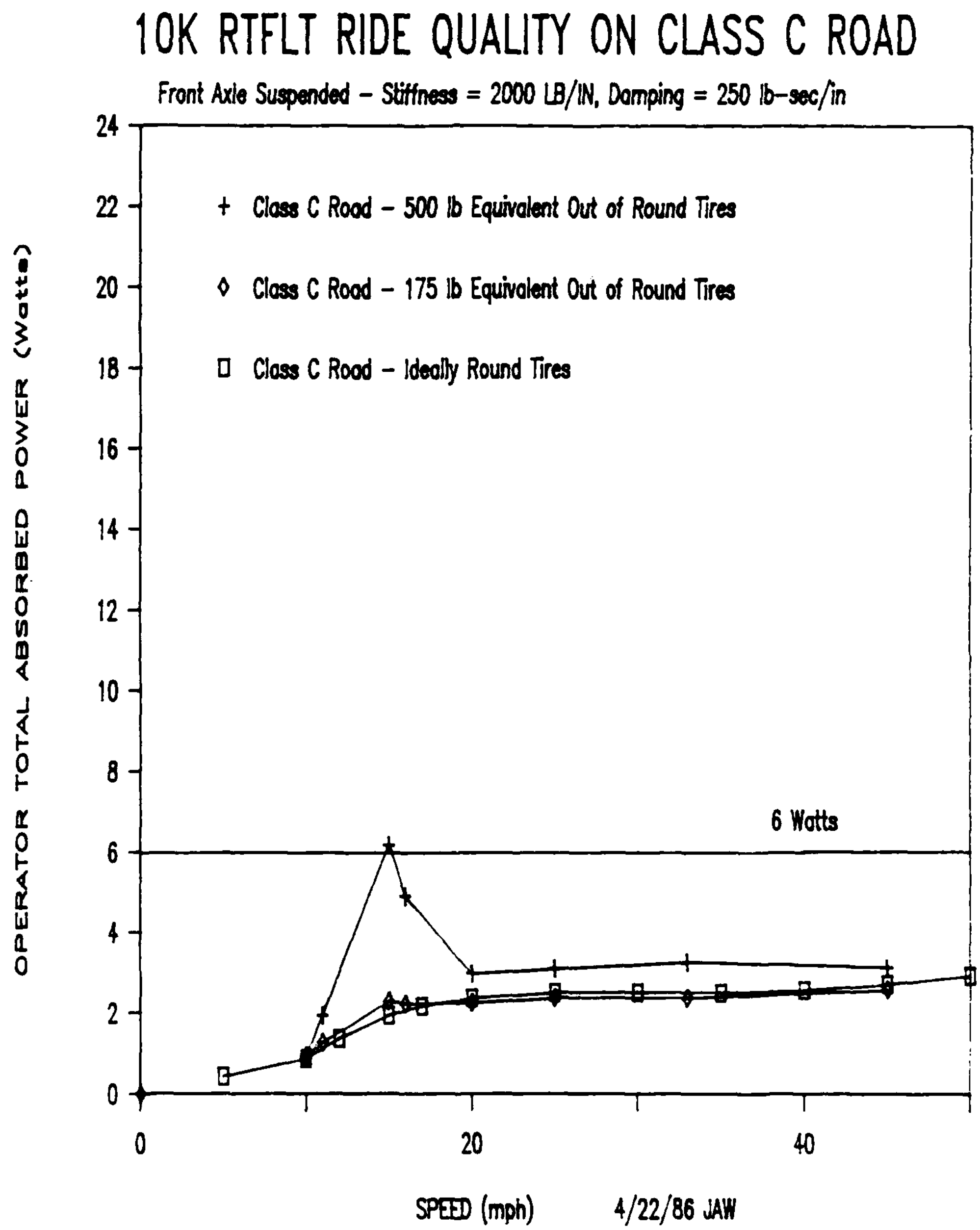

Figure 97. Effect of Tire Roundness on Ride Quality on Class C Road (10K RTFLT Front Axle Suspended)

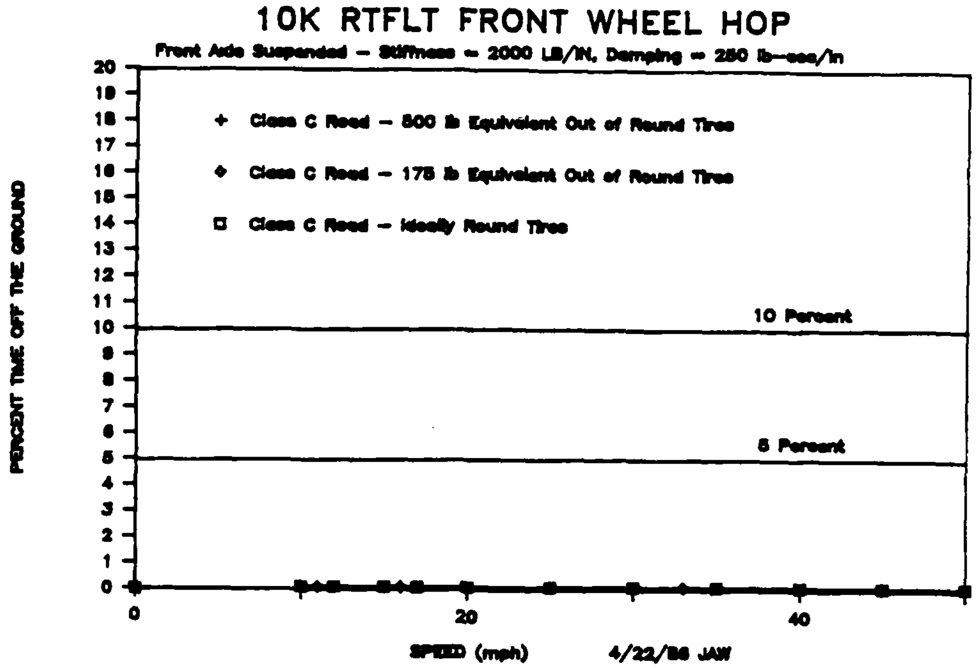

Figure 98. Effect of Tire Roundness on Front Wheel Hop on Class C Road (10K RTFLT Front Axle Suspended)

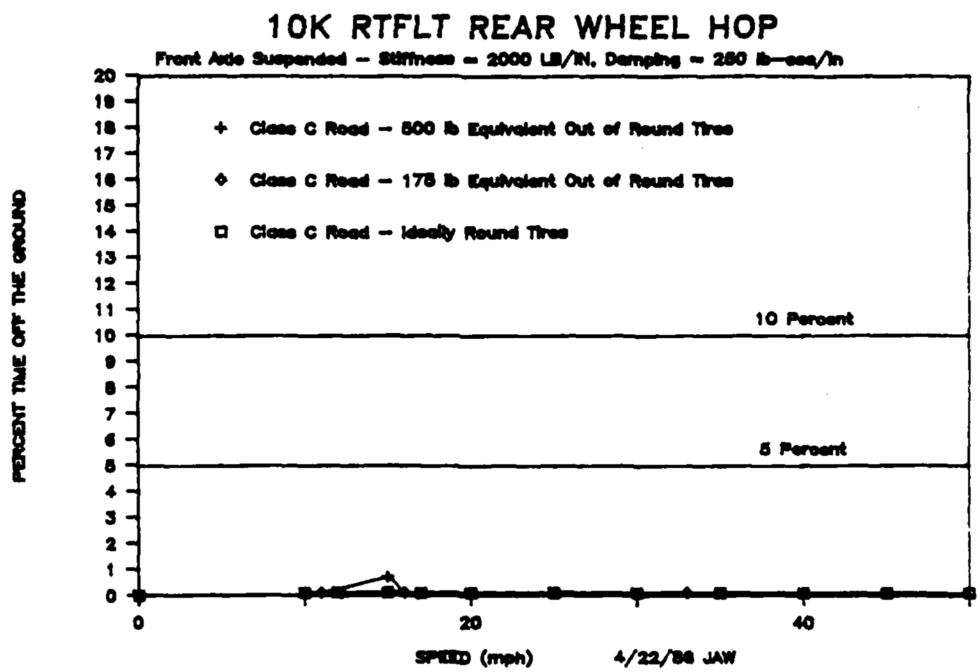

Figure 99. Effect of Tire Roundness on Rear Wheel Hop on Class C Road (10K RTFLT Front Axle Suspended)

Tire out-of-roundness has little effect on ride quality (Fig. 100) and wheel hop (Fig. 101 and Fig. 102) when both axles are suspended.

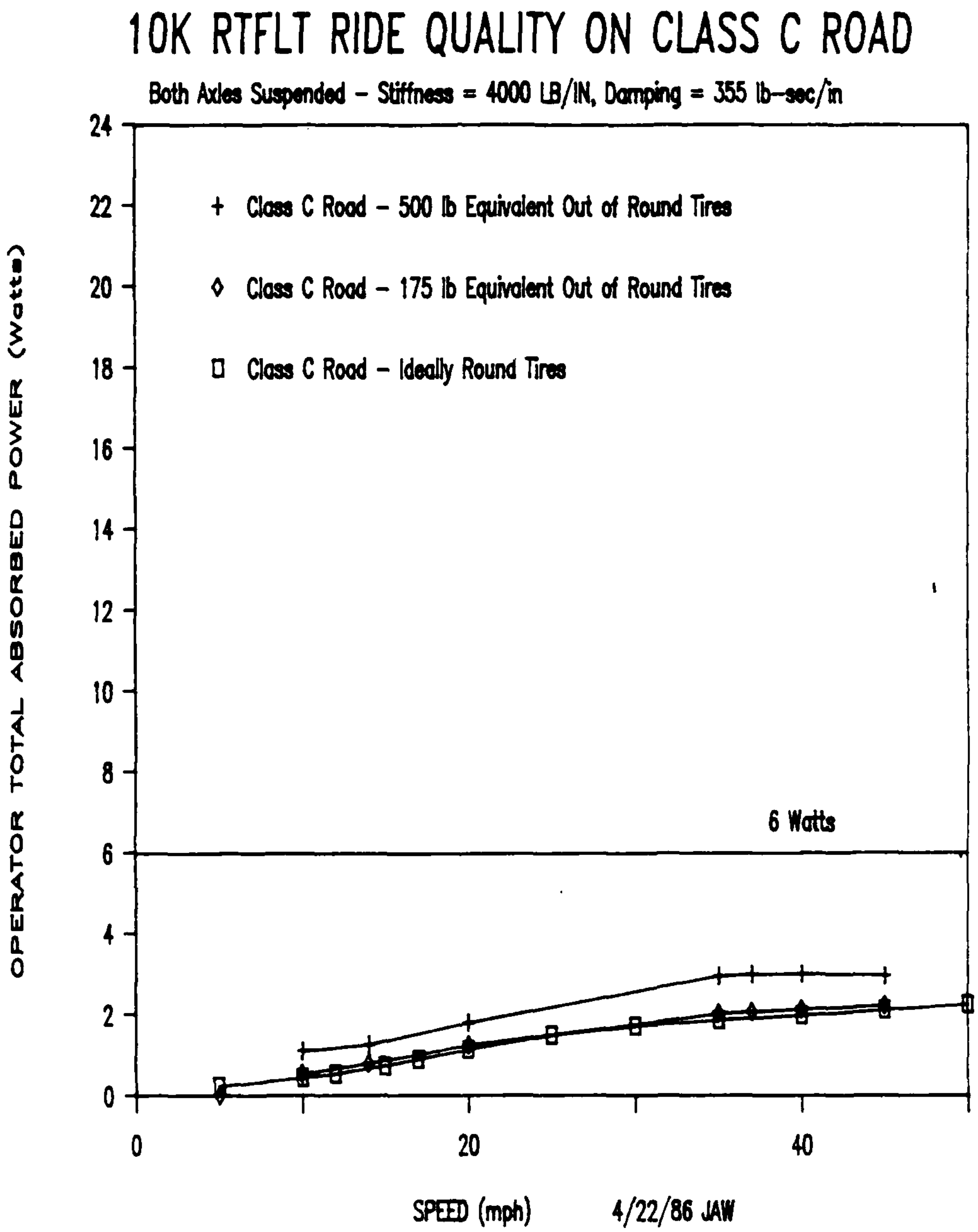

Figure 100. Effect of Tire Roundness on Ride Quality on Class C Road (10K RTFLT Both Axles Suspended)

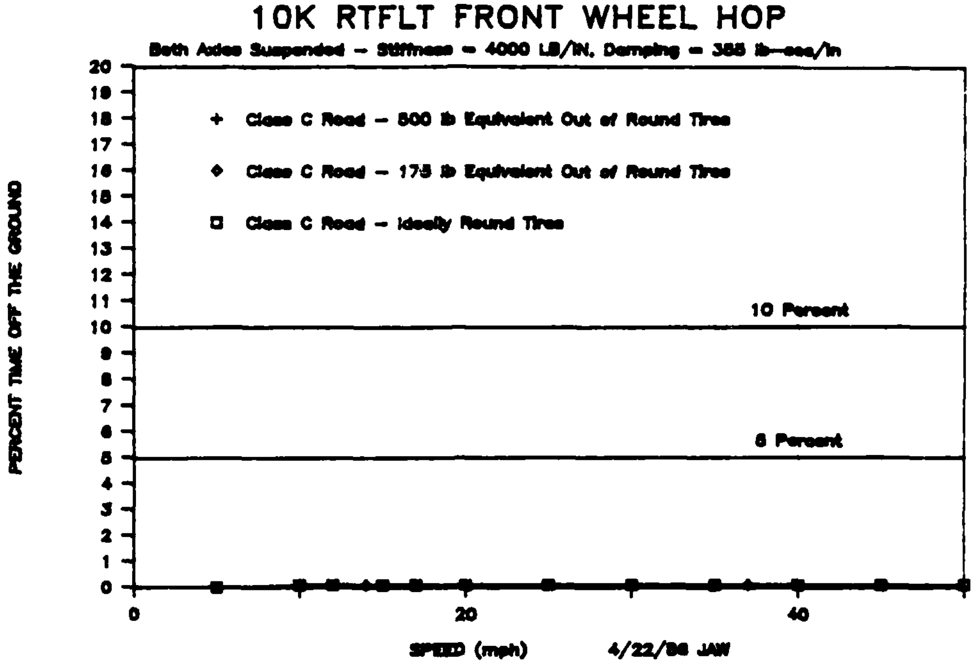

Figure 101. Effect of Tire Roundness on Front Wheel Hop on Class C Road (10K RTFLT Both Axles Suspended)

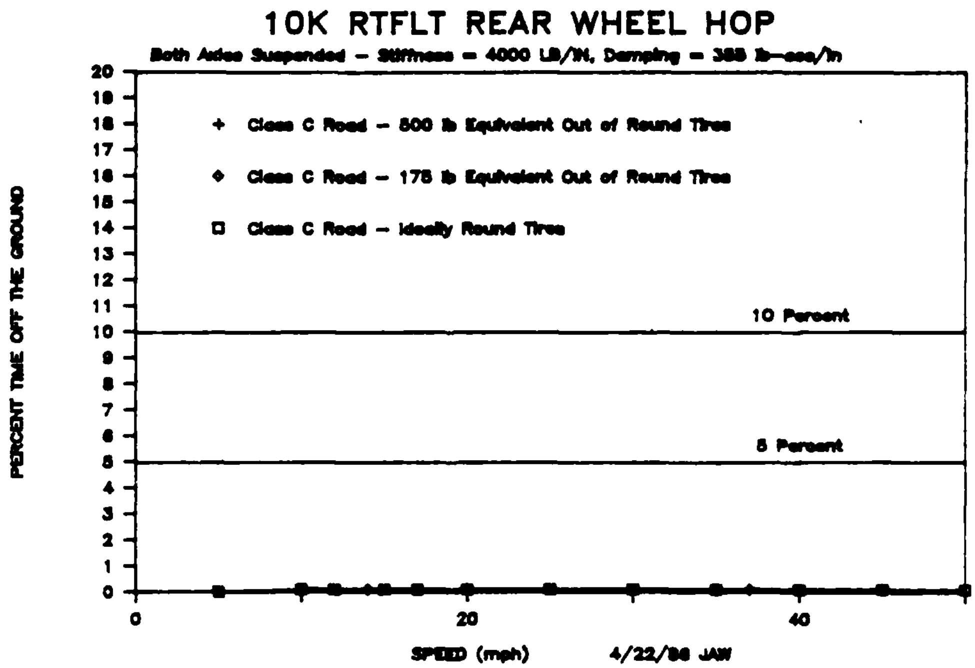

Figure 102. Effect of Tire Roundness on Rear Wheel Hop on Class C Road (10K RTFLT Both Axles Suspended)

The effects of tire out-of-roundness on the 10K RTFLT suspension options are summarized in Table 25. The unsuspended baseline does not meet the operator total absorbed power requirement for even ideally round tires on the class C road surface. In addition, the front wheel hop is unacceptable for 500 pound equivalent out-of-round tires. The front axle suspension option is marginal in ride quality with 500 pound equivalent out-of-round tires on the class C road. However, the front and rear axle suspension option is acceptable for 500 pound equivalent out-of-round tires.

Table 25. 10K RTFLT Summary of Suspension Alternatives with Out-of-Round Tires

Class C Road

Suspension Option	Peak Absorbed Power (Watts)			Front Wheel Hop Percent			Rear Wheel Hop Percent		
	Ideal Tires	175lb Out	500lb Out	Ideal Tires	175lb Out	500lb Out	Ideal Tires	175lb Out	500lb Out
Unsusp. Baseline @ 20 mph	9	12	26	0.1	1	12	0.1	0.1	0.1
Front Axle Suspended @ 15 mph	2	2	6	0.1	0.1	0.1	0.1	0.1	1
Both Axles Suspended @ 45 mph	2	2	3	0.1	0.1	0.1	0.1	0.1	0.1

KEY:

Tire Roundness (Applies to All Four Tires Equally)

- Ideal - Ideally Round Tires
- 175 lb - 175 lb Equivalent Out-of-Round Tires
- 500 lb - 500 lb Equivalent Out-of-Round Tires

Total Absorbed Power Max Limit - 6 Watts

Wheel Hop / Controllability

- Acceptable - Less than 5% Time Off the Ground
- Marginal - 5-10% Time Off the Ground
- Unacceptable - Greater than 10% Time Off the Ground

= Acceptable = Marginal = Unacceptable

2.2.3.8 Peak operator acceleration and bounce/pitch stability. The operator acceleration requirement states that the peak acceleration at the operator's station should not exceed 2.5 g's when hitting an 8 inch radius speed bump at 7 miles per hour. Four suspension alternatives for each vehicle size were evaluated with the 3-D vehicle dynamic analysis model. The suspension characteristics were determined from the 2-D analysis of operator absorbed power and wheel hop. The suspension alternatives used in the 3-D vehicle dynamics model are identified by machine rated capacity followed by the letters A, B, C, or D and have the characteristics listed in Table 26.

Table 26. Description of Suspension Alternatives

Suspension Option	Front Axle Motion	Rear Axle Motion	Spring Rate* Lb./in.	Damping Rate* Lb./(in/sec)
6k A	None	Oscillating	None	None
6k B	Vertical only	Oscillating	5000	200
6k C	Vertical and Roll	Oscillating	5000	200
6k D	Vertical and Roll	Vertical and Roll	5000	200
10k A	None	Oscillating	None	None
10k B	Vertical only	Oscillating	2000	250
10k C	Vertical and Roll	Oscillating	2000	250
10k D	Vertical and Roll	Vertical and Roll	4000	350

* Spring and damping rates are the total for each axle.

In addition to peak operator station acceleration, the simulation gives a good indication of how stable the vehicle will be while traveling at higher speeds. Current vehicles tend to get into a severe pitch or bounce mode due to several excitations. These excitations include road disturbances, tire unbalance, tire out-of-roundness, and drive line dynamics. The time required for the motion to decay after hitting the bump is a direct indication of how sensitive the vehicle will be to the excitations mentioned above. Table 27 summarizes the results of the speed bump impact simulation. Fig. 103 and Fig. 104 are traces of the motion at the operators station, right front wheel spindle, and right rear wheel spindle as the vehicle traverses the bump for the 6K and 10K RTFLTs respectively. The response of the vehicles from the simulations has been animated and is available on video tape.

Table 27. Response to 8 Inch Speed Bump at 7 Miles Per Hour

Suspension Option	Maximum Accel. (g's)	Seat Displ. (in.)	Pitch Angle (deg.)	Time to Decay to 1 Degree (seconds)
6K A	2.3	22	27	8
6K B	1.8	15	12	5
6K C	1.8	15	12	5
6K D	0.8	11	12	5
10K A	2.0	17	22	12
10K B	1.2	11	12	4
10K C	1.2	11	12	4
10K D	1.0	10	14	5

These results indicate that both vehicles are within the 2.5 g acceleration limit even without suspension. The important factor to note is that all of the suspended options reduced operator acceleration, operator vertical motion, and vehicle pitch by about 50%. In addition, a vehicle with suspension can travel faster over bumps and not exceed the acceleration limit than a vehicle without suspension. The reduction in time for the pitch to decay is a good indication of the ability of the vehicle to handle internal and external excitations without becoming unstable. Again the suspended options have a clear advantage over the unsuspended vehicle. Adding suspension to the rear axle does not provide much gain over the suspended front axle alone. Omitting rear axle suspension would allow the use of an oscillating rear axle. An oscillating axle is desirable in order to keep all the wheels on the ground on highly irregular terrain.

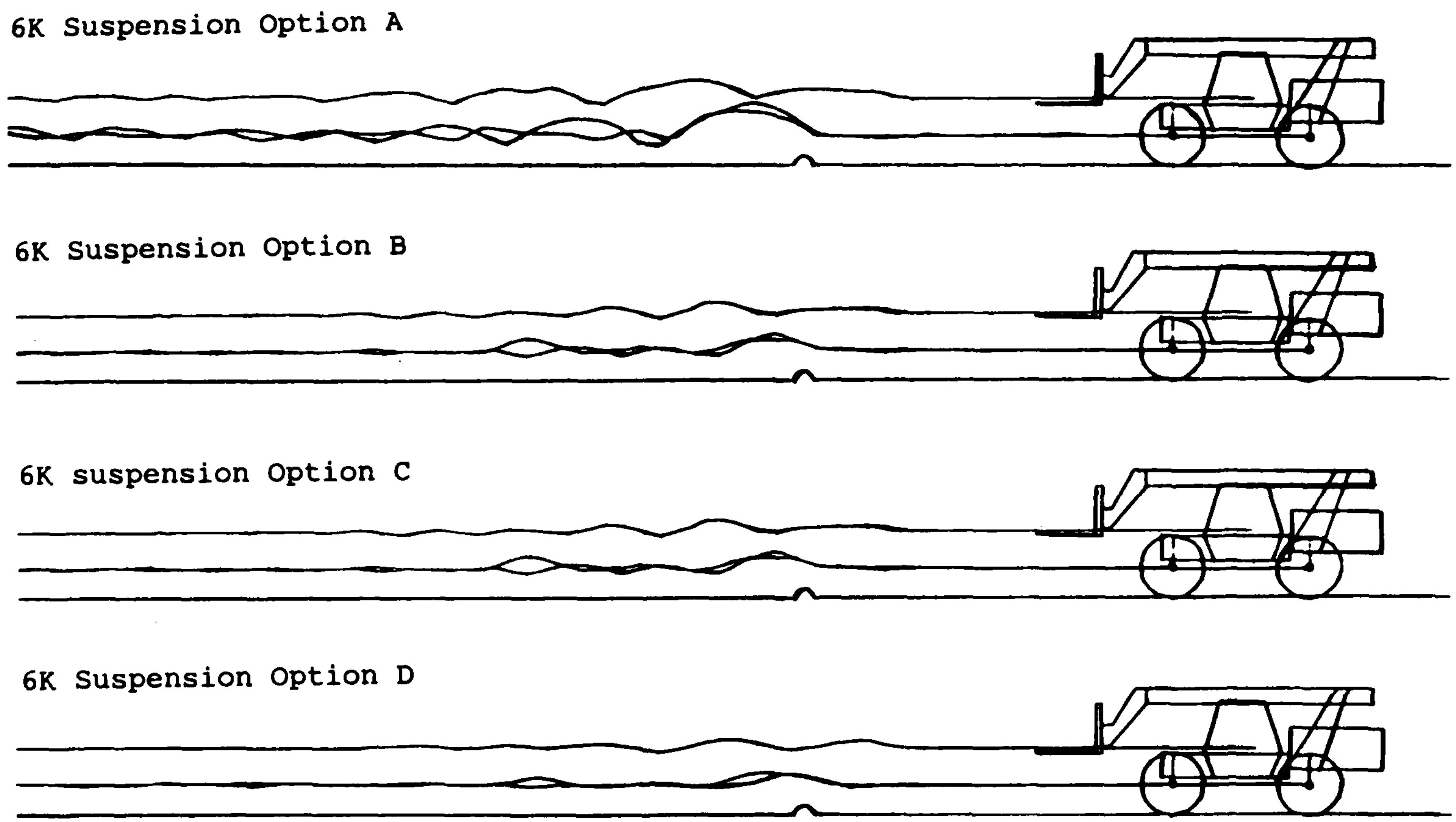

Figure 103. Four Suspension Options of the 6K RTFLT Traversing an 8 Inch Bump at 7 mph

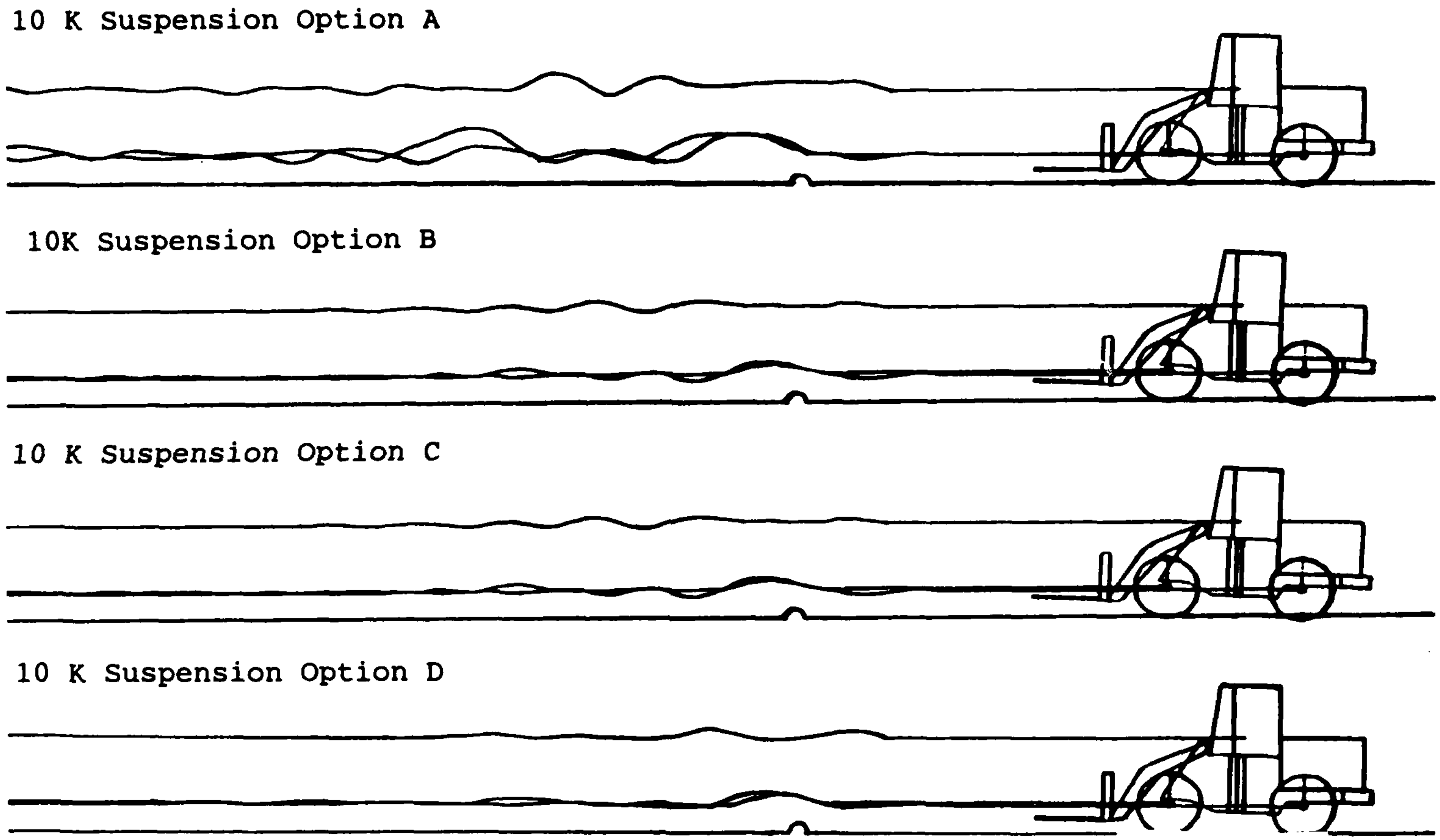

Figure 104. Four Suspension Options of the 10K RTFLT Traversing an 8 Inch Bump at 7 mph

2.2.3.9 Lateral (roll) stability. The stability requirements state that the requirements in the existing military specification should be met after the vehicle has been modified for 45 mile per hour capability. The 3-D vehicle dynamics models were used to evaluate the rolling motion as the vehicle traversed the pothole section of the durability test course (see MIL-T-52843C Fig. A-3, Durability course). The model was also used to simulate a full circle turn at maximum steer angle on a side slope while carrying rated load. The pothole course analysis was made with the vehicle travelling at 2.5 miles per hour without load. Operator vertical and lateral motion and total acceleration were used to compare the suspension options. The descriptions for the options are given in Table 26. Table 28 is a summary of the results of this analysis. Fig. 105 and Fig. 106 are traces of the motion at the operators station, right front wheel spindle, and right rear wheel spindle. In these traces the potholes alternate left and right with the first pothole on the left side of the vehicle. The response of the vehicles from the simulations has been animated and is available on video tape.

Table 28. Response to 12 Inch Potholes at 2.5 Miles Per Hour

Suspension Option	Maximum Accel. (g's)	Seat Displacement Vertical (in.)	Seat Displacement Lateral (in.)	Roll Angle (deg.)
6K A	1.5	15	15	25
6K B	1.3	15	12	23
6K C	0.8	12	10	21
6K D	0.6	11	7	14
10K A	1.6	7	10	12
10K B	1.4	7	10	12
10K C	0.9	7	13	14
10K D	0.7	6	7	8

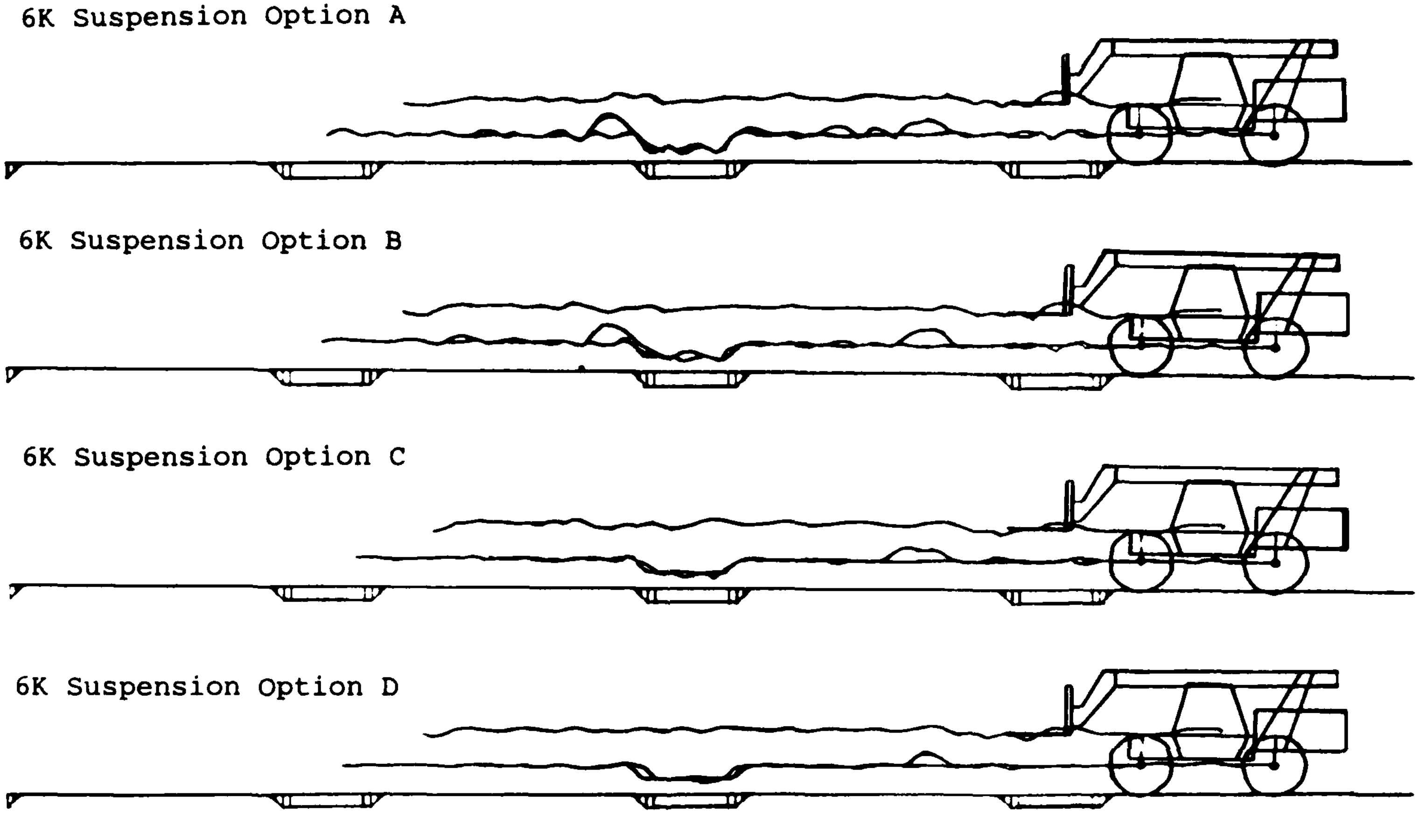

Figure 105. Four Suspension Options of the 6K RTFLT Traversing a 12 Inch Potholes at 2.5 mph

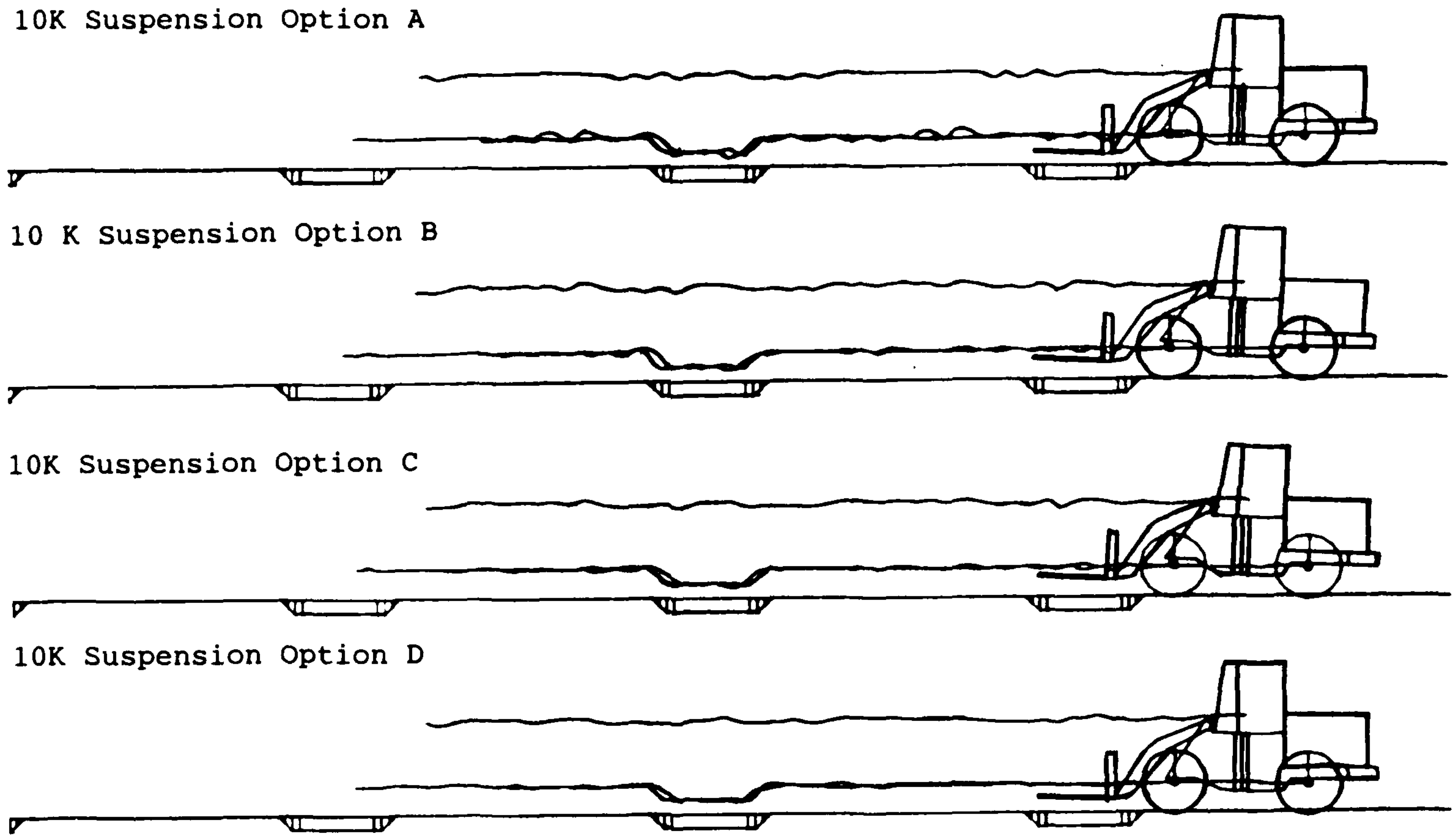

Figure 106. Four Suspension Options of the 10K RTFLT Traversing a 12 Inch Potholes at 2.5 mph

These results indicate that the only suspension options with a significant impact on lateral stability are the ones with both axles suspended (6K D and 10K D). The axles on these two options tended to move relative to the vehicle, allowing the wheels to follow the ground without forcing motion of the main chassis of the vehicle. With all the other options the vehicle experiences nearly the same amount of motion. All of the suspended options do have an effect on the acceleration at the operators station. The suspended vehicles did not experience as high a peak in acceleration when the tires drop into the pothole as when they hit the rising ramp as they exit the pothole. The suspended options which allow the axles to roll relative to the vehicle (6K C, 6K D, 10K C, 10K D) were able to follow the ground with reduced tire liftoff. However, the gains from adding roll capability to the suspension are not absolutely necessary and may even be a disadvantage from the standpoint of sideslope stability and high speed stability which are addressed in sections 2.2.3.10 and 2.3.3.2.

2.2.3.10 Sideslope and load handling stability. Sideslope and load handling stability was evaluated by having the vehicle carry its rated load and make a full circle turn at maximum steer angle on a sideslope. The side slope requirement was 30% for the 6K vehicle and 15% for the 10K vehicle. When the rated load was applied to any of the suspended options, the front axle suspension springs bottomed out. The bottom out point was set to occur when any spring was compressed 6 inches beyond the static length in the vehicle without load. This amount of deflection at the axle would produce an unacceptable amount of motion at the forks and would therefore require that the suspension be locked out during material handling operations. The springs used in the model were constant stiffness springs (a characteristic of coil springs) which may not have the same characteristics as the springs that would be used in the actual vehicles. Gas/oil struts, rubber Marsh Mellow springs, and air springs all have the characteristic of increasing stiffness as the spring is compressed and are able to resist bottoming out better than constant stiffness springs under high load. In the case of the vehicles in this study, the front axle load is more than tripled when rated load is lifted and it is felt that any suspension element will have to have some form of lock out for acceptable material handling stability. This requirement makes the question of whether or not to allow the suspended axles to roll relative to the vehicle irrelevant for material handling. The impact of axle roll on controllability will be addressed further in section 2.3.3.2.4. All of the options were stable during the full circle turn on the sideslope and since the front axle suspensions had bottomed out, the amount of roll varied less than 2 degrees from the static position on flat ground.

2.2.4 Suspension Requirements

The results of the analysis study presented in 2.2.3 are summarized in Table 29 for the 6K RTFLT. Controllability (wheel hop) on the class D road limits the maximum suspension stiffness on the front axle suspended version to 5,000 - 8,000 lb/in. The lower limit of 3,000 - 4,000 lb/in/axle on suspension stiffness is set to keep the lowest natural frequency of vibration of the vehicle greater than 1 Hertz to avoid "sea sickness" in the operator. Controllability on the class D road is also the limiting factor on suspension when both axles are suspended. In this case the upper limit is 5,000 - 10,000 lb/in/axle and the lower limit is 4,000 - 5,000 lb/in/axle. Note that the ride quality on the class C road with ideally round tires or 175 pound equivalent out-of-round tires can be met with a very stiff or rigid (no suspension) suspension. The desired (but not required) limit of 6 Watts total absorbed power on the class D road cannot be met with a front suspension only, even with ideally round tires.

Table 29. 6K RTFLT Workable Suspension Stiffness Range

<table>
<tr><td></td><td colspan="3">Max Suspension Stiffness per Axle to Meet Requirement</td></tr>
<tr><td>Suspension Option</td><td>Ride Quality Class C Road @ 45 mph</td><td>Controllable Class D Road @ 45 mph</td><td>Ride Quality* Class D Road @ 45 mph</td></tr>
<tr><td>Front Axle Suspended
- Ideal Tires
- 175 lb Out
- 500 lb Out</td><td>
Rigid
Rigid
< 5,000</td><td>
8,000
5,000
< 5,000</td><td>
cannot be met
cannot be met
cannot be met</td></tr>
<tr><td>Max Allowable</td><td colspan="3">5,000 - 8,000 lb/in/axle</td></tr>
<tr><td>Min Allowable**</td><td colspan="3">3,000 - 4,000 lb/in/axle</td></tr>
<tr><td>Both Axles Suspended
- Ideal Tires
- 175 lb Out
- 500 lb Out</td><td>
Rigid
Rigid
10,000</td><td>
10,000
5,000
5,000</td><td>
10,000
10,000
5,000</td></tr>
<tr><td>Max Allowable</td><td colspan="3">5,000 - 10,000 lb/in/axle</td></tr>
<tr><td>Min Allowable**</td><td colspan="3">4,000 - 5,000 lb/in/axle</td></tr>
</table>

* Not required but desired
** Minimum suspension stiffness for greater than 1 Hertz natural frequency of lowest mode of vibration.
Damping - approximately 200 lb-sec/in/axle at 5000 lb/in/axle stiffness

The workable suspension stiffness range for the 10K RTFLT is listed in Table 30. Ride quality (6 Watts total operator absorbed power at 45 mph) is the limiting factor on this vehicle with front axle suspended only. Therefore, the upper limit on suspension stiffness is 2,000 - 8,000 lb/in/axle and the lower limit is 2,000 - 3,000 lb/in/axle. With both axles suspended, the ride quality on the class C road at 45 mph and the controllability on the class D road at 45 mph reach the limit at about the same stiffness. The maximum for the two axle suspension is 8,000 - 10,000 lb/in/axle and the lower limit is 4,000 - 5,000 lb/in/axle. The desired (but not required) ride quality 6 watt total operator absorbed power limit on the class D road at 45 mph cannot be met with either the front only suspended or with both axles suspended. The class D road ride quality is out of reach with both axles suspended since it requires a suspension stiffness that results in less than 1 Hertz natural frequency for the lowest mode of vibration of the vehicle.

Table 30. 10K RTFLT Workable Suspension Stiffness Range

	Max Suspension Stiffness per Axle to Meet Requirement		
Suspension Option	Ride Quality Class C Road @ 45 mph	Controllable Class D Road @ 45 mph	Ride Quality* Class D Road @ 45 mph
Front Axle Suspended			
- Ideal Tires	8,000	10,000	cannot be met
- 175 lb Out	7,500	10,000	cannot be met
- 500 lb Out	2,000	10,000	cannot be met
Max Allowable	2,000 - 8,000 lb/in/axle		
Min Allowable**	2,000 - 3,000 lb/in/axle		
Both Axles Suspended			
- Ideal Tires	10,000	10,000	3,000 ***
- 175 lb Out	10,000	8,000	2,000
- 500 lb Out	8,000	8,000	2,000
Max Allowable	8,000 - 10,000 lb/in/axle		
Min Allowable**	4,000 - 5,000 lb/in/axle		

* Not required but desired
** Minimum suspension stiffness for greater than 1 Hertz natural frequency of lowest mode of vibration.
*** Unacceptable - Results in natural frequency less than 1 Hertz for the lowest mode of vibration.
Damping - approximately 250-350 lb-sec/in/axle at 2000-4000 lb/in/axle stiffness

2.2.5 Suspension Components

Several types of suspension components are in common use in vehicles today. It is therefore appropriate to discuss the characteristics and commercial availability of some of the main candidates.

2.2.5.1 Types of suspension elements. Several types of suspension elements are available for use in vehicle suspensions. These include steel coil springs, elastomeric springs, gas / oil struts, and air springs. Following is a discussion of the characteristics of each type.

2.2.5.1.1 Steel coil spring. Helical coil springs are typically used in automotive applications. External dampers or shock absorbers are required with coil springs due to the very low internal damping. Coil springs typically provide linear spring rates, require relatively large volumes, and are heavy. A steel coil spring for a vehicle of the RTFLT weight would be excessively large and heavy. A separate suspension lockout system would be required to lock the suspension when not needed.

2.2.5.1.2 Elastomeric spring. An elastomeric spring consists of a cylindrically shaped rubber element wrapped with reinforcing plies to hold the uniform cylindrical shape when compressed. An example is the Firestone Marsh Mellow spring. Marsh Mellow springs are relatively small, light weight, inexpensive, and have enough internal damping in the rubber to eliminate the need for an external damping mechanism. Elastomeric springs provide an increasing spring rate when compressed and are limited in compression to 50% of the free length of the spring. A separate suspension lockout system would be required to lock the suspension when not needed.

2.2.5.1.3 Gas / oil system. This suspension system consists of hydraulic cylinders connected to a gas / oil (usually nitrogen) charged accumulator. This type of suspension can be varied to produce the appropriate spring rate and damping rate for a particular vehicle by varying the amount of gas precharge or orifice areas. In addition, these systems typically feature "load leveling" and "lock out" capability. The systems are in use on construction equipment. For example, the Caterpillar 615 tractor scraper (27 mph max speed capability) and the Caterpillar D30C articulated truck (35 mph max speed capability) use gas / oil suspension systems.

2.2.5.1.4 Air springs. This device is a sealed rubber bladder that is filled with low pressure gas (typically air up to 120 psi max). Air springs have very little internal damping and would require an external damper. The Firestone Airstroke actuator is an example of an air spring. Air springs are light and inexpensive. The relatively low load capacity of these air springs may mean that more suspension units would be required than for other types of suspension components.

2.2.5.2 Feasibility / Availability. Several types of suspension components exist which have the required characteristics for the high speed roading capability (e.g. air springs, elastomeric springs, and gas / oil systems). The gas / oil system looks the most promising due to its built in lock out capability and packaging flexibility. The elastomeric spring (e.g. the Firestone Marsh Mellow spring) may not have the loaded / unloaded vehicle load range capability required unless it is always locked out under loaded vehicle conditions. Both air springs (e.g. Firestone Aistroke Acuators) and elastomeric springs may present packaging problems due to the relatively large diameters required for the axles loads that will be encountered and will require an external lock out mechanism.

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2.3 Steering System

Part of this feasibility study was to evaluate the high speed controllability of the rough terrain fork lift trucks and to compare articulated steering to Ackermann steering. The ability of the rough terrain fork lift trucks to travel at 45 miles per hour and at the same time be controllable is dependent on two factors. The inherent stability of the machine from the standpoint of oversteer/understeer characteristics and the ability of the steering system to respond to inputs from the operator. The oversteer/understeer characteristics are functions of the weight distribution of the machine and the characteristics of the tires. Common automotive stability theory and fairly simple equations can be used to evaluate this characteristic. However, the evaluation of steering system response and the interaction with the operator and machine is very complex and requires sophisticated modeling techniques. The analysis techniques and results will be presented in the following sections.

2.3.1 Computer Models

The computer model used for analysis of the high speed handling characteristics of the vehicles was the 3-D vehicle dynamics model discussed in section 2.2.1.3 with some special features added. These features included models of the steering system characteristics and an operator model. These two items allowed us to define a desired path that the vehicle is to follow in the simulation and study the difficulty that an operator would have following that path for any given set of steering system characteristics. Following is a discussion of the steering system and operator models.

A schematic of the steering system model is given in Figure 107. The steering system characteristics that can be studied include steering system reduction ratio, deadband, maximum rate, and time lag. Each of these characteristics has an impact on how the vehicle will respond to inputs from the operator and how much work will be required from the operator to maintain his desired course. The steering system model computes the resulting motion of the steering linkage as a result of any input at the steering wheel.

DEFINITIONS:

- Steering system reduction ratio is the amount of steering wheel rotation required to produce one degree of rotation at the tires.
- Steering system deadband is the amount of steering wheel motion required before any tire motion occurs.
- Steering system maximum rate is the maximum change in tire angle that can occur in one second.
- Steering system time lag is the time required for the tires to respond to steering wheel motion.

The operator model is used only for evaluation of high speed controllability. The operator model is given a desired path for the vehicle to follow and determines the steering wheel input required to follow that path. The operator model modifies the steering wheel angle based on the lateral position of the vehicle relative to the desired path and the heading angle of the vehicle relative to the desired path. The theory for the operator response equations used in our model was developed by D.T.McRuer and E.S.Krendel of Wright Patterson Air Force Base. Their operator response model was developed from experiments involving test subjects with a control device attempting to track a moving target. The operator model which was developed primarily for gun laying devices and aircraft control is well suited for simulating vehicle handling. A schematic of the elements included in the operator model is given in Figure 108.

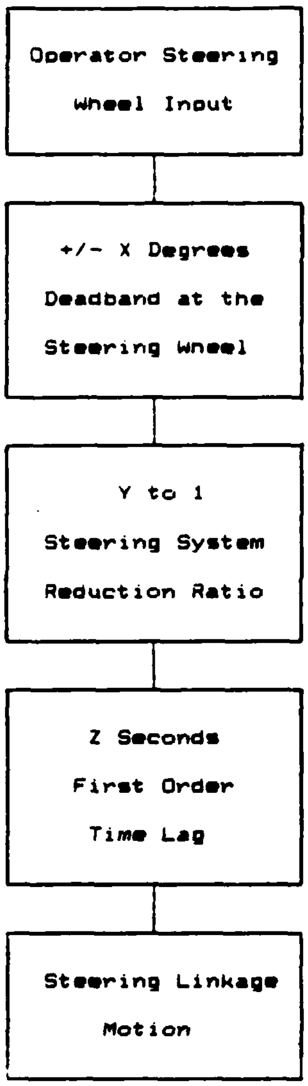


Figure 107. Steering System Model Schematic

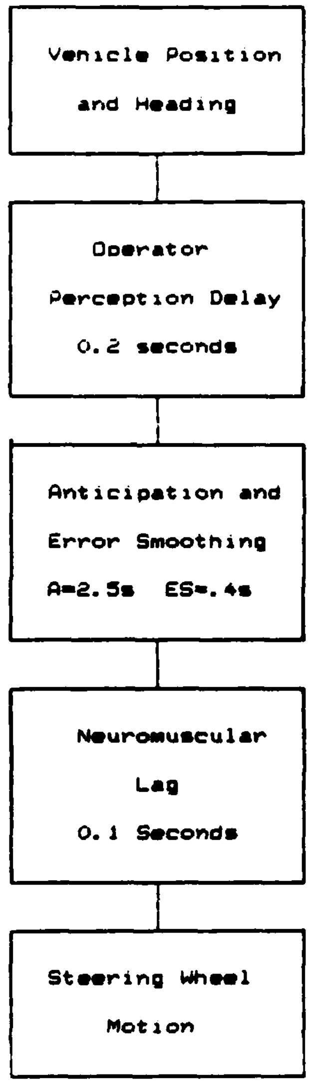


Figure 108. Operator Model Schematic

2.3.2 Operating Conditions

The operating conditions and acceptance criteria for the vehicle's steering system are fairly generic in nature. The steering system and controllability study is basically an attempt to determine if it is possible to control these vehicles at 45 miles per hour and if so, what type of steering system is required. The only requirement for the steering system we were given was that the vehicle should be controllable on a class D surface (.62 inch rms) at 45 miles per hour. There are several sections in the current military specification that may be affected by the addition of 45 mile per hour capability. These primarily refer to the type of steering system used and its performance characteristics. These sections are addressed in detail in Section 2.3.4.

2.3.3 Analysis

The high speed controllability analysis was done in three parts. The first concerns the ability of the machine to travel at 45 miles per hour on a class D surface and maintain sufficient tire/ground contact so that the tires can develop the lateral forces required to steer the vehicle. This has been addressed in Section 2.2.3.5 and the appropriate suspension parameters were found to meet the requirement. The second part concerns the evaluation of the understeer/oversteer characteristics of the vehicle and the third concerns the steering system characteristics required to give the operator sufficient control at high speed. These last two items will be discussed here.

2.3.3.1 Oversteer/understeer/critical speed. The oversteer/understeer characteristic has been used for a long time to evaluate the inherent stability of automobiles. The amount of oversteer/understeer a vehicle has is dependent on the fore-aft weight distribution, mass, wheelbase, and the cornering stiffness characteristics of the tires. For a given steering angle an understeer vehicle has the tendency to follow a larger radius path as speed is increased while an oversteer vehicle will follow a smaller radius path as speed is increased. A detailed analysis of the oversteer/understeer characteristic reveals that a vehicle with oversteer will have some critical speed above which the vehicle is inherently unstable. This does not mean that the machine is necessarily uncontrollable, but indicates that even experienced operators with perfect steering systems will find it increasingly difficult if not impossible to control the machine at speeds above the critical speed. Due to the rearward center of gravity that these machines have, they will demonstrate oversteer characteristics. Cornering stiffness data for the specific tires to be used on these machines is not available so existing data from the most similar tires available was used. The information used for the critical speed calculations and the calculated critical speeds are given in Table 31. These speeds will apply to both articulated and Ackermann steered machines.

Table 31. Critical Speed Calculation

Machine size	Weight (Lbs)	Wheelbase (inches)	Rear Axle to CG (in)	Cornering Stiffness Front (Lb/rad)	Rear
6k	30000	110	39	81200	110900
10k	37000	108	35	94700	139200

6k Critical Speed = 1140 in/sec = 56 mph
10k Critical Speed = 886 in/sec = 50 mph

These calculations indicate that the machines should be stable up to 45 miles per hour if the weight distribution and tire characteristics do not change significantly from the values used above. When a machine is designed, the critical speed should be re-calculated to make sure that the critical speed is not reduced any further, and if possible should be increased to provide a greater margin of stability. This can be done by simply transferring some of the machines weight towards the front axle when in the roading configuration.

2.3.3.2 High speed controllability analysis. The steering system characteristics were evaluated by telling the operator model to attempt a lane change maneuver as quickly as possible while traveling at 50 miles per hour on smooth pavement. The lane change maneuver involved steering the vehicle such that it would move to the left a distance of 12 feet and resume a course parallel to the original course. The speed of 50 miles per hour was used instead of the required 45 miles per hour to account for travelling down a grade. The effects that deadband, gain, response time, and maximum steering system rate have on stability were examined and compared to the response of an ideal steering system. The operator model is of course not as intelligent as a real operator and some of the situations that demonstrate mild instability will be acceptable in actuality. A real operator has the ability to analyze the response problems a steering system has and as his experience on the machine increases he learns to compensate for those problems. The steering system options used in the analysis are identified by the machines rated capacity followed by the number 1, 2, 3, or 4 and the corresponding characteristics are listed in Table 32.

Table 32. Rapid Lane Change Steering System Characteristics

Machine Size	Steering Sys. Type	Deadband (Degrees)	Reduction Ratio	Time Lag (seconds)	Max Rate (Deg/sec)
6k #1	Ackermann	0	20:1	0.0	100
6k #2	Ackermann	10	20:1	0.5	30
6k #3	Ackermann	20	20:1	0.5	30
6k #4	Ackermann	10	20:1	1.0	30
10k #1	articulated	0	20:1	0.0	100
10k #2	articulated	10	20:1	0.5	30
10k #3	articulated	20	20:1	0.5	30
10k #4	articulated	10	20:1	1.0	30

Table 33 summarizes the results of the rapid lane change simulations.

Table 33. Rapid Lane Change Results

Machine Size	Time to move 144 inches (seconds)	Time to stabilize within 12 inches (seconds)
6k #1	3.0	5.5
6k #2	3.0	6.0
6k #3	3.0	10.0
6k #4	3.0	UNSTABLE
10k #1	3.0	3.0
10k #2	3.0	5.0
10k #3	3.0	7.0
10k #4	3.0	UNSTABLE

Figures 109 thru 112 are plots of the lateral position of the vehicle versus time for the 6k RTFLT lane change simulations. The traces in Figure 113 are overhead snapshots of the four 6k steering system options at .5 second intervals. The first shot of the vehicle is at time equal to one second in the simulation. This corresponds to the point in time that the operator model gets the signal to begin the lane change maneuver. The last shot of the vehicle is approximately at time equal to eight seconds. Many of these superimposed sequences appear similar during the first eight seconds but differ greatly afterwards as can be seen in the lateral position plots. The clarity of the superimposed snapshots prevented us from showing the entire simulation on the page. The results of this analysis have been animated and are available on video tape.

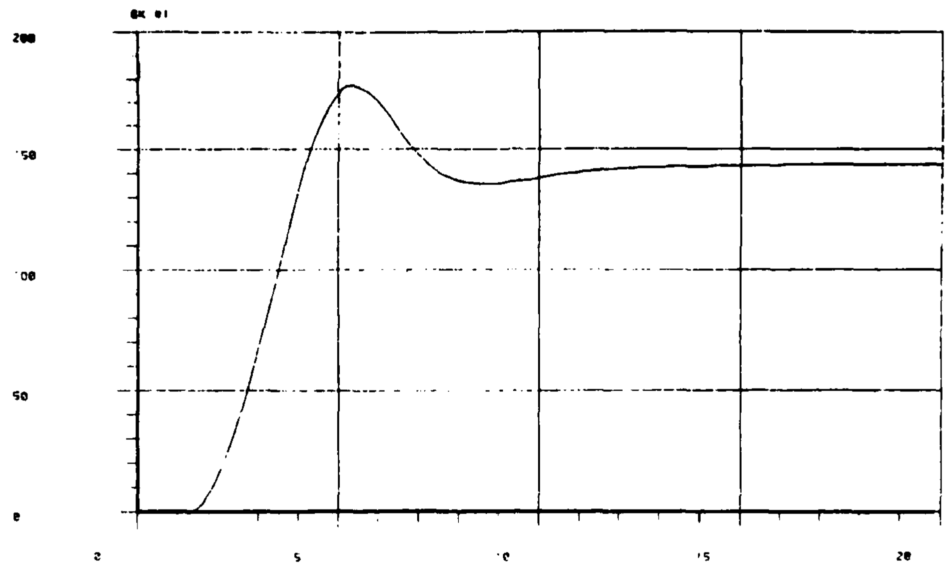


Figure 109. Lane Change Response for Steering System 6k #1

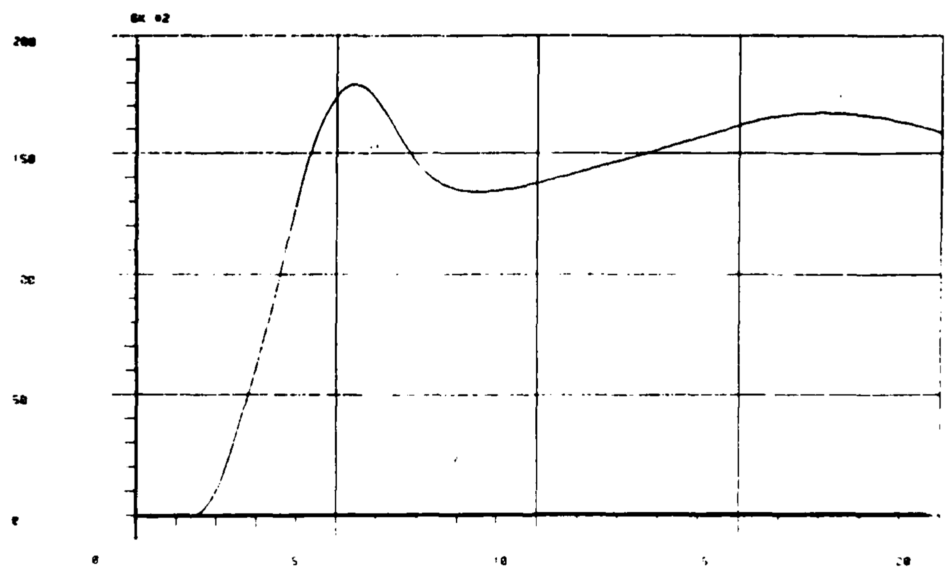


Figure 110. Lane Change Response for Steering System 6k #2

LATERAL POSITION (INCHES) VS TIME (SECONDS)

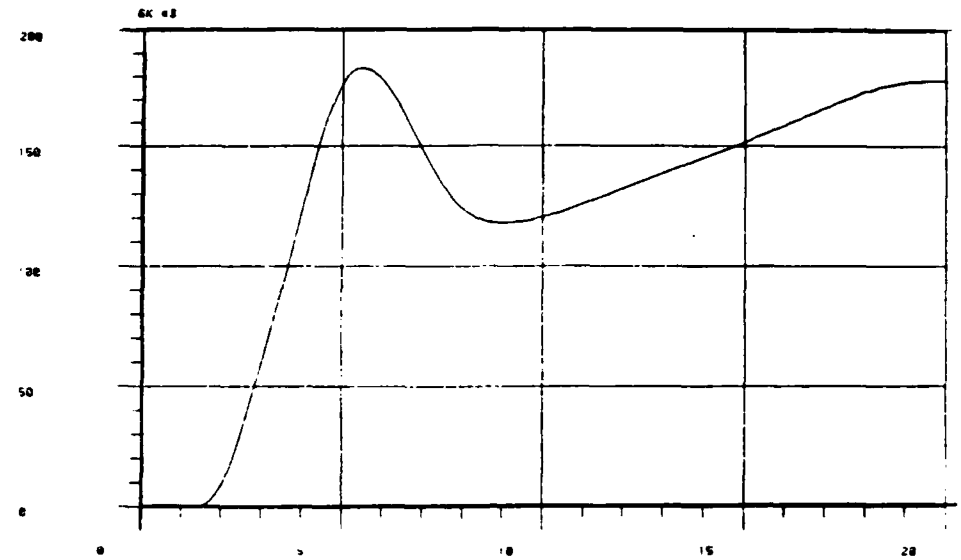


Figure 111. Lane Change Response for Steering System 6k #3

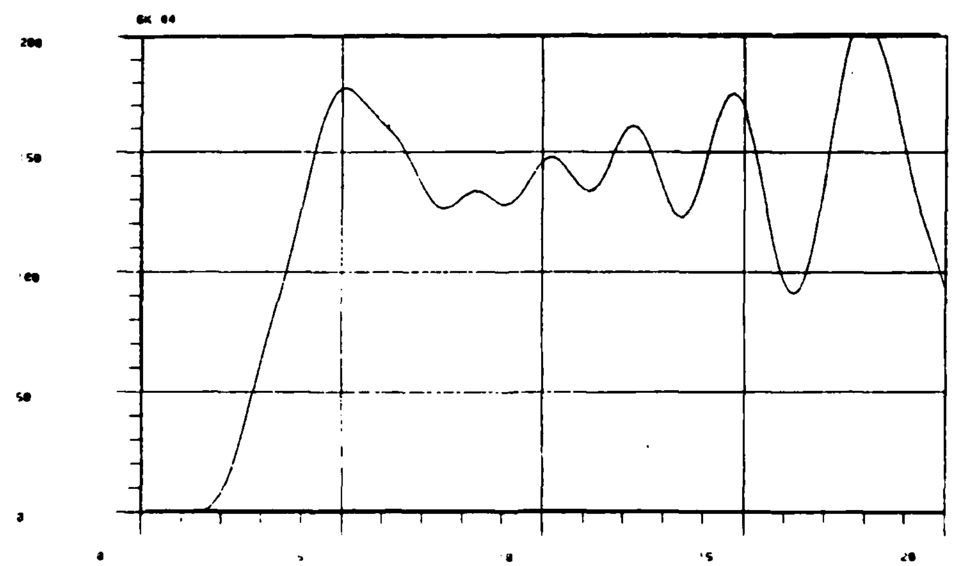
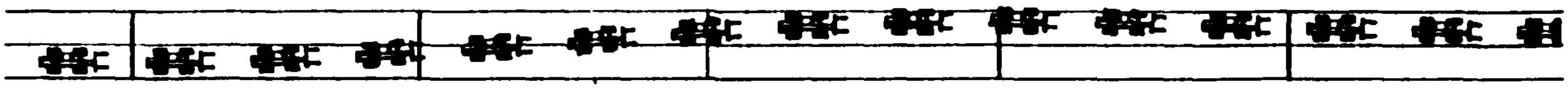
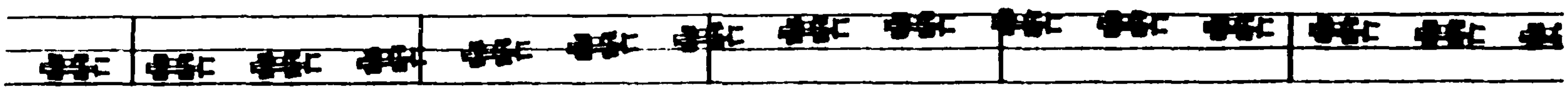


Figure 112. Lane Change Response for Steering System 6k #4

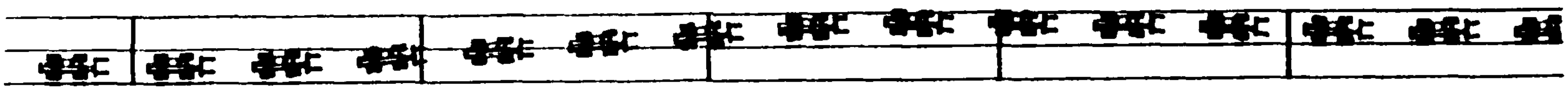
Steering System 6K #1



Steering System 6K #2



Steering System 6K #3



Steering System 6K #4

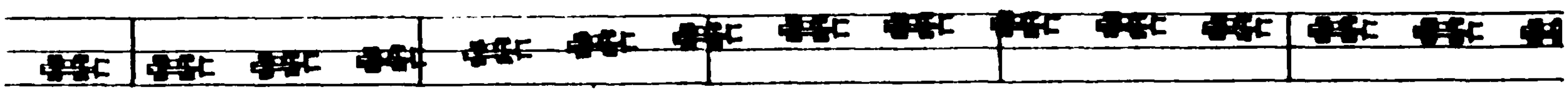


Figure 113. Lane Change Response Animation for the Four 6K Steering Systems

Figures 114 thru 117 are plots of the lateral position of the vehicle versus time for the 10k RTFLT lane change simulations. The traces in Figure 118 are overhead snapshots of the four 10k steering system options at .5 second intervals beginning with the start of the lane change and ending at time approximately equal to 8 seconds. The results of this analysis have been animated and are available on video tape.

LATERAL POSITION (INCHES) VS TIME (SECONDS)

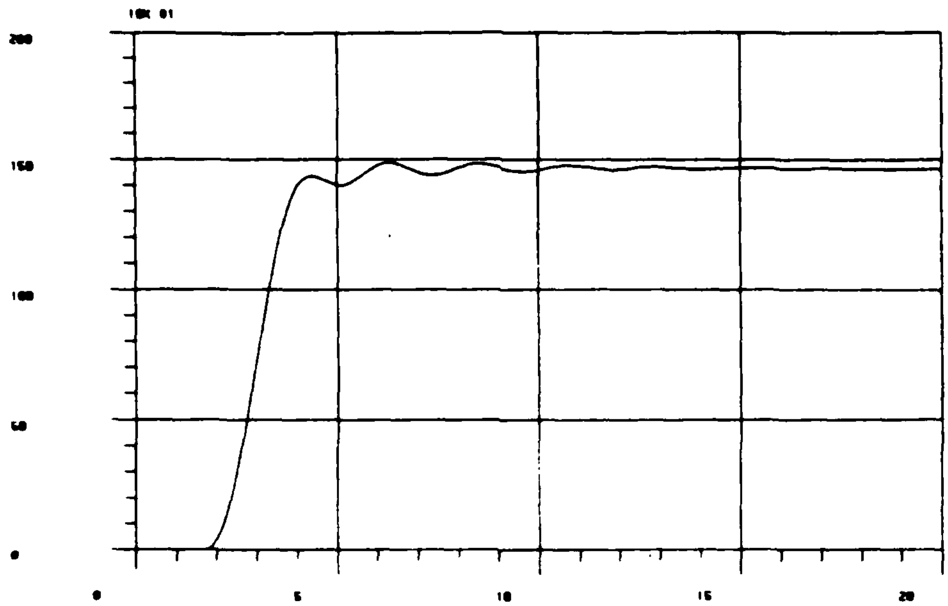


Figure 114. Lane Change Response for Steering System 10k #1

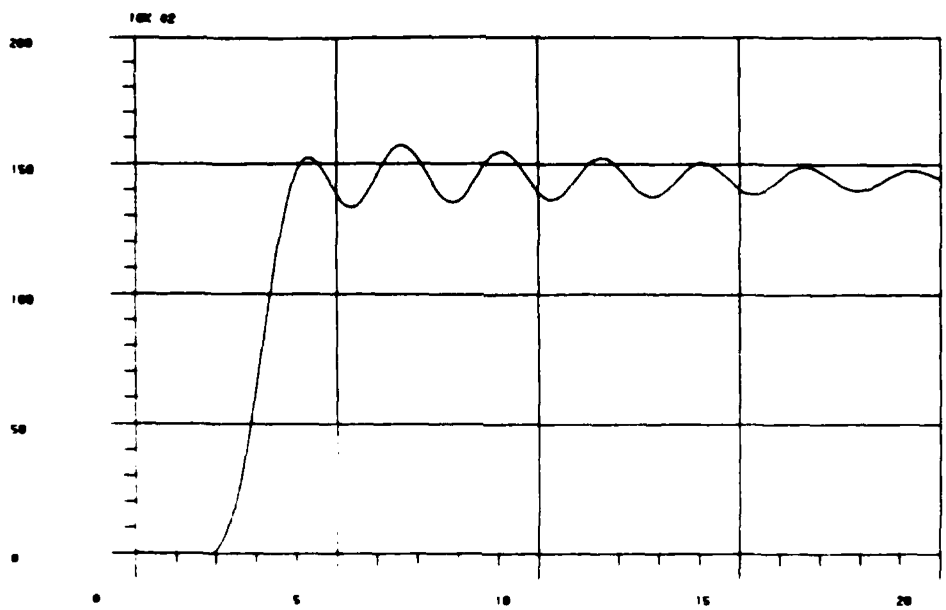


Figure 115. Lane Change Response for Steering System 10k #2

LATERAL POSITION (INCHES) VS TIME (SECONDS)

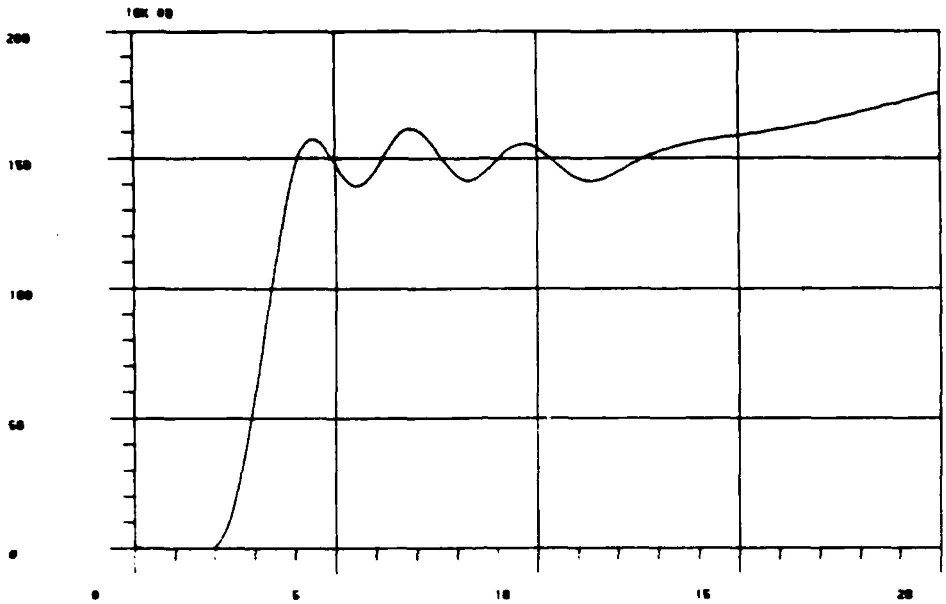


Figure 116. Lane Change Response for Steering System 10k #3

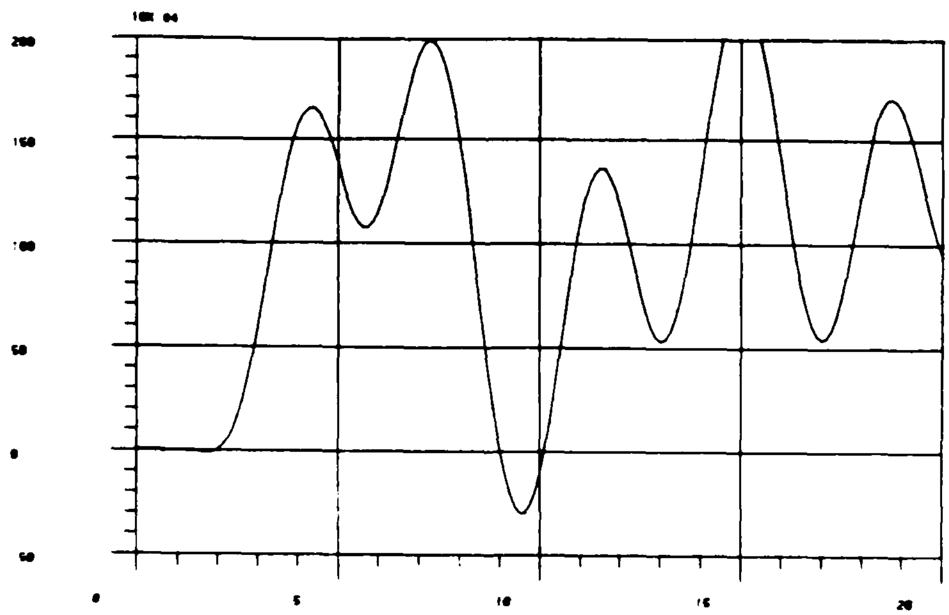


Figure 117. Lane Change Response for Steering System 10k #4

Steering System 10K #1

Steering System 10K #2

Steering System 10K #3

Steering System 10K #4

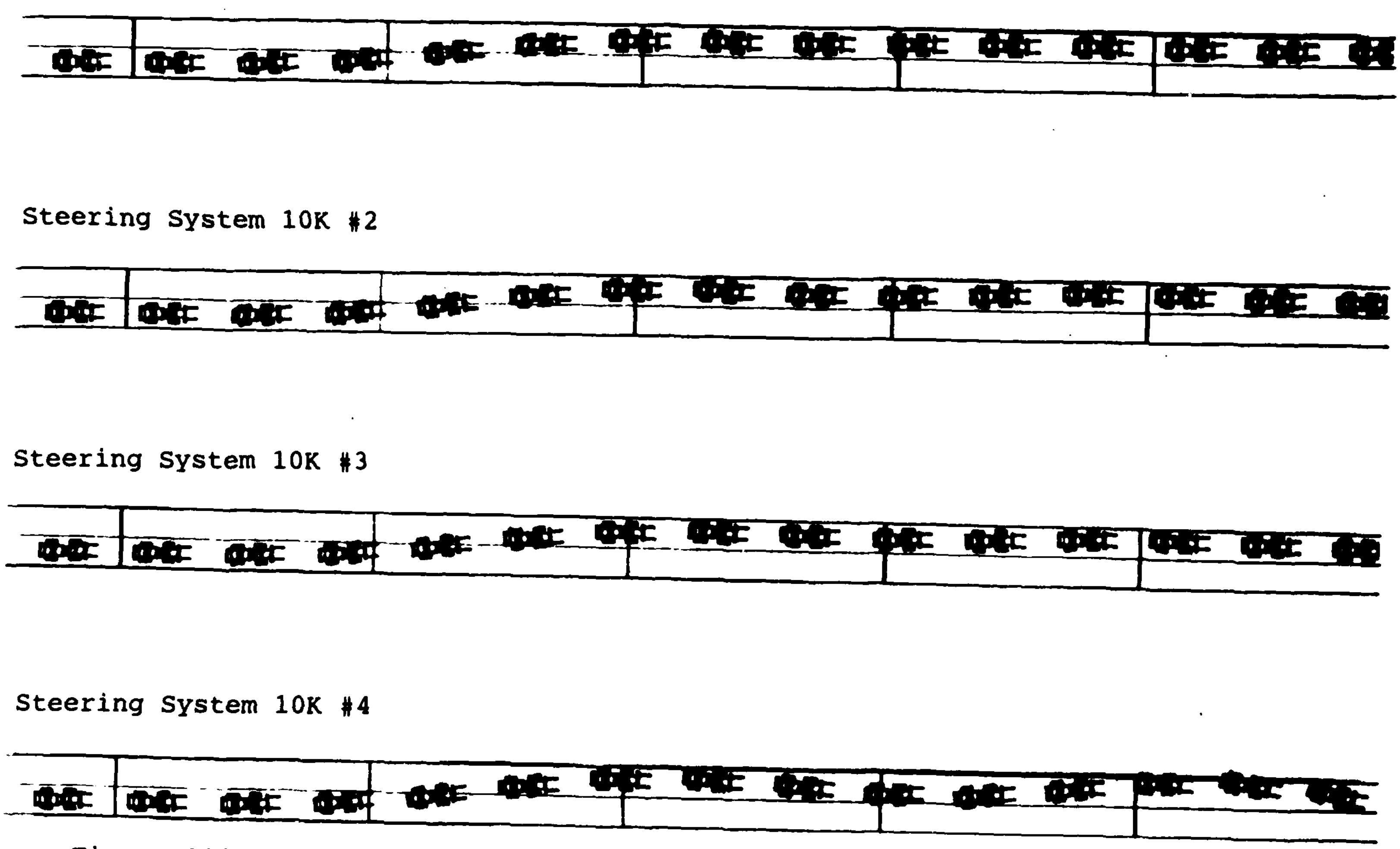


Figure 118. Lane Change Response Animation for the Four 10K Steering Systems

The machines were able to make the 12 foot move to the left in 3 seconds under the control of the operator model. The time required for a machine travelling at 45 miles per hour to reach a skidding object initially at the same speed and one second ahead is approximately 2.5 seconds. In all cases, the vehicles would have been clear of another vehicle of equal width in 2.5 seconds. The lateral tire forces did not reach their maximum levels during this maneuver at which point skidding would occur. A human operator could probably reduce the time by .5 to 1 seconds by utilizing the full capabilities of the tires. Unfortunately, the operator model used in these simulations does not have the intelligence to push the vehicle to its limit. The fact that the operator model can control the vehicle indicates that it will be stable with a human operator behind the wheel.

2.3.3.2.1 Reduction ratio and maximum rate. Steering system option #1 for both the 6K and 10K machines is the ideal system and is used as the baseline to compare the other systems. The results documented here deal only with the effects that deadband and time lag have on response. The evaluation of the effects of the steering system reduction ratio indicated that the operator can adjust to a wide range so long as it is not below 15:1 which would make the system very sensitive to steering wheel motion or above 25:1 which would require more than +/- 90 degrees of steering wheel rotation in emergency situations. The evaluation of steering system maximum rate indicated that as long as the maximum rate is above 20 degrees per second at the wheels, it will be able to keep up with the operators demand. The steering systems in off road equipment such as wheel loaders, dump trucks, and tractor scrapers generally have reduction ratios and maximum rates within these limits.

2.3.3.2.2 Deadband. The evaluation of steering system deadband indicates that it should be somewhere between +/- 5 and +/- 10 degrees at the steering wheel. Some amount of deadband is desirable to prevent accidental steering when traversing rough terrain. When deadband exceeds 10 degrees, the operator is forced to make rather large corrections with the steering wheel even when attempting to maintain a straight line. The 20 degree deadband in 6k #3 and 10k #3 is the cause of the deviation from the desired lateral position from 10 seconds on. The operator is steering to correct the heading of the vehicle, but the amount he has steered is less than the system deadband. The operator would have eventually changed his heading when his deviation from the desired path reached a high enough value to call for a steering correction greater than the system deadband. Human operators can compensate for fairly large amounts of deadband if required to do so, but fatigue will be experienced after some time under these conditions due to the large amount of steering required.

2.3.3.2.3 Time lag. The evaluation of time lag indicates that it may be the most important factor in determining high speed controllability. The only way an operator can compensate for a system with a large amount of time lag is to anticipate the upcoming maneuver and start his steering wheel motion in advance. This anticipation is of course not always possible and could be potentially dangerous in emergency situations. As can be seen from the response of 6k #4 and 10k #4, a time lag of 1 second causes unstable response of the vehicle. By the time the operator's corrections affect the motion of the vehicle, it has gone well beyond the desired position, forcing an even more severe maneuver in the opposite direction. The results of the analysis indicate that it is desirable for the time lag to be under .25 seconds if possible, and should not exceed .5 seconds at the most. Typical wheel loader steering systems are generally in the .5 to .6 second range. Off highway truck steering systems generally fall in the .25 to .5 second range and would be acceptable for use at high speeds.

2.3.3.2.4 Roll characteristics. A suspended version of the 10k machine was simulated to examine the affect of suspension roll on handling characteristics. The response of the vehicle was nearly identical to the machine without suspension roll capability. The roll angle for the suspension without roll was +/- 1 degree versus +/- 4 degrees for the suspension with roll. Four degrees of roll translates into approximately 5 inches of lateral motion at the operators station. This amount of roll will not cause any stability problems, but may produce an impression of instability for the operator. Since roll characteristics have had little affect in all of the analyses, the final decision of whether or not to include it should be made from the standpoint of ease of manufacture, cost, weight, and structural integrity.

2.3.3.2.5 Ackermann versus articulated differences. No significant differences appear between the high speed maneuvering capabilities of an Ackermann steered machine and an articulated steer machine. However, there are differences in the size of the steering system required, complexity, reliability, self centering characteristics, and the reaction of the vehicle while the steering system is in motion. An articulated steering system must be able to overcome a much larger inertia than an Ackermann steering system and must withstand a much higher moment when lateral tire forces are present. Articulated machinery steering systems are usually designed to handle high loading conditions due to the severe applications they are often subjected to. Articulated systems are generally the simplest to design and less prone to failure in severe applications. Most Ackermann steering systems tend to be self centering while articulated steering systems are not. When the steer angle is changing in an

Ackermann steering system, there is no effect on machine motion other than an increase or decrease in the angular velocity of the machine in the same direction as the steering wheel motion. In an articulated machine, the initial motion of the steering system causes the front of the machine to rotate in the direction of the steering maneuver and the rear of the machine to rotate in the opposite direction. This motion is visible on the lateral position traces for the 10k machine even though the steering wheel motion is approximately the same for both types of systems. After a short transient period, the articulated machine then reacts in the same way as the Ackermann steered machine. The operator will most likely be able to adjust to the transient motion of an articulated machine in a fairly short period of time. It is easier to add drive capability to the wheels in an articulated steering system than in an Ackermann steering system and articulated steering systems are more maneuverable. The rear tires will automatically follow the tracks of the front tires when the articulation pivot is at the center of the wheelbase and the motion of the articulation system can often be used to work the machine through areas that would immobilize Ackermann steered machines.

2.3.4 Requirements

The requirements for the steering system and handling characteristics that are currently in the military specifications are compared to the new requirements that have been determined in this feasibility study in Tables 34 and 35.

Table 34. Steering System Requirements.

	CURRENT REQUIREMENT	FEASIBILITY STUDY
STEERING SYSTEM TYPE	6k - Ackermann 10k - Articulated	No change No change
STEERING SYSTEM DEADBAND	No specification	+/- 10 degrees max
STEERING SYSTEM REDUCTION RATIO	5.5 turns maximum stop to stop (28:1)	15:1 minimum to 25:1 maximum
STEERING SYSTEM TIME LAG	No specification	.5 seconds maximum
STEERING SYSTEM TIME STOP TO STOP		
-Low idle max.	8 seconds	No change
-High idle min.	4 seconds	20 deg/s minimum
-Roading	No specification	20 deg/s minimum

Table 35. Machine Handling Requirements

	CURRENT REQUIREMENT	FEASIBILITY STUDY
OBSTACLE COURSE SPEEDS		
- #1 With load	15 miles per hour	No change
- #1 No load	20 miles per hour	No change
- #2 No load	Max 1st gear speed	No change
STRAIGHT LINE TRAVEL AT HIGH SPEED	No specification	ISO 5010, Travel through corridor 1.25 times width over tires at max travel speed.
MAXIMUM NUMBER OF STEERING CORRECTIONS	No specification	20 per minute max on ISO 5010
EMERGENCY LANE CHANGE MANEUVER	No specification	Maintain control during rapid lane change at 45 mph

2.3.5 Components

The components required to build an acceptable steering system are readily available as off the shelf items. The design procedure involves selecting the appropriate combination of pumps, valves, cylinders, and actuators that provide the desired characteristics. The closest existing system for this application would be similar to an off highway articulated or Ackermann steered dump truck, depending on which steering systems are used in the final designs.

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2.4 Tires

2.4.2 Feasibility/Availability

Goodyear currently has a 17.5R x 25 AT2A wide base radial tire that could be used on the 6K machine. It is equivalent to a loader tire or earthmover tire with traction type tread (L-2 or E-2). This tire model is currently being used on Osh Kosh "Crash Trucks" at vehicle speeds of 60 mph and maximum travel distance of 30 miles. The tires can be balanced and ground to reduce force variations if required. This tire rounding helped the ride significantly in the crash trucks at 60 mph but may not be needed on the 45 mph RTFLT. The 17.5R x 25 AT2A Goodyear tires can operate at 45 mph continuously on the 6K RTFLT when the tires are inflated to 55 psi. The 20.5R x 25 AT2A wide base radial tire is not in production but will be available within one year. It has the same high speed capability as the 17.5R x 25 AT2A. This larger tire is the same size as the tires used on the M10A forklift. The 20.5R x 25 AT2A Goodyear tires can operate at 45 mph continuously on the 10K RTFLT when the tires are inflated to 70 psi.

Michelin Tire Corporation manufactures a family of wide base radial tires designated "XLB**". These tires are commonly used on "On Highway Cranes" and military vehicles. They are DOT approved to 50 mph and have a maximum speed capability of 62 mph. The "XLB" tires are not commonly used on loaders due to lower sidewall strength. However, normal operating conditions for the RTFLT are not as demanding as typical loader operation. Therefore the 17.5R x 25 XLB** tire could be used on the 6K RTFLT when inflated to 75 psi and the 20.5R x 25 XLB** could be used on the 10K RTFLT when inflated to 75 psi. Standard radial loader tires "XRA*" are available in both the 6K and 10K size. These tires have reinforced sidewalls and are rated for continuous travel up to 40 mph.

Current and proposed tire requirements are listed in Table 36.

Table 36. Tire Requirements Affecting or Affected by 45 mph Capability

	Current RTFLT Requirements	Feasibility Study
Tire Capacity (max load/tire)		
6k Low Speed (empty) -	-	10,000 lb
6k Low Speed (loaded) -	-	15,000 lb
6k High Speed (empty) -	-	10,000 lb
10k Low Speed (empty) -	-	12,500 lb
10k Low Speed (loaded) -	-	17,500 lb
10k High Speed (empty) -	-	12,500 lb
Tire Load Rating		
Rated Load Speed -	5 mph	5 mph
Unloaded Speed -	-	45 mph
Specification for Tire Series	Wide Base or 65 Series Low Pressure Tubeless L-2 Traction Type Tread	Wide Base or 65 Series Low Pressure Tubeless L-2 Traction Type Tread
Floatation Index (with & without rated load) (inflation press. ±30% of TRA yearbook press. at 5 mph)	<25 FI	<25 FI

2.5 Other Affected Vehicle Systems

The vehicle cooling system will have to be sized for the increased horsepower required for roading. Special attention will also have to be given to the design of the hydraulic cooling system. Roading the vehicle does not circulate hydraulic fluid through the hydraulic lines and cylinders for cooling, therefore the total hydraulic cooling load during roading will have to be handled with oil coolers. However, roading during very cold weather may overcool the fluid in the steering circuit resulting in very poor steering response.

The current RTFLT's specifications for vehicle lighting; windshield wipers, washers, and defrosters are adequate for 45 mph. Rearview mirrors are not currently specified but should be considered for roading.

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3 CONCLUSIONS

The overall conclusion is that both the 6K and 10K machines can be designed to travel at 45 miles per hour. The componentry required to build these machines is currently available but a significant design effort will be required to package the powertrain and axle suspensions. Components such as two speed axles that would minimize changes to the existing powertrain can be developed within one year.

3.1 Powertrain

3.1.1 Technical Problems

No major technical problems exist that would prevent development of RTFLTs with 45 mph speed capability.

3.1.2 Trade-offs

High speed operation can be accomplished by use of a wide range transmission or two speed planetary axles, or some combination of these two components. Wide range transmissions are commercially available today but are larger and weigh more than current RTFLT transmissions. The use of two speed planetary axles to obtain high speeds would require the fewest changes to existing RTFLTs powertrains. However, two speed planetary axles are not commercially available today but could be developed within one year. Note, the proposed powertrain modifications for the 10K RTFLT may limit typical bucket wheel loader operation (such as high pile crowd force). Dry disc type brakes commonly used on RTFLT type vehicles today may not be sufficient for high speed operation. Higher capacity drum and wet disc brakes capable of meeting nearly any braking specifications are available but at a higher cost.

3.1.3 Commercial Availability and Compatibility

Generally, the needed commercial powertrain components for the smaller 6K RTFLT are more readily available than components for the 10K RTFLT because the 6K-sized components are already used in higher speed applications. As a result, a 6K RTFLT prototype could be assembled more easily than a 10K RTFLT prototype. Two speed planetary axles (steerable and rigid) were the only powertrain components evaluated that are not commercially available today.

3.1.4 Inherent 6K and 10K Differences

Power requirements are approximately 20 percent greater for the 10K RTFLT than the 6K RTFLT due to the increase in vehicle weight. Steerable planetary axles are required for the 6K RTFLT and rigid planetary axles are required for the articulated 10K RTFLT.

3.2 Suspension

A suspension is required for both the 6K and the 10K RTFLT vehicles to be controllable (based on wheel hop) on a class D road surface at 45 mph and to meet the required ride quality limit of 6 watts total operator absorbed power at 45 mph on a class C road surface. Commercially available suspension components exist that meet the high speed roading suspension requirements for both the 6K and 10K size vehicles. However, adding a suspension for the high speed roading capability will compromise material handling capability due to large suspension deflections under load carrying conditions unless the suspension is locked out. There is little difference in suspension components required for the two types of vehicles due to the similar vehicle weight.

The desired (but not required) 6 watt operator total absorbed power limit on the class D road cannot be met by adding axle suspension to either the 6K or the 10K RTFLTs.

3.2.1 Technical Problems

It is not anticipated that any insurmountable technical problems would prevent a 45 mph capable suspension from being built. However, it is unlikely that a suspension element can be found that provides the appropriate spring rate for the unloaded high speed travel axle loads, and can also handle the axle loads carrying rated load without undergoing excessive deflection. Therefore, it will be necessary to lock out the suspension system while in material handling operation. The gas / oil suspension system currently used on some construction equipment (e.g. Caterpillar 615 wheel tractor scraper) incorporates a hydraulic lockout capability. Other types of suspension components, e.g. elastomeric springs, air springs, or coil springs would require that a separate lockout system be incorporated.

3.2.2 Trade-offs

Suspending the front axle alone allows the 6K and the 10K RTFLT vehicles to meet the class C road surface ride quality requirement and the class D road controllability (wheel hop) requirements at 45 mph and allows the use of an oscillating rear axle for good stability on uneven terrain. A front axle suspension that allows vertical motion of the axle but does not allow roll motion of the axle relative to the vehicle frame is preferred over an axle that allows roll motion. The roll motion of the axle slightly reduces the motion of the operators station when encountering large variations in terrain between the left and right sides of the vehicle. However, this advantage is outweighed by the possible detriment to the handling characteristics at high speeds and side to side loaded stability.

Suspending both front and rear axles does allow greater tire out-of-roundness to be accommodated for a given suspension stiffness than when only the front axle is suspended. However, when both axles are suspended the machine is not able to keep all wheels on the ground when on uneven terrain as well as an oscillating rear axle.

3.2.3 Commercial availability and compatability

Several types of suspension components exist which have the required characteristics for the high speed roading capability (e.g. air springs, elastomeric springs, and gas / oil systems). The gas / oil system looks the most promising due to its built in lock out capability and packaging flexibility. The elastomeric spring may not have the loaded / unloaded vehicle load range capability required unless it is always locked out under loaded vehicle conditions. Both air springs and elastomeric springs may present packaging problems due to the relatively large diameters required for the axles loads that will be encountered and the need for an external lock out mechanism.

3.3.4 Inherent 6K and 10K differences

The 10K vehicle is more sensitive to fore-aft pitch than the 6K due to the higher operator's station. The greater fore-aft pitch coupled with high sensitivity of humans to fore-aft vibration results in higher fore-aft operator total absorbed power in the 10K than in the 6K vehicle.

Suspension requirements for the 6K and the 10K are similar due the relatively close vehicle weights and wheelbase. Wheelbase was not varied in this study. Vehicles with longer wheelbases than those studied here for the 6K (110 inches) and 10K (108 inches) may have better ride qualities.

3.3 Steering System

The overall conclusion on the steering system is that one can be built from existing components that will be stable at high speeds and provide adequate steering response. There are relatively minor differences between the characteristics required for an articulation steered machine versus an Ackermann steered machine. The only significant difference between the components required for the two types of systems is that the articulated components will have to be significantly larger and capable of handling higher loads.

3.3.1 Technical Problems

It is not anticipated that there will be any significant problems with the steering systems. A wide range of components are available as off the shelf items to build a stable system. There may be problems getting the right components matched to give the desired response characteristics, particularly in very cold applications. In such applications, a warm up period is often required for machines of the type in this study.

3.3.3 Commercial Availability and Compatibility

All of the required components are currently available and can be made compatible with relatively little effort. Some current systems may be adequate as they are in the machine. Others may require no more than a change in the control valve section to meet the requirements for controllability at high speed. Some existing off highway trucks have adequate characteristics for controllability at 45 miles per hour. Many existing wheel loaders and rough terrain fork lift trucks are slightly over the .5 second lime lag limit.

3.3.4 Inherent 6K and 10K Differences

The main difference between the systems required for the 6k and 10k machines is the magnitude of loads that will be imposed on the system. The 10k with articulated steer will have significantly higher loads induce during steering maneuvers than the 6k with an Ackermann steering system. This means that the 10k will require larger capacity components. Articulated machines have their steering systems designed to handle loads higher than those encountered during even the most severe steering maneuvers. An articulated steering system is simpler to design and build and is more reliable than Ackermann steering systems, particularly in severe applications. A few other differences between articulated and Ackermann steer are discussed in section 2.3.3.2.5.

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3.4 Tires

3.4.1 Technical Problems

No technical problems exist that would prevent sustained 45 mph RTFLT operation based on tire limitations. Tires can be balanced and ground if out of round to reduce force variations to acceptable limits at high speed.

3.4.2 Trade-offs

Tires that can withstand continuous 45 mph operation generally have lower sidewall strength than typical loader tires. However, the proposed high speed tires should still provide sufficient life based on the RTFLT application (typical loader tire sidewall strength should not be required since the RTFLTs will be used for container handling only). High speed operation requires higher tire inflation pressures which results in decreased floatation unless the inflation pressure can be adjusted to the vehicle's environment.

3.4.3 Commercial Availability and Compatibility

Large radial tires with high speed capability are becoming more commercially available. Tires are available for both the 6K and 10K RTFLTs from at least one supplier today. Within a year, two suppliers will have tires available for both machines.

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3.5 Other Affected Vehicle Systems

The vehicle cooling system will have to be designed for the increased horsepower required for roading. The hydraulic cooling system will have to handle cooling requirements during roading both in hot and cold weather. But no new cooling system component technology is required.

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4 RECOMMENDATION

The US Army Belvoir Research & Development Center should issue RFPs for procurement of both 6K and 10K RTFLTs with 45 mph capability for demonstration of concept.

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5 CONTACTS

Several manufacturers were contacted for information concerning availability of components for the rough terrain fork lift trucks with 45 mph capability. Following is a list the companies contacted and the names of the main contact at each company.

Caterpillar Inc.
William H. Zimmerman
(309) 675-6275
Complete Vehicle

Roger R. Warner
Clark Components Co.
(616) 697-4467
Complete Vehicle

Robert E. Wellman
Twin Disc Inc.
(414) 634-1981
Torque Converters, Transmissions

Ted Kaufman
Dana Corp.
(219) 483-3059
Axles, Brakes, Universal Joints & Driveshafts

Gene Wright
Funk Manufacturing
(316) 251-3400 ext. 154
Torque Converters, Transmissions

Frank Coronado
Rockwell International
(313) 435-7705
Axles, Brakes, Universal Joints & Driveshafts

Al Musci
Goodyear Tire & Rubber
(216) 796-3868
Tires

Lou Arbore
Michelin
(803) 234-5285
Tires

Ronald C. Anderson
Firestone
(317) 773-0650
Airide and Marsh Mellow springs

Two manufacturers that were contacted did not respond to our requests. These companies are:

John Deere

Dresser Industries

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