AN ABSTRACT OF THE PAPER OF

EDWIN ARTHUR STRYKER for the degree of MASTER OF FORESTRY

in FOREST ENGINEERING presented on February 28, 1977

Title: GRADEABILITY OF LOG TRUCKS

Abstract approved: John E. O'Leary

John E. O'Leary

Gradeability of log trucks is limited by either vehicle rimpull or ground-tire adhesion. The analysis presented shows that log trucks in current use in the Pacific Northwest are limited by tire-slip gradeability rather than by rimpull. Analytic techniques were used to determine that gradeability is greatest for piggyback and least for empty truck configurations. Effects of horizontal curvature and superelevation rates on truck "seen" grade were analyzed as well as road design gradient limitations. Road design criteria graphs are presented in the appendices by log truck configuration for varying curve radii, superelevation rates, and centerline gradients.

LIST OF SYMBOLS

Symbol		Units
A	horizontal distance to center of mass	ft
A_T	vehicle translational acceleration	ft/sec^2
c_1	tire force	lb/deg
D	engine displacement	in^3
D_1	momentum grade stall point	ft
D_2	momentum grade shift point	ft
D_{TE}	drive train efficiency	none
E	superelevation rate	ft/ft
g	gravitational acceleration	ft/sec^2
G	grade in percent	%
GR	total gear reduction	none
$\hat{G}s$	maximum "seen" grade allowable	%
h	vertical distance to center of mass	ft
HP	horsepower	ft-lb/min
i	coefficient for axle (f-front, d-drive, t-trailer)	none
I_{zz}	polar moment of inertia (engine)	ft-lb-sec^2
I_T	polar moment of inertia (tire)	ft-lb-sec^2
L	tractor wheelbase	ft
$L1$	tandom spacing	ft
$L2$	stinger length	ft
$L3$	bunk spacing	ft
$L4$	steering axle to trailer axle	ft

List of Symbols -- continued

Symbol		Units
Te	engine torque	ft-lbs
TR	traction ratio	none
V	velocity	ft/sec
W	gross vehicle weight	lbs
W1	total weight on tandom axles	lbs
X	cramp angle	degrees
Y	tractor width	ft
Y1	road width	ft
μ	coefficient of traction	none
θ	grade in degrees	degrees

List of Symbols -- continued

Symbol		Units
L5	variable reach length	ft
n	number of tires on the ground	none
N	number of drive train components	none
Ni	normal force on ith axle	lbs
R	horizontal curve radius	ft
R_A	air resistance	lbs
R_{AT}	auxiliary transmission gear ratio	none
R_C	cornering resistance	lbs
R_{IT}	translational inertia resistance	lbs
R_{IR}	rotational inertia resistance	lbs
R_I	inertia resistance	lbs
R_G	grade resistance	lbs
R_L	loaded tire radius	ft
R_{MP}	rimpull	lbs
RPM	revolutions per minute	r/m
R_{RE}	rear-end gear ratio	none
R_{Ri}	rolling resistance for ith axle	lbs
R_S	total resistive forces	lbs
R_T	main transmission gear ratio	none
R1	radius to trailer axle	ft
R2	radius to drive axle	ft
R3	radius to steering axle	ft
T	tractive effort	lbs
Ta	axle torque	ft-lbs

LIST OF FIGURES

TABLE OF CONTENTS

LIST OF TABLES

INTRODUCTION

The fundamentals of vehicular motion are well known and documented but they have not been applied to log trucks in a strict sense. Many publications present the fundamentals of motion for heavy transport vehicles (the common "semi") on flexible pavement structures in city and highway environments. Direct application of these procedures will not necessarily represent the motion of the on- or off-highway log transport vehicle.

The modern log truck incorporates "state-of-the-art" technology in vehicular engineering and designs; e.g., diesel engine power systems, transmissions, drive trains, tires, and use of space-age materials of light-weight and high strength. Because of these attributes, modern log trucks are capable of handling gross vehicle weights (GVW's) from 80 kips[1/] to more than 300 kips, from startup to posted speed limits, with high reliability. The physical capabilities of most modern log trucks far exceed the road design limits imposed by most forest road design engineers.

This paper attempts to quantify log truck physical capabilities and road design limitations in a single document. In so doing, reference is provided the forest road

---

[1/] 1 kip = 1000 pounds.

design engineer for analysis of critical grade situations and the effects of road design upon truck performance. Methodology is provided so maximization of potentials can be better achieved in critical design situations.

Gradeability

In order for any vehicle to propel itself over a road surface its tractive effort must be transmitted. The transmission of this force is assumed dependent upon driven tire loading and the coefficient of traction between the two surfaces (Taborek, 1957). For rubber-tired vehicles, the coefficient of traction is greatest when the tire is actually slipping slightly over the road surface. This coefficient will change as the tire is made to slip in a given direction (forward, backward, or sideways) and, consequently, an exact value is impossible to determine (McNally, 1975). Tire-road adhesion coefficients (traction coefficients) are presented in Table 3.

To see if gradeability does depend upon this coefficient of traction and upon the weight on the driven wheels, it will be analyzed under two different conditions:

1. Gradeability at tire spin-out.
2. Gradeability at maximum rimpull.

A comparison of grade capability under these two conditions will be made and conclusions drawn as to what situations warrant the application of either formulation.

TIRE-SLIP GRADEABILITY

To evaluate gradeability at spin-out, two assumptions are required:

1. Tractive effort is limited by tire-slip and not by drive axle torque.
2. Acceleration of the vehicle is zero at the instant on grade of tire-slip.

The following free body diagram (FBD) depicts this situation and will aid in the formulation of equations.

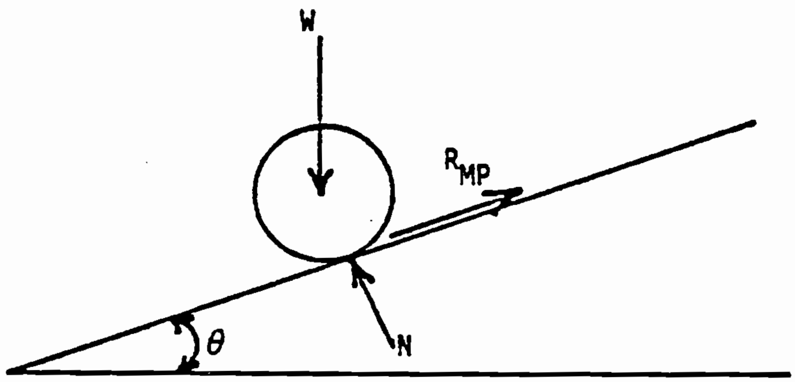


Figure 1. FBD of a driven wheel.

Symbols used in Figure 1: W = weight on wheel

R_{MP} = rimpull

N = normal force

θ = grade in degrees

Balancing forces perpendicular to the ground,

$$N = W\cos\theta.$$

Balancing forces parallel to the ground,

$$R_{MP} = W\sin\theta.$$

At spin-out,

$$R_{MP} = \mu N.$$

Combining equations,

$$\mu = \tan\theta$$

$$\%\ \text{Grade} = 100\mu \qquad \text{(eq. A.1)}$$

This result implies that for a single powered wheel, gradeability is limited by the coefficient of traction. One can conclude that an all-wheel-drive vehicle is limited in its grade climbing ability only by this factor (unless power limited).

Analysis of the common logging truck (with its trailer in the piggyback configuration and rear wheels driven) allows tire-slip gradeability to be formulated as shown in Figure 2.

The result shown in Figure 2 implies that tire-slip gradeability is, again, independent of vehicle weight but dependent upon the coefficient of traction and the position of center of mass. To facilitate utilization of the resulting expression and to put it in terms of more easily measured variables, the term "traction ratio" is introduced.

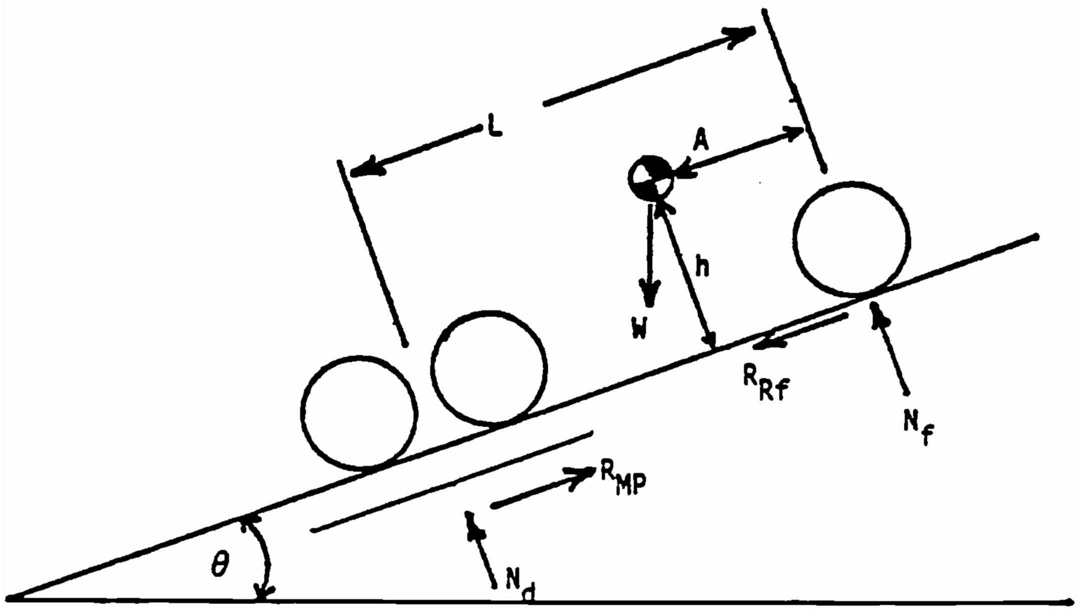


Figure 2. FBD of tractor.

Symbols used in Figure 2: W = GVW (lbs)

L = wheelbase (ft)

R_{Rf} = rolling resistance (lbs)

A,h = position of c.g. (ft)

$$\Sigma Fy = 0: W\cos\theta = N_f + N_d$$

$$\Sigma Fx = 0: W\sin\theta = R_{MP} - R_{Rf}$$

Balancing moments about point C,

$$N_d = \frac{W\cos\theta\ A}{L} + \frac{W\sin\theta\ h}{L}$$

At spin-out,

$$R_{MP} = \mu N_d$$

$$R_{Rf} = 0$$

$$N_d = W\sin\theta/\mu$$

$$\% \text{ Grade} = 100 \left(\frac{\mu A}{L - \mu h}\right) \qquad \text{(eq. A.2)}$$

This ratio is defined as the weight on the driving axle compared to the total weight of the vehicle, or

$$A/L = N_d/W = TR$$

The expression for tire-slip gradeability then becomes:

$$\% \text{ Grade} = 100\mu TR/(1 - \mu h/L) \qquad \text{(eq. A.3)}$$

If one assumes that ($\mu h/L$) is negligible, then one may conclude that tire-slip gradeability is estimated by:

$$\% \text{ Grade} = 100\mu \left(\frac{W_{drivers}}{GVW}\right) = 100\mu TR \qquad \text{(eq. A.4)}$$

In the case of the all-wheel-drive vehicle, TR=1, and the initial result is confirmed. Typical "TR" values found by the author for logging trucks are:

1. Piggyback $TR \doteq .60$
2. Empty $TR \doteq .31$
3. Loaded $TR \doteq .40 - .44$

Log truck drivers have expressed their feeling that tire-slip gradeability is at a maximum for the piggyback configuration. These results seem to confirm their belief.

To formulate an expression for tire-slip gradeability for loaded rear-wheel drive logging trucks, the free body diagram in Figure 3 is required.

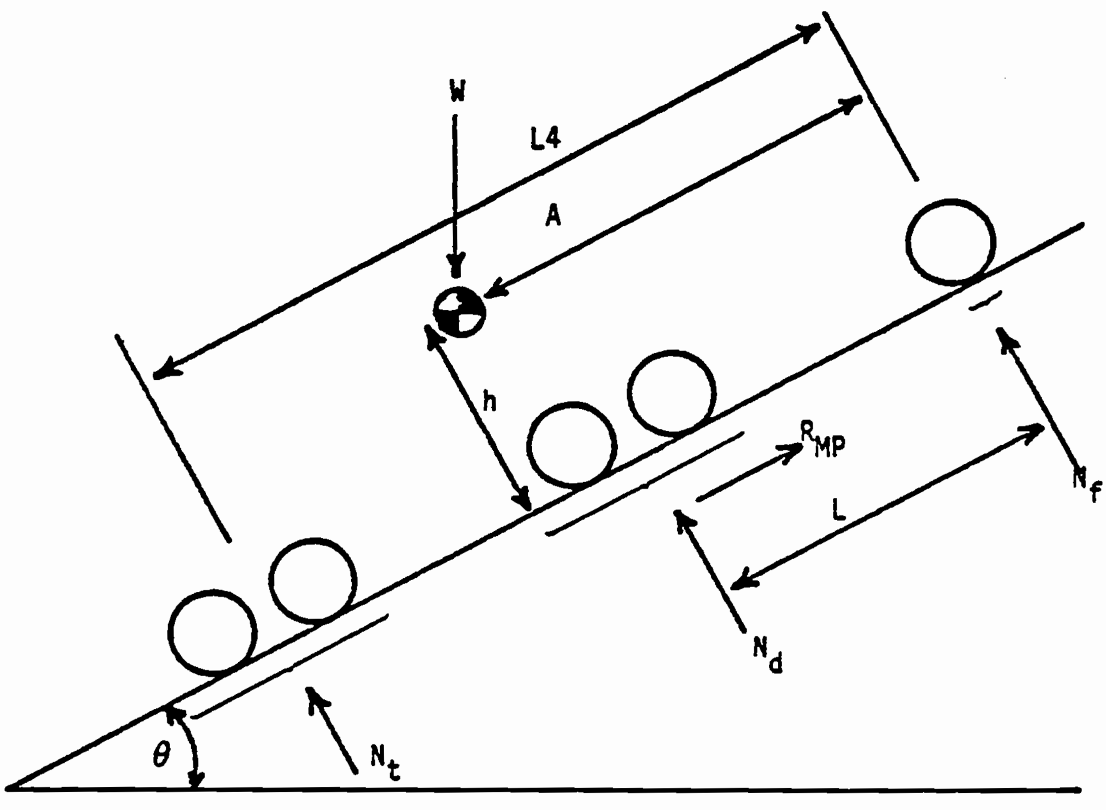


Figure 3. FBD of a loaded log truck.

At tire-slip, static equilibrium is again assumed and the following equations can be written:

$$\Sigma Fx = 0$$

$$R_{MP} = W\sin\theta$$

$$R_{MP} = \mu N_d$$

$$N_d = \frac{W\sin\theta}{\mu}$$

$$\Sigma Fy = 0$$

$$W\cos\theta = Nf + N_d + N_t$$

$$\Sigma M \text{ about the front contact point} = 0$$

$$W\cos\theta\ A + W\sin\theta\ h - N_t\ L4 - N_d\ L = 0$$

$$N_t = N_d$$ [2/]

$$W\cos\theta\ A + W\sin\theta\ h - W\sin\theta\ \frac{(L4 + L)}{\mu} = 0$$

$$\mathrm{Tan}\theta = A\mu/(L4 + L - \mu h)$$

$$\%\ \mathrm{Grade} = 100\ A\mu/(L4 + L - \mu h) \qquad \text{(eq. A.5)}$$

This result implies that at the point of spin-out, gradeability of a loaded log truck is independent of weight, but dependent upon the coefficient of traction and the position of the center of mass.

Sample Calculations

To summarize tire-slip gradeability and to indicate relative grade capability for a log truck assuming:

1. $\mu = 0.5$
2. $A = 12.42$ ft
3. $L = 19.28$ ft
4. $h = 5.0$ ft

[2/] Sampled log trucks confirmed this to be true.

the following result is obtained for the piggyback configuration:

$$\%\ \text{Grade} = 100\ \left(\frac{(.5)(12.42)\ \text{ft}}{[19.28 - (.5)(5)]\ \text{ft}}\right) \qquad \text{(eq. A.2)}$$

$$\%\ \text{Grade} = 37$$

For the loaded log truck assuming:

1. $\mu = 0.5$
2. A = 30.44 ft
3. L = 19.28 ft
4. L4 = 50.7 ft
5. h = 7 ft

the following result is obtained:

$$\%\ \text{Grade} = \frac{(100)(30.44)(0.5)\ \text{ft}}{[50.7 + 19.28 - (0.5)(7)]\ \text{ft}} \qquad \text{(eq. A.5)}$$

$$\%\ \text{Grade} = 23$$

How do these results compare with the quick method outlined by equation A.4?

Piggyback

$$\%\ \text{Grade} = (100)(.5)(.60) = 30$$

Loaded

$$\%\ \text{Grade} = (100)(.5)(.43) = 21$$

The loaded vehicle compares quite favorably, but the piggyback configuration is in error. The assumption that the ratio of "CG" height to wheelbase is negligible is probably not valid for short wheelbase vehicles and consequently equation A.4 should be used cautiously -- conservative estimates result.

The results of this analysis indicate that loaded and piggyback log trucks have differing grade climbing abilities at the point of spin-out; the concept expressed by log truck drivers. Additionally, it appears that tire-slip gradeability is not dependent on total vehicle weight -- only upon the coefficient of traction and position of the center of mass.

RIMPULL GRADEABILITY

Truck Capability

Truck capability will be defined as the ability of a tractor unit to transmit its power from the engine through the drive train to the rims of its driven wheels. Rimpull and tractive effort capability are common expressions for truck capability.

Engine size, type and size of accessories, type of main and auxiliary transmissions, type of drive axle rear-ends, and tire size all relate to rimpull. Once these components are assembled into a given tractor unit, then tractive effort capability becomes a property of that powered unit alone and does not change when the truck or tractor is coupled with various trailers and loading (Western Highway Inst., 1976). To formulate an expression for rimpull requires following the power flow from the engine to the rim of the driven wheels. At the wheels, the forces can be represented by the free body diagram in Figure 4.

The radius of a loaded tire differs from the nominal radius due to the flexure of the tire under load. Tire pressure, load, and tire construction all play important roles in determining the loaded radius of the tire. Knowing the loaded tire radius (R_L) and multiplying it by the rimpull yields the torque at the driving axle. Conversely,

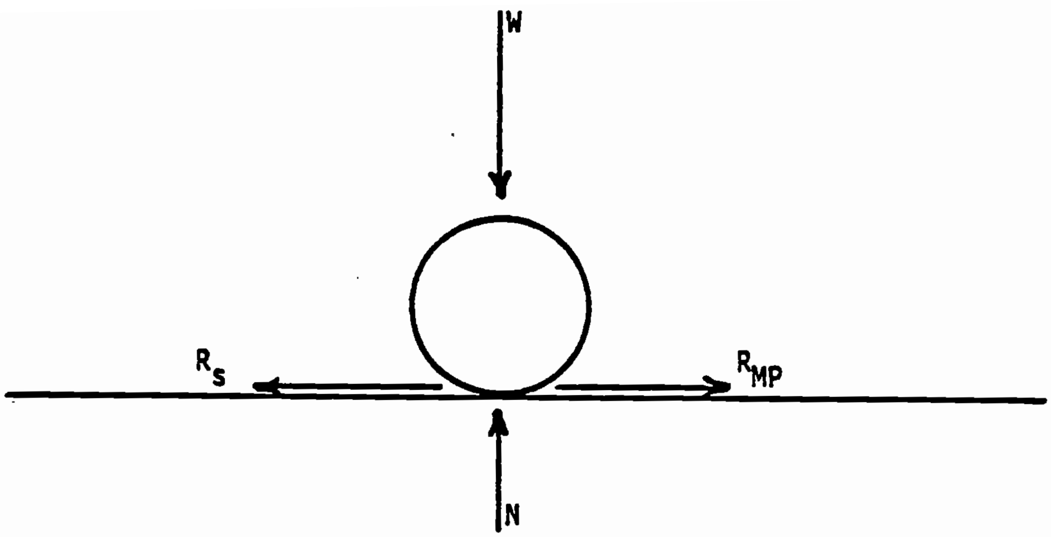


Figure 4. FBD of a driven wheel.

if the available torque at the axle is known, rimpull can be formulated as follows:

$$R_{MP} = \frac{Ta}{R_L} \qquad \text{(eq. 1.1)}$$

The torque available at the drive axles is a function of engine torque multiplied by transmission and rear-end gear ratios discounting friction losses through the drive train.

$$Ta = (Te)(R_T)(R_{AT})(R_{RE})(D_{TE}) \qquad \text{(eq. 1.2)}$$

$$Te = \frac{(5252)(HP)}{RPM} \qquad \text{(eq. 1.3)}$$

where Te = net engine torque at a given rpm (ft. lbs.)

R_T = gear ratio of main transmission

R_{AT} = gear ratio of auxiliary transmission

R_{RE} = gear ratio of rear-end

D_{TE} = drive train efficiency expressed as a decimal

It should be noted that horsepower is net horsepower at the appropriate RPM of the engine. Engine manufacturers graph horsepower-RPM curves for their respective products and the necessary information for these calculations is thus obtained.

Drive train efficiency is a discounting factor utilized to describe the effects of energy losses from engine output shaft through drive axles. Values for drive train efficiency generally fall within the range of 0.75 to 0.95. If unknown, or not readily measured, it can be estimated by:

$$D_{TE} = (1 - 0.05\ N) \qquad \text{(eq. 1.4)}$$

where N is the number of components in the drive train. For example, a main transmission, auxiliary transmission, and double rear-end would yield a D_{TE} = 0.80 (Meyers, 1975). It must be noted that for nonmechanical transmissions, i.e. hydrostatic, engine torque is not only multiplied by drive line gear ratios in the mechanical linkage, but also by the hydraulic capabilities of the specific hydrostatic transmission.

Utilizing the specifications in Table 2 and applying them to the preceding equations allows the calculation of maximum tractive effort which this unit can produce.

$$Te = 950 \text{ ft lbs (limited by transmission)}$$

$$D_{TE} = [1 - (0.05)(3)] = 0.85$$

$$Ta = (950)(12.5)(6.21)(0.85) \text{ ft lbs}$$

$$Ta = 62682 \text{ ft lbs}$$

$$R_{MP} = (Ta/R_L) = 62682 \text{ ft lbs}/1.742 \text{ ft}$$

$$R_{MP} = 35983 \text{ lbs force}$$

Consequently, this typical log truck has 35983 lbs of rimpull which can theoretically be developed at the rims of the driven wheels. This force is an independent measure of the given power unit and will not change regardless of loading and/or trailer arrangement. Whenever the resistive forces to vehicular motion exceed rimpull, then the truck is power limited and without clutch disengagement, engine stall will occur.

Motion Resisting Forces

The forces which oppose vehicular movement are rolling resistance, air resistance, grade resistance, cornering resistance, and inertia resistance. For any vehicle to move at a constant velocity, its rimpull or tractive effort capability must equal the sum of these forces; for a vehicle to accelerate, its rimpull must exceed the sum of these forces; consequently, a surplus of power is required to accelerate a vehicle (either from rest or for velocity changes) or to ascend a grade.

In order to calculate a vehicle's performance characteristics (acceleration, maximum velocity, velocity up a given gradient, gradeability), means of quantifying the resistive forces must be known. Development of these equations will be based upon the free body diagram in Figure 5.

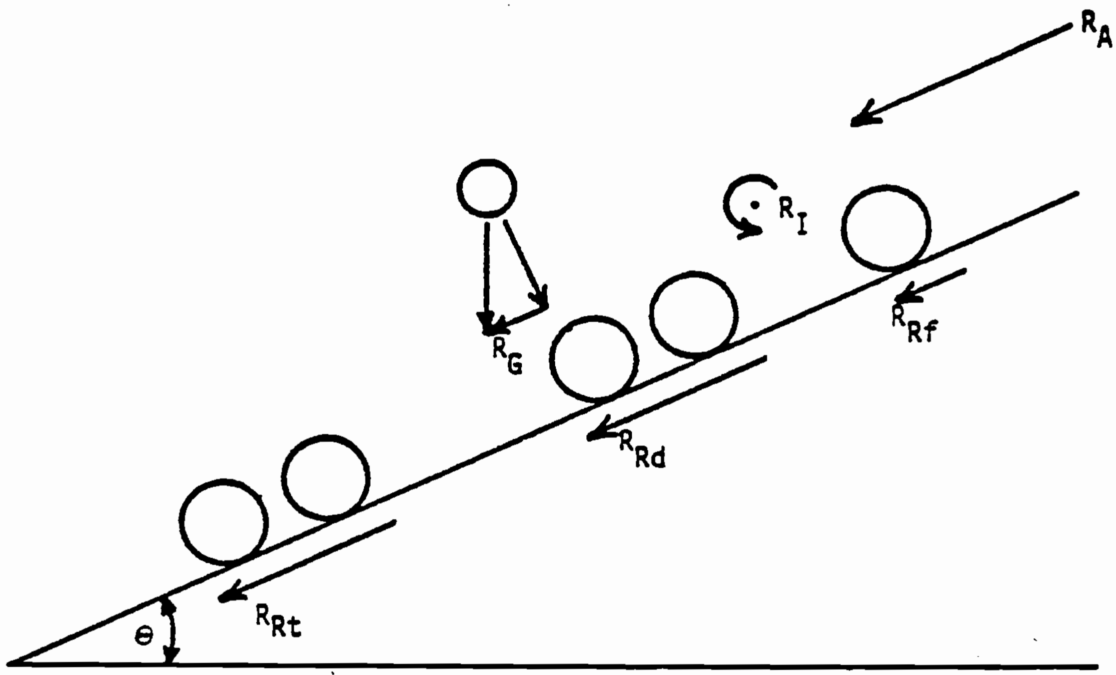


Figure 5. FBD of loaded log truck.

Terms used in Figure 5 are defined as:

R_S = total resistive force

R_R = rolling resistance of steering, driving, and trailer wheels

R_G = grade resistance

R_A = air resistance

R_I = inertia resistance

R_C = cornering resistance (not shown)

Grade Resistance

Grade resistance is resistance offered to movement of a vehicle up a grade. This force is equivalent to the change in potential energy due to plus grade. For each foot of distance traveled up the grade we have the change in potential energy given by:

$$\Delta PE = mgh_2 - mgh_1$$

$$\Delta PE = mg(h_2 - h_1) = mgx$$

$$\Delta PE/\Delta D = mg\ \sin\theta = W\sin\theta$$

where W = GVW of the vehicle

θ = grade in degrees

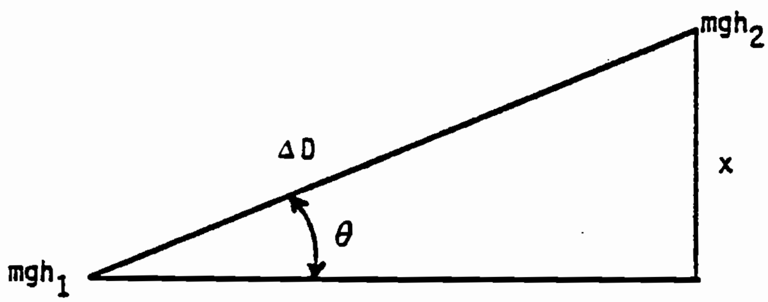


Figure 6. Change in potential energy.

Grade resistance is therefore given by:

$$R_G = W\sin\theta \qquad \text{(eq. 2.1)}$$

Engineers generally express grade as a percent slope which is equal to 100 times the tangent of the angle. For small angles, the tangent $\doteq$ the sine so the equation can be expressed as:

$$R_G = W\ G/100 \qquad \text{(eq. 2.1a)}$$

Rolling Resistance

Rolling resistance is the composite of resistances of an object rolling over a surface. It can be thought of as the force opposing rolling motion and is comprised of (Levesque, 1975):

1. Work to compress and deflect the roadway surface.
2. Work to flex the tire.
3. Work to overcome rolling friction.
4. Work to overcome air frictions, both inside and outside of the tire.

Rolling resistance will vary with tire loading, tire size, wheel bearing friction, and condition of the tires (Western Highway Institute, 1976).

Some authors (Fitch, 1956; Coleman, 1961) choose to treat rolling resistance as a grade equivalent. Others attempt to quantify the variables influencing rolling resistance and develop coefficients which account for them. Three formulations are offered:

Paved surface (SAE, 1975)

$$R_R = W\ (0.0076 + 0.00009\ V) \qquad \text{(eq. 2.2)}$$

Paved surface (Smith, 1970)

$$R_R = W\ (0.0068 + 0.000074\ V)\ S \qquad \text{(eq. 2.3)}$$

Gravel surface (Paterson *et al*., 1970)

$$R_R = W\ (0.0151 + 0.000088\ V) \qquad \text{(eq. 2.4)}$$

where W = GVW

V = velocity in mph

$S \geq 1$ depending on surface

In all formulations wheel loading is by far the dominant variable with vehicle velocity becoming more important at highway speed limits.

Air Resistance

Air resistance is the force opposing vehicular motion relative to the air mass. This can be visualized as a drag force and, since air is a fluid, drag force is proportional to fluid density, viscosity, and surface area perpendicular to movement. Drag force is given as

$$F_D = \frac{C_D \rho V^2 A}{2g} \qquad \text{(eq. 2.5)}$$

where C_D = drag coefficient relating vehicle shape and fluid viscosity

ρ = density of the air mass at some temperature and pressure

A = surface area perpendicular to the direction of motion

V = relative velocity of the object and air mass

g = gravitational acceleration

With velocity expressed in miles per hour and assuming standard temperature and pressure, drag force can be expressed as

$$R_A = K\ A\ V^2 \qquad \text{(eq. 2.6)}$$

where K is a proportionality constant

"K" values for various trucks and trailer bodies have been found to range between 0.00164 and 0.0028. The particular value utilized being a function of the aerodynamic drag characteristics of the truck and the type, number, and placement of the trailer. No specific tests for log trucks could be found, but skin drag on a loaded log truck will be greater than on a metal semi-van body so the higher "K" values are probably in order.

Cornering Resistance

When a rolling tire is made to change directions as in cornering, a drag force is produced. This cornering drag force results from centripetal acceleration and is an added component opposing forward motion. Cornering drag force was

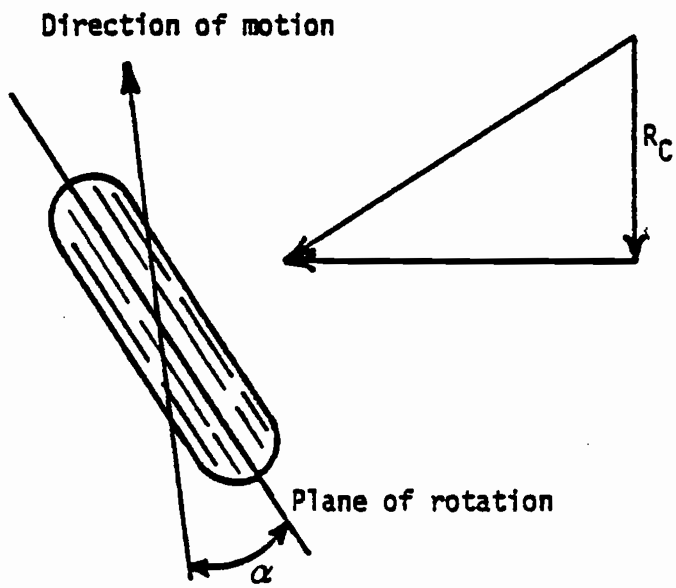


Figure 7. FBD of turning wheel.

initially studied by Smith (1963) empirically by means of drawbar tests. His preliminary findings indicated cornering drag force was of a magnitude generally greater than air and rolling resistance combined. Analytical results by Smith (1970) gave extremely high correlations with his original empirical findings. His formulation for cornering drag force as applied to the entire vehicle follows:

$$R_C = fe[(\frac{\mu W1\ L1}{2R}) + (\frac{1}{n\ c_1})(\frac{W\ V^2}{111\ R})^2] \qquad \text{(eq. 2.7)}$$

where $fe = (1 - \frac{14.97\ R\ E}{V^2}) = 1$ on flat surfaces

R = radius of curve (ft)

V = speed (mph)

W1 = weight on tandem axles (lbs)

L1 = tandem axle spacing (ft)

n = number of tires

E = superelevation rate (ft/ft)

W = GVW (lbs)

μ = coefficient of traction

Cl = tire cornering force = 504 lb/deg loaded

= 189 lb/deg empty

The accompanying graphs demonstrate the effect upon cornering drag force due to curve radius and four super-elevation rates. Smith (1970) also evaluated the effects of lateral wind loading on trucks by this same procedure, but concluded that drag force resulting from the lateral wind component is negligible.

Inertia Resistance

The final resistive force is due to inertia resistance. In accordance with Newton's first law of motion, a body at rest will tend to remain at rest, and a body in uniform motion will tend to remain in uniform motion unless influenced by an outside force. This tendency to maintain its original state can be thought of as the inertia of a body. Whenever a vehicle is accelerated, its inertia must be changed from one state to the next. The inertia of the vehicle is comprised of two components; one associated with translational acceleration of the vehicle and one associated with rotational acceleration of the vehicle's revolving components.

The translational force is easily formulated by application of Newton's second law of motion which says that force is proportional to mass and rate of velocity change.

$$F = m\dot{v} = ma$$

The weight of a body is given as mass times the acceleration of gravity so the resisting force due to translational acceleration is given by

$$R_{IT} = (W/g)A_T \qquad \text{(eq. 2.8)}$$

where W = GVW of the vehicle

A_T = translational acceleration

g = the standard acceleration of gravity; 32.2 ft/sec^2

The force required to overcome the rotational inertia is more difficult to derive. The following analytic development follows that presented by Taborek (1957). The rotating components of a vehicle start at the flywheel of the engine and continue through the drive train to emerge at the wheels. At the contact point between the tires and the road surface, translational acceleration must equal the tangential component of rotational acceleration assuming no slippage. In other words, the instantaneous vehicle velocity is equal to the instantaneous rotation rate of the wheel. We will look at the contact point "B".

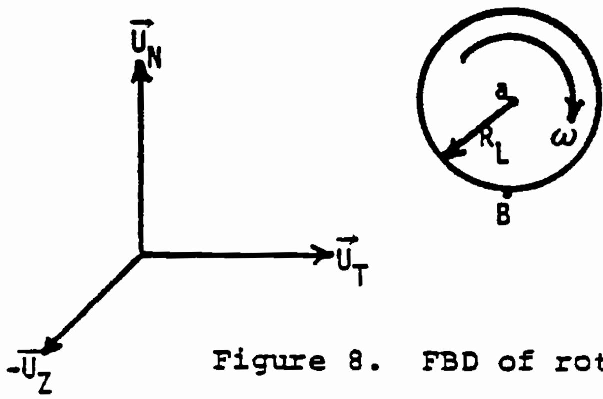


Figure 8. FBD of rotating wheel.

$$V_B\vec{U}_T = V_A\vec{U}_T + (\omega\vec{U}_Z) \times (-r\vec{U}_N)$$

since no slip $V_B\vec{U}_T = 0$

$$V_A\vec{U}_T = -(\omega\vec{U}_Z) \times (-r\vec{U}_N)$$

$$V_A\vec{U}_T = -(-\omega r\vec{U}_T)$$

$$V_A\vec{U}_T = \omega r\ \vec{U}_T$$

$$\omega = \frac{V_A}{r} = \frac{\text{Velocity truck}}{R_L}$$

$$\dot{\omega}_{\text{wheel}} = \frac{\dot{V}_A(t)}{R_L} = \frac{A_T}{R_L}$$

Engine rotation rate is related to tire rotation rate by the gear ratios in the drive train. By similar analysis

$$\dot{\omega}_{engine} = (\dot{\omega}_{wheel})(\text{gear reduction})$$

Furthermore, dynamics principles relate the moment of a rotating mass to its rate of change of angular roation and its polar mass moment of inertia. This can be formulated as

$$M_{engine} = (\dot{\omega}_{engine})(I_{zz})_{engine}$$

by the previous equation, we now have

$$M_E = (\dot{\omega}_{wheel})(GR)(I_{zz})$$

or

$$M_E = (A_T/R_L)(GR)(I_{zz})$$

From the initial discussion of rimpull, recall that wheel torque equalled engine torque times gearing ratios discounting drive line losses. Torque is equivalent to moment, so

$$M_{wheel} = (M_{engine})(GR)(D_{TE})$$

Combining this expression with the previous one yields an expression for the moment at the wheel in relation to the engine as

$$M_W = (A_T/R_L)(GR)^2(I_{zz})(D_{TE})$$

and solving for the inertia resistance force at the ground contact point "B" yields

$$R_{IR} = \left[\frac{A_T}{(R_L)^2}\right](GR)^2(I_{zz})(D_{TE}) \qquad \text{(eq. 2.9)}$$

where R_{IR} = resistance due to rotational engine inertia

A_T = acceleration of the vehicle

R_L = loaded radius of powered wheels

GR = gear reduction

I_{zz} = polar moment of inertia of engine

D_{TE} = drive train efficiency.

Smith (1970) stated that rotating inertias consist primarily of engine and wheel inertias and that the inertias of propeller shafts, axle shafts, and gears are small and can be ignored. The inertia of a four cycle diesel engine is given by (Smith, 1970):

$$I = \frac{1}{32.2}\left[4 + 1.6\left(\frac{D}{100}\right)^2\right] \text{ft lb sec}^2 \qquad \text{(eq. 2.10)}$$

where D = engine displacement, in^3.

Tire inertia is given by graph in the appendices and when applied to the vehicle will yield

$$I_t = (\text{no. tires})(\text{Inertia/tire}) \qquad \text{(eq. 2.11)}$$

It must be remembered that inertia resistance is due to vehicle acceleration or deceleration and is not a factor during constant velocity operations. Combining all expressions for the total resistive forces due to inertia yields

$$R_I = R_{IT} + R_{IR}$$

$$R_I = \frac{WA_T}{g} + [\frac{A_T}{(R_L)^2}][(GR)^2(I)(D_{TE}) + I_t n] \quad \text{(eq. 2.12)}$$

It should be noted that several authors (Taborek, 1957; Levesque, 1975) chose to express inertia resistance in terms of an equivalent mass or "truck-felt" increased weight. This approach may be valid, but their expressions yield increases in equivalent mass for high gear reductions in excess of truck rimpull capabilities. I suspect that the coefficients in their expressions are valid for trucks operating in the higher gears, but not for trucks at low speeds in low gear (at least not at modern truck ratios).

To summarize the forces resisting vehicular motion and to calculate truck performance requires the merging of equations in sections one and two. Combining equations 1.1, 1.2, 2.1, 2.4, 2.6, 2.7, and 2.12, one obtains the equation of motion governing vehicular movement. This equation in symbolic terms is

$$\frac{(T_e)(R_T)(R_{AT})(R_{RE})(D_{TE})}{R_L} = R_G + R_R + R_A + R_C + R_I \quad \text{(eq. 2.13)}$$

Under conditions of constant velocity and no curves, the last two terms drop out and required rimpull for steady state conditions can be calculated. Conversely, knowing the maximum rimpull available allows the acceleration rate for the vehicle to be calculated for turning or non-turning motion. It must be understood that circumstances can prevail where rimpull available exceeds maximum rimpull which can be applied due to adhesion between the driven tires and the road (tire-slip gradeability). In using eq. 2.13, maximum truck limitations are imposed and traction (ground surface adhesion capability) is assumed non-limiting.

For the sampled log truck in section one with an engine displacement of 855 cu-in, a frontal area of 95 sq ft, a GVW of 73,100 lbs, a tire inertia of 7.7 lb ft sec^2, and a velocity of 15 mph in non-turning motion a minimum rimpull of 1260 lbs is required. Maximum rimpull available (35983 lbs) allows an acceleration rate of 4.0 ft/sec^2 under these conditions. Alternatively, ignoring adhesion capacity of the road surface, available rimpull yields a maximum gradeability of:

$$\%\ \text{grade} = 100\ \tan[\text{arc}\ \sin(\frac{R_{MP}}{GVW})] \qquad \text{(eq. 2.14)}$$

$$\%\ \text{grade} = 56.6$$

This far exceeds tire-slip gradeability and it will be found that for modern log trucks, tire-slip gradeability is always limiting.

HORIZONTAL CURVE EFFECTS

Sample calculations to this point have considered only forces involved in rectilinear motion of a vehicle along a path. All roads have curves and those that a logging truck traverses are especially crooked (compared to highways), consequently the effects of curvature must be accounted for if all forces are to be reckoned with.

A logging truck is a "stinger" steered vehicle. Its off tracking characteristics differ from the "semi" since the "semi" trailer pivots around a fifth wheel located slightly forward of the driving axle. The distance forward is called the offset which affects steering forces as well as the total weight distribution on the driving and steering axles. A similar condition is exhibited by the placement of the front bunk on a log truck. It will be found to have positive offset as does the fifth wheel. Again, its position affects steering forces and weight distribution. However, the log trailer is not connected at the front bunk and does not steer about that point. The log trailer steers about the pintel hook by means of its compensator and consequently exhibits less off tracking than a comparable length "semi" or "lowboy."

Off-tracking by stinger steered vehicles can be calculated by the following (Sessions, 1975):

$$R3 = L/\sin X \qquad \text{(eq. 3.1)}$$

$$R2 = L/\tan(x) - Y/2 \qquad \text{(eq. 3.2)}$$

$$R1 = \frac{R2 \cos [\sin^{-1} [\frac{L3}{R2} \cos (\tan^{-1} \frac{L2}{R2}) - \tan^{-1} \frac{L2}{R2}]}{\cos [\tan^{-1} \frac{L2}{R2}]}$$

(eq. 3.3)

where x = cramp angle of outside front wheel

R3 = radius to outside front wheel

R2 = radius to center of truck driving axles

R1 = radius to center of trailer axles

L2 = length of stinger

L3 = distance between bunks

L = wheel-base of tractor

Y = tractor width (out to out of tires)

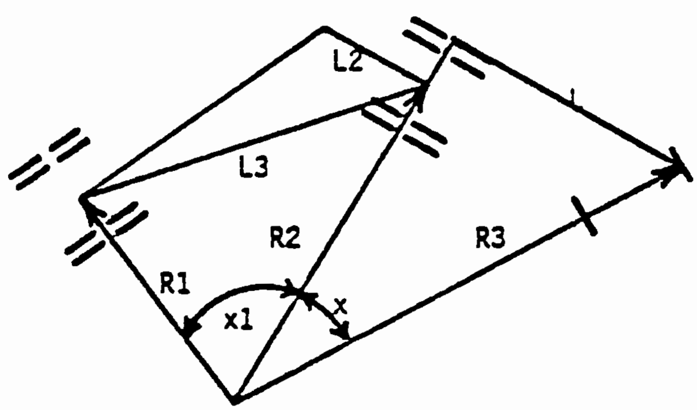


Figure 9. Off-tracking of a loaded log truck.

"Seen" Grade

Application of the off tracking characteristics of a log truck for evaluation of curvature and superelevation of road prisms effects on truck gradeability led to the following analysis. On varying radius of simple horizontal curves and four superelevation rates, the truck was placed upon the curve such that the inside rear dual of the trailer axle matched the inside shoulder of the road prism. The driving axles and steering axles were positioned according to the results of equations 3.1, 3.2, 3.3 and the grade the truck "sees" as it negotiates the curve evaluated. The grade the truck "sees" is defined as the grade between the outside driving axle wheel and the outside steering axle wheel. I chose this definition because comparative analysis could be performed for all vehicle types regardless of configuration of trailers, etc., and because the driving axle is the axle performing the work. This definition yields the minimum "seen" grade by the truck (maximum "seen" grade would be from trailer axle to steering axle).

In order to formulate an expression for "seen" grade, I evaluated the effects of superelevation rate and centerline gradient independently and then combined the results.

Superelevation effects can be found by the relative change in elevation between the outside drive axle tire and outside steering axle tire. In equation form, this becomes:

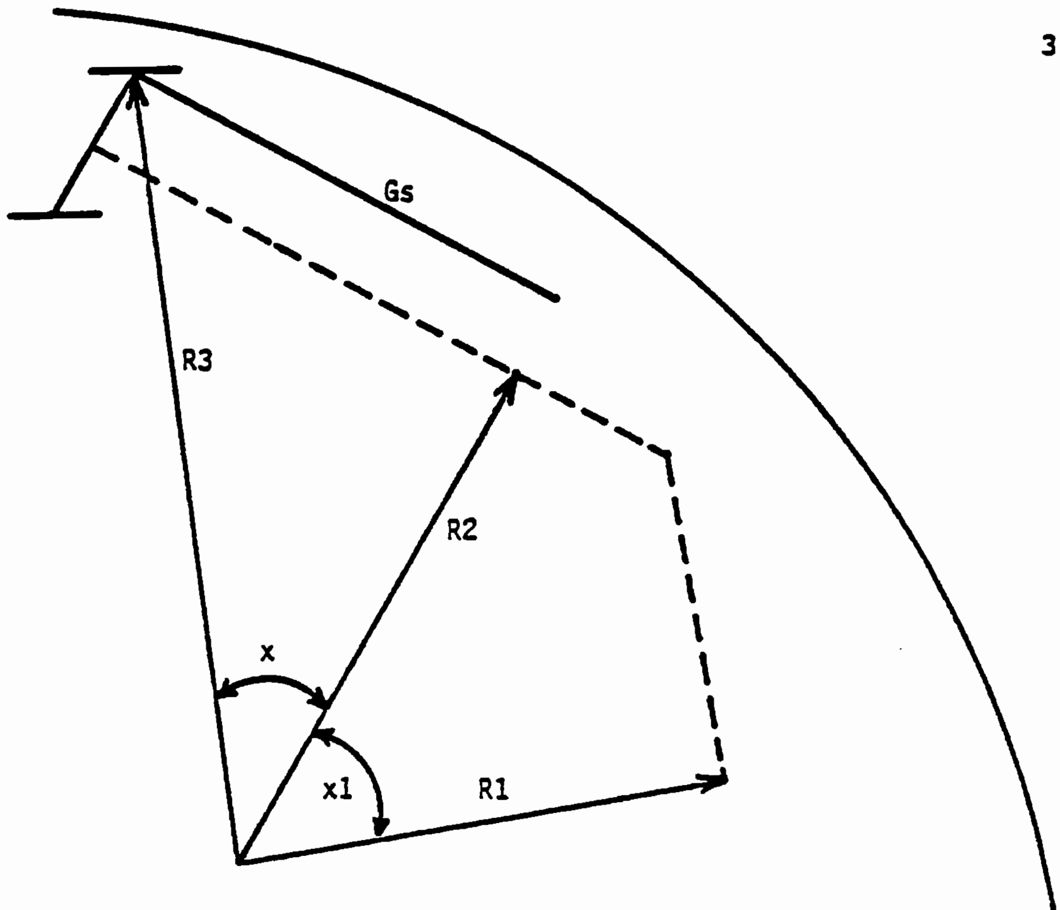


Figure 10. Log truck "seen" grade.

$$(R3 - R2 - Y/2)E = \Delta ELEV(S) \qquad \text{(eq. 3.4)}$$

The relative change in elevation between these two points due to gradient can be found by:

$$\left[\frac{.01G\ \pi}{180}\right][(R3)(X + X1) - (R2 + Y/2)(X1)] = \Delta ELEV(C) \qquad \text{(eq. 3.5)}$$

Combining equations 3.4 and 3.5 and solving for "seen" grade, the expression becomes,

$$Gs(\%) = \left[\frac{(R3 - R2 - Y/2)E + (.01G)(L9 - L8)}{L}\right] 100\% \quad \text{(eq. 3.6)}$$

where

$$L9 = (R3)\left(\frac{\pi}{180}\right)\left[\text{arc } \sin\left(\frac{L}{R3}\right) + \text{arc } \sin\left(\frac{L3}{R2}\cos\left(\tan^{-1} \frac{L2}{R2}\right)\right)\right]$$

$$L8 = (R2 + Y/2)\left(\frac{\pi}{180}\right)\left[\text{arc } \sin\left(\frac{L3}{R2} \cos\left(\tan^{-1} \frac{L2}{R2}\right)\right)\right]$$

The accompanying graphs illustrate that a truck negotiating short radius curves with positive superelevation rates can "see" grades of three to five percent more than the actual center line gradient designed and constructed into the road. One might infer from this development that "negative" superelevation could be a possible method of aiding truck gradeability on adverse haul where steep gradients cannot be avoided. The graphs confirm why designed gradients are significantly reduced at switchback locations and utilization of this technique for known log trucks can result in a more optimum curve gradient rather than a rule of thumb.

Limiting Design Gradient

In order to formulate an expression for the maximum gradient which can be allowed on a superelevated curve, the following method was developed:

1. Use equations 2.1, A.2 and/or A.5 to evaluate the maximum rimpull force which can be developed under stated conditions.
2. Reduce this maximum rimpull force by the sum of the appropriate resistance forces; e.g., rolling resistance, air resistance, cornering resistance, and inertia resistance.
3. Use equation 2.1 to evaluate the maximum "seen" grade which can be negotiated.
4. Use equation 3.4 to evaluate the maximum design gradient allowable based upon the maximum "seen" grade allowed in step no. 3.

The resulting expression for the design limited gradient at centerline follows:

$$G(\%)_{\substack{\text{design}\\ \text{limit}}} = 100\left(\frac{.01\ L\ \hat{G}s - (R3 - R2 - Y/2)E}{L9 - L8}\right)\% \qquad \text{(eq. 3.7)}$$

Use of this method will provide the road design engineer a rational approach for specifying maximum centerline gradient for a known truck configuration, curve radius, and superelevation rate.

MOMENTUM GRADES

A momentum grade is encountered where the grade of the road is greater than the traction limited grade of the vehicle. There exists no means for the vehicle to "pull" itself up that grade as tire slip or "spin-out" would occur. In these situations a vehicle's kinetic energy must be converted into a gain in potential energy and allow the vehicle to "coast" up the increased gradient. This kinetic energy is a finite quantity and eventually will be totally converted; if the truck has reached the top of the momentum grade at this point, then all is well. If not, then stall occurs and the truck is forced to back down the grade. For known gradients, calculations will show minimum velocities for vehicles to attain a specified momentum grade of finite length.

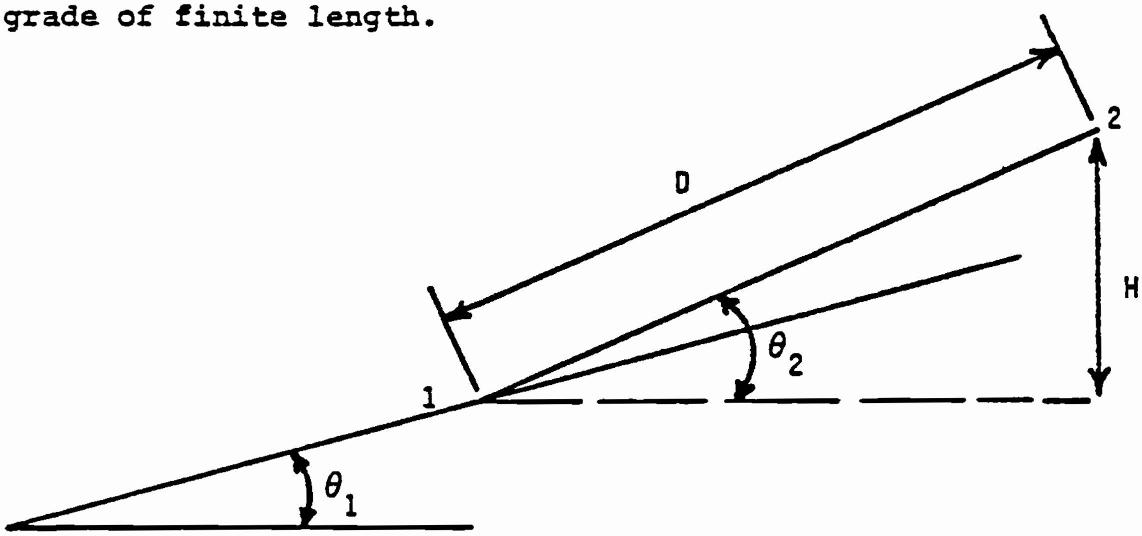


Figure 11. Momentum grade.

where θ_1 = traction limited gradient

θ_2 = gradient to be negotiated which is greater than θ_1

From dynamics we can formulate the conservation of energy axiom that

$$\left[\begin{array}{c}\text{The truck's change}\\\text{in kinetic energy}\end{array}\right] + \left[\begin{array}{c}\text{The work done by}\\\text{the truck's surplus}\\\text{traction}\end{array}\right] = \left[\begin{array}{c}\text{The truck's gain}\\\text{in potential}\\\text{energy}\end{array}\right]$$

Mathematically we have for the system

$$\frac{1}{2} Mv_1^2 + mgh_1 + \int F_{NC} \cdot dr = \frac{1}{2} Mv_2^2 + mgh_2$$

and rearranging terms to match the bracketed axiom yields

$$[\frac{1}{2} Mv_1^2 - \frac{1}{2} Mv_2^2] + [\int F_{NC} \cdot dr] = mg[h_2 - h_1]$$

If we assume that position 1 is where the truck encounters the change in gradient and that position 2 is the top of the hill and is defined at stall, then

mgh_1 = potential energy at position 1 = 0

mgh_2 = potential energy at position 2 = mgH

v_1 = approach speed in ft/sec

v_2 = speed at stall = 0 ft/sec

the equation becomes

$$\frac{1}{2} Mv_1^2 + \int F_{NC} \cdot dr = mgH$$

rearranging yields

$$v_1^2 = \frac{mgH - \int F_{NC} \cdot dr}{.5m}$$

$$v_1^2 = \frac{\cancel{m}g\ D\sin\theta_2 - \cancel{m}g\ \tan\theta_1\ D}{.5}$$

for small angles, $\sin\theta = \tan\theta$; therefore,

$$v_1^2 = 2\,g\,D\ \Delta grade$$

$$D = \frac{.5\ v_1^2}{g\ \Delta grade} = 0.01554\ v_1^2/\Delta grade \qquad \text{(eq. 4.1)}$$

where D = distance in feet to stall point

v = velocity in ft/sec at position 1

$\Delta g = \tan\theta_2 - \tan\theta_1$

Use of equation 4.1 allows the calculation of the maximum distance a momentum grade can be constructed for an initial design velocity with truck stall the result.

Pearce (n.d.) derived a similar expression to allow the calculation of maximum distance a vehicle can ascend a grade before being forced to shift into the next lower gear-step. His expression is:

$$D = \frac{.5m\ v_2^2 - .5m\ v_1^2}{(R_G + R_R)W - R_{MP}} \qquad \text{(eq. 4.2)}$$

where R_{MP} = rimpull calculated for the gear ratio involved at the start of the grade

GRADEABILITY IMPROVEMENT DEVICES

We have seen that the grade climbing ability of a vehicle is dependent upon both the coefficient of traction and the vehicle weight on the driving axles. Consequently, methods to improve a vehicle's gradeability must concentrate in these two areas.

Improvements to the traction coefficient are basically methods to increase traction where it has been impaired due to "weather" below normal limits. Devices such as sanders, studded tires, and chains can be placed directly on the vehicle as well as pre-sanding of the road surface by other equipment. Traction tests at Mt. Hood (Western Highway Institute, 1969) found that either chains and/or sanding on packed snow increased the coefficient of traction from .25 to 0.33 but together no significant additional benefit was derived.

Internal devices available to truck purchasers which aid a given truck in achieving an improved gradeability are:

1. all wheel drive
2. powered trailer dollies
3. non-slip or lock-out rear ends
4. automatic transmissions

These devices will be covered briefly to clarify their respective roles in achieving a more "grade-able" tractor.

All-Wheel Drive

By far the greatest achiever which can be incorporated into a tractor unit is all-wheel drive. Theoretically, all weight is on the driving axles and consequently gradeability is directly limited to the coefficient of traction ($G = \mu\, 100\%$). This option has not been sought by log hauling firms in the Pacific Northwest according to correspondence with major truck dealers. All-wheel drive units are sold by these firms extensively overseas and in the midwest for mining operations. For a Kenworth C500, the additional cost of all-wheel drive is approximately $6600 (Nelson, 1976). Economic analysis for a given operation could confirm or deny the desirability of such a logging vehicle. None have been requested for a logging application to date.

Powered Trailer Dollies

These devices conform to the all-wheel drive concept and attempt to achieve more weight on driven axles. Dollies can be purchased which operate from an engine built into the axle, from an engine slung at some point along the reach, or from the tractor's power unit itself. These dollies generally supply power by pumping hydraulic fluid to hydraulic motors in the axle differential or in the wheel directly. Controls for these auxiliary engines are positioned in the cab of the tractor and most have automatic shut-downs or idle capabilities to prevent over-heating and loss of oil pressure to the auxiliary engine. These units can be controlled to supply power at the driver's convenience for critical situations such as start up, grade climbing, or increased traction on slippery roads. They have seen limited application -- mostly as test units for Freightliner Corporation and its parent company, White Motor Corporation (McNally, 1975).

Inter-Axle Differentials

Axle differentials are required to allow powered wheels to travel at differing speeds in relation to one another on the same axle and in relation to each other from axle to axle in tandem drive. Differential wheel velocities are required due to the different paths (consequently distances) that the wheels must traverse while negotiating curves; they are also necessary to compensate for differences in tire diameters. If differential wheel velocities were not allowed, then tire scrubbing would occur and tire life would be greatly reduced. Also, on curves with tandem drive axles, one axle would drag the other along (McNally, 1975). These facts were recognized long ago and have been incorporated in axle designs for many years.

Due to the nature of differentials, a pitfall develops. By allowing wheels and axles to maintain differing velocities, power must be divided between them. Differential designs call for equal torque splits (50-50) and consequently when one driven wheel or axle encounters slippery conditions, then torque delivered to that unit limits the torque that can be delivered to the less slippery set, as the 50-50 split must be maintained. In other words, the maximum torque that can be applied (in the case of a tandem axle) is four times the least tractive wheel or twice the least tractive axle. If one wheel of a pair were to loose traction completely, then it would spin at twice differential input speed while its mate were stalled. This shortcoming of the common differential has led to a number of designs which try to prevent a vehicle from stalling when one driven wheel loses traction.

Designs to prevent wheel stall can be grouped into those that utilize friction plates (induced friction differentials), those that incorporate over-running clutches, and those that utilize a cam-and-plunger arrangement. Complete descriptions of these designs can be found in Western Highway Institute (1976). In situations where limited traction is available to all driven wheels, provisions must be made to achieve better traction than four times that of the least wheel. To accomplish this, most truck differentials can be equipped with a lock-out feature and nearly all logging trucks (in the Pacific Northwest) are so equipped. When the differential is "locked-out" (not allowed to operate), positive drive to each axle is provided and each axle will drive up to its maximum tractive ability without regard to the other. Utilization of the lock-out feature must be done during low speed operation and should be selected when poor traction is anticipated but not after spin-out has occurred. These devices do not increase the tractor's maximum gradeability, but aid traction during poor conditions.

Automatic Transmissions

Another device achieving greater acceptance in the trucking industry is the automatic and/or semi-automatic transmission. With improved design, fabrication, and maintenance, these non-manual transmissions are increasing in popularity. Since power from the engine must be transmitted through a transmission and capabilities must be incorporated for power interruption (clutch), power flow under critical conditions must be optimized or else the tractor cannot achieve "theoretical" performance. With manual transmissions and dry clutches, driver skill is an extremely important component in matching rimpull to ground capability in a smooth and uniform manner through the gear system. With the advent of wet clutches, torque converters, and automatic transmissions, driver skill can be less than perfect and optimum power flow can still be achieved. Consequently, the basic advantage of the automatic transmission is not in improving traction or gradeability but only ensuring that the tractor can operate at maximum capability. An example is provided by Western Highway Institute (1969) which states the commonly used rule of thumb that at least ten percent more net gradeability is required to start a vehicle equipped with a manual transmission from rest than to keep that vehicle moving at a constant speed on that grade. However, the use of wet clutches, hydraulic torque converters, free-shaft turbines, or hydrostatic transmissions can reduce this percent to about three to five.

CONCLUSIONS

Application of the derived equations and inference from the preceding discussion leads to the conclusion that present day log trucks are traction limited. This is not a new or startling revelation; however, the interesting point is that the limitation is imposed due to the coefficient of traction and not due to lack of rimpull. Modern engines and drive trains have resulted in truck capabilities far exceeding ground-tire adhesion capacity. Consequently, design of forest haul road gradients (in critical situations) should concentrate on methods to improve tractive coefficients. Incorporation of more costly surfacing materials on "critical" gradients may lower total costs by reducing the amount of road construction and taking better advantage of truck rimpull capacities.

Comments are in order on the ramifications of the graphs depicting increases in "seen" grade due to superelevation rates on the one hand, and reduction in cornering drag force on the other. It should be noted that the benefits of positive super-elevation rates in reducing cornering drag force exceed the negative effects of increased gradient. Most forest roads of gravel surface do not incorporate superelevation in their curves and although beneficial in reducing "seen" grade, it costs the vehicle net tractive force due to the increase in cornering drag. An optimum condition probably exists and could be calculated for a given speed and radius of curve. No attempt at optimization is included in this paper, but methodology for computing limiting design gradient criteria is established.

Of significant interest is the concept that tire-slip gradeability is independent of drive-wheel loading. This appears contrary to the accepted laws of friction force at first glance. However, tire-slip gradeability is not wholly independent of vehicle weight — it is dependent on the position of the center of mass and although the "mass" disappears from the equations, its effect is included due to the "lever-arm" distances remaining. Thus, drive-wheel loading is masked in the formulations but the effects remain.

Road design ramifications to be gleaned from the paper include:

1. Maximum gradients should be based upon tire-slip conditions and ultimately, upon the coefficient of traction at the assumed time of haul.
2. Super-elevation of gravel surfaced forest roads is of positive benefit to log trucks, even at low speeds (5 mph).
3. Momentum grade concepts should be looked at on tangent sections to reduce driver shift efforts and to utilize kinetic energies efficiently.

4. Turn-out gradients should differ from road gradients to allow start-up for the returning piggyback log truck.
5. Traction improvement devices are available to the truck purchaser and should be evaluated based upon the economics of his particular situation, and roads designed accordingly.

In summary, the formulations of major interest are:

1. Tire-slip grade for a piggyback log truck

$$G(\%) = 100 \left(\frac{\mu A}{L - \mu h}\right) \%$$

2. Tire-slip grade for loaded truck

$$G(\%) = \left(\frac{100\ \mu A}{L4 + L - \mu h}\right)$$

3. Rimpull gradeability for any log truck

$$G(\%) = 100 \tan[\arc \sin(\frac{R_{MP}}{GVW})]$$

4. Cornering drag force due to curvature and super-elevation

$$R(\text{lbs}) = fe\left[\frac{\mu W1\ L1}{2R} + \frac{1}{n\ c_1}\left(\frac{W\ V^2}{111\ R}\right)^2\right]$$

where

$$fe = \left(1 - \frac{14.97\ R\ E}{V^2}\right) = 1 \text{ for flat surfaces}$$

5. Truck "seen" grade due to curvature and superelevation

$$GS(\%) = \left[\frac{(R3 - R2 - Y/2)E + (.01\ G)(L9 - L8)}{L}\right] 100\%$$

where

$$L8 = (R2 + Y/2)\left(\frac{\pi}{180}\right)\left[\text{arc sin}\left(\frac{L3}{R2} \cos\left(\tan^{-1} \frac{L2}{R2}\right)\right)\right]$$

$$L9 = (R3)\left(\frac{\pi}{180}\right)\left(\text{arc sin}\left(\frac{L}{R3}\right) + \text{arc sin}\left(\frac{L3}{R2} \cos\left(\tan^{-1}\frac{L2}{R2}\right)\right)\right]$$

6. Limiting design gradient

$$G_{\substack{\text{centerline}\\ \text{limit}}}(\%) = \left[\frac{L\ .01\ Gs - (R3 - R2 - Y/2)E}{L9 - L8}\right] 100\%$$

BIBLIOGRAPHY

Billingsley, John D., editor, 1960. Elements of Automotive Engineering. Department of Ordnance, U.S. Military Academy, West Point, New York.

Caird, D., 1973. Notes on Adverse Grades on Logging Roads. Unpublished report, U.S.D.A. Forest Service, Region Six, Portland, Oregon.

Caterpillar Tractor Company, 1976. Caterpillar Performance Handbook, Peoria, Ill.

Coleman, W., 1961. Design and Manufacture of Spiral Bevel and Hypoid Gears for Heavy-Duty Drive Axles. SAE-SP-211, Society of Automotive Engineers, Warrendale, Pa.

Fitch, J.W., 1956. Motor Vehicle Engineering Guide. Arrow Lithographing Company, Chicago, Illinois.

Kenworth Truck Company, 1964. Kenworth Swami, Seattle, Wash.

Lecklider, G.R., and Lund, J.W., 1971. Road Design Handbook. Oregon Institute of Technology, Klamath Falls, Oregon.

Levesque, Yves, 1975. A Deterministic Simulation of Logging Truck Performance. M.F. thesis. University of British Columbia, Vancouver, British Columbia, Canada.

McNally, J.A., 1975. Trucks and Trailers and Their Application to Logging Operations: A Reference Manual. University of New Brunswick, Fredericton, New Brunswick, Canada.

Meriam, J.L., 1975. Statics--SI-Version. John Wiley and Sons, Inc., New York, N.Y.

Meyer, C.F., 1969. Route Surveying and Design. International Textbook Company, New York, N.Y.

Meyers, P.S., 1975. The Diesel Engine for Truck Application. SAE-SP-391, Society of Automotive Engineers, Warrendale, Pa.

Nelson, R.W., 1976. Roberts Motors Company, Portland, Oregon. Personal correspondence.

Paterson, W.G., McFarlane, H.W., and Dohaney, W.J., 1970. A proposed forest roads classification system. Pulp and Paper Research Institute of Canada, Woodlands Papers, W.P. No. 20.

Pearce, J.K., n.d. Forest Engineering Handbook. U.S.D.I., Bureau of Land Management, Oregon State Office, Portland, Oregon.

Peterbilt Motors Company, 1974. Engineering Data Book. Newark, California.

Poe, G.F., 1976. Mack Trucks Inc., Portland, Oregon. Personal correspondence.

Sessions, J.L., 1976. Off tracking of log trucks. Unpublished report, U.S.D.A. Forest Service, Region Six.

Smith, C.E., 1976. Applied Mechanics: Dynamics. John Wiley and Sons, Inc., New York, N.Y.

Smith, G.L., 1970. Commercial Vehicle Performance and Fuel Economy. SAE-SP-355, Society of Automotive Engineers, Warrendale, Pa.

Taborek, J.J., 1957. Mechanics of Vehicles. Machine Design, The Penton Publishing Company, Cleveland, Ohio.

Tannler, A., 1976. White Trucks Division of White Motor Company. Portland, Oregon. Personal correspondence.

Timoshenko, S.P., and Gere, J.M., 1972. Mechanics of Materials. D. Van Nostrand Company, New York, New York.

Western Highway Institute, 1976. Drive Traction Chacteristics of Trucks and Truck Combinations. Western Highway Institute, San Francisco, Calif.

______________, 1970. Traction Characteristics of Trucks and Truck Combinations, 1969 Winter Traction Tests, Report No. 1. Western Highway Institute, San Francisco, Calif.

______________, 1969. Horsepower Considerations for Trucks and Truck Combinations. Western Highway Institute, San Francisco, Calif.

Western Highway Institute, 1969. Horsepower Considerations for Trucks and Truck Combinations, 1969 Acceleration Tests, Report No. 2. Western Highway Institute, San Francisco, Calif.

_______________, 1970. Offtracking characteristics of Trucks and Truck Combinations, Report No. 3, Western Highway Institute, San Francisco, Calif.

_______________, 1971. Tire Wear Characteristics of Trucks and Truck Combinations, Report No. 4. Western Highway Institute, San Francisco, Calif.

APPENDICES

APPENDIX I

UNIQUE FEATURES OF LOG TRUCKS

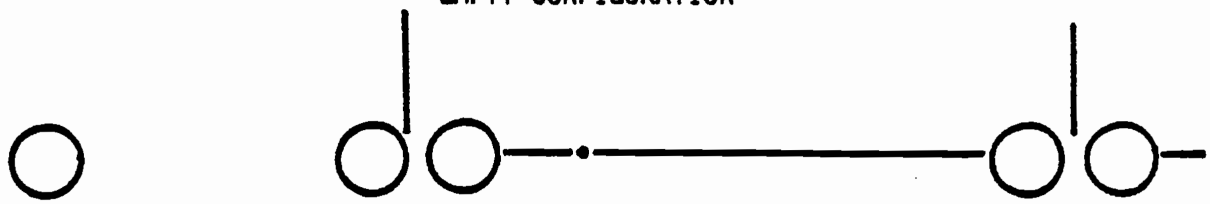
Log trucks utilized in the Pacific Northwest exhibit three distinct features not normally found on the common "semi" highway truck trailer unit. These features are:

1. Ability to carry the empty trailer piggyback.
2. Variable length compensator.
3. Variable bunk spacing or distance between the drive and trailer axles.

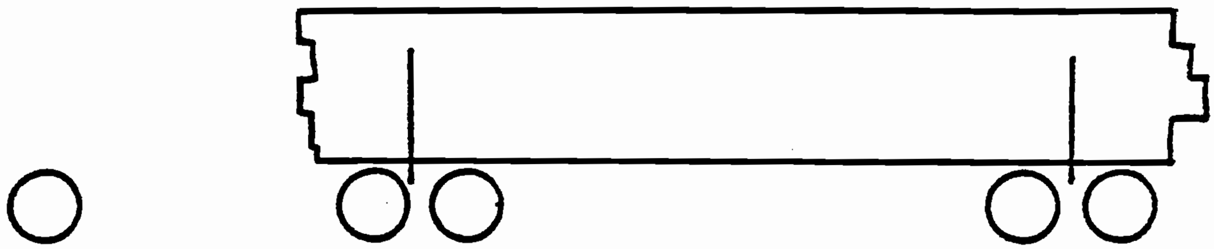
When the trailer is being towed it connects to the tractor's stinger by means of a pintel hook and variable length compensator. This compensator moves in and out of the trailer reach, allowing the trailer reach to extend and contract as necessary for negotiation of vertical and horizontal curves. This variable length reach is a major difference between log trucks and standard semi-type vehicles. A log truck is stinger-steered and its off-tracking characteristics are therefore quite different.

Another unique feature of the log truck is allowance for positioning the trailer duals at any point along the reach. The reach is a box beam of steel construction and slides through a similar box built into the trailer bogie assembly. By loosening a turn-screw at the front of the bogie, the reach can be slid back and forth to allow variable bunk spacing. Consequently, variable log lengths can be accommodated up to the reach positioning limit.

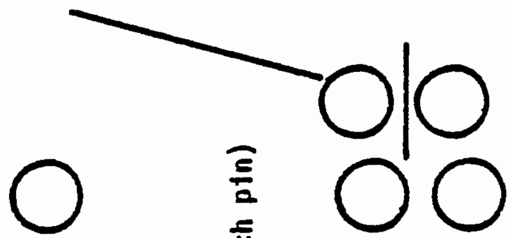
EMPTY CONFIGURATION



LOADED CONFIGURATION



PIGGYBACK CONFIGURATION



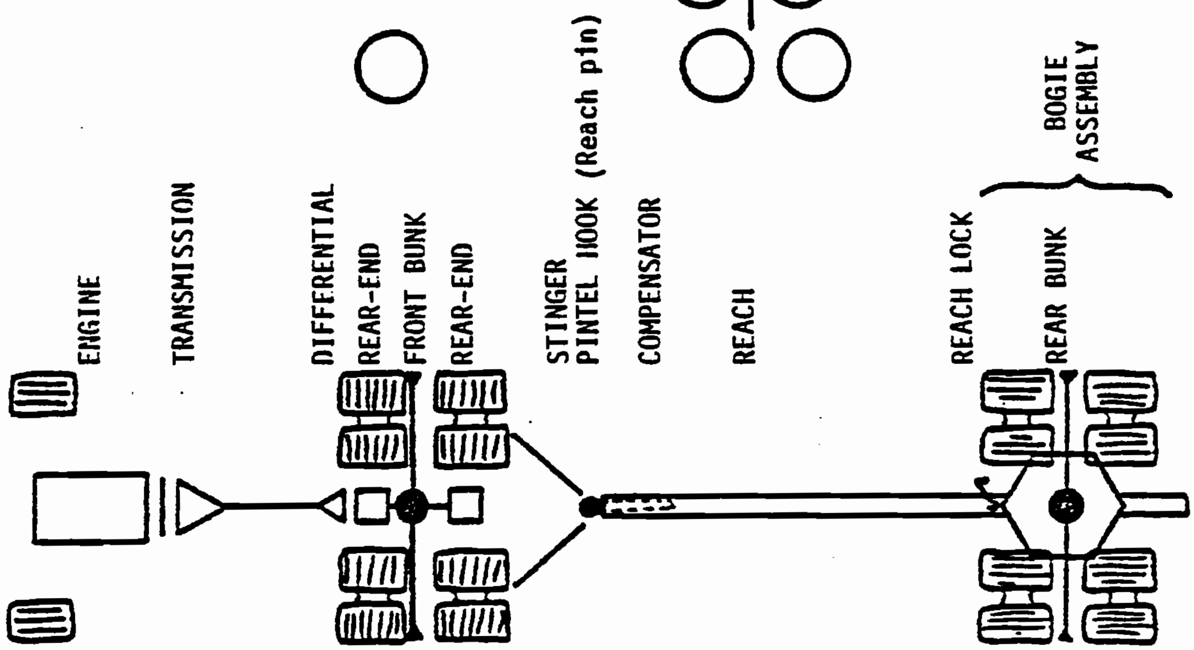


TABLE 1. SAMPLED TRUCK DATA.

Sample points:	Foster, Oregon and Philomath, Oregon
Sample size:	45 loaded log trucks
Sample date:	26 October, 1976

LOADED LOG TRUCKS

Ave. Steering Axle Weight:	9728 lbs
Ave. Driving Axle Weight:	32536 lbs
Ave. Trailer Axle Weight:	32535 lbs
Ave. Gross Vehicle Weight:	74805 lbs
Ave. Gross Log Scale:	6396 fbm

EMPTY LOG TRUCKS*

Est. Steering Axle Weight:	8500 lbs
Est. Driving Axle Weight:	9000 lbs
Est. Trailer Axle Weight:	7000 lbs

PIGGYBACK LOG TRUCKS*

Est. Steering Axle Weight:	9000 lbs
Est. Driving Axle Weight:	16300 lbs

---

*Actual weights could not be obtained for these configurations. The weights are from personal communications with Kenworth Truck Co., Seattle, Wash., General Trailer Co., Springfield, Ore., and North-side Lumber Co., Philomath, Ore.

TABLE 2. LOG TRUCK SPECIFICATIONS*

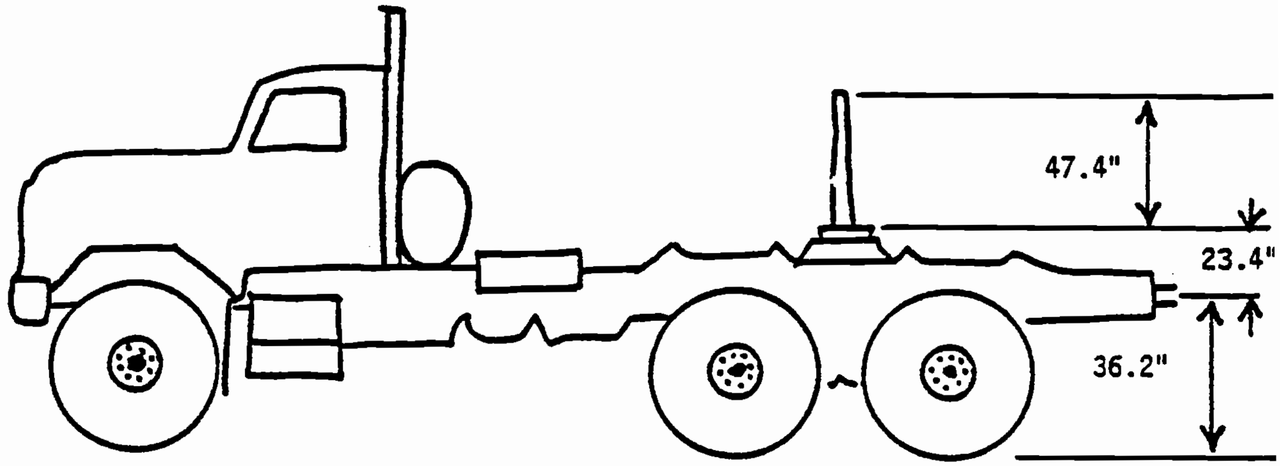
TRACTOR	1975 KENWORTH	MODEL W900
TRAILER	1975 PEERLESS	
OWNER	MEL ROUNDS AND SONS	
WHEELBASE		240 inch
ENGINE	CUMMINS	MODEL NTC-350
TRANSMISSION	FULLER	MODEL RTO-9513
REAR-END	EATON	MODEL DT-380
DRIVE AXLE TIRES		10:00-22
STEERING AXLE WEIGHT		10,300 lbs
DRIVE AXLE WEIGHT		30,900 lbs
TRAILER AXLE WEIGHT		31,900 lbs
LOG SCALE	(10 pieces)	6,543 fbm

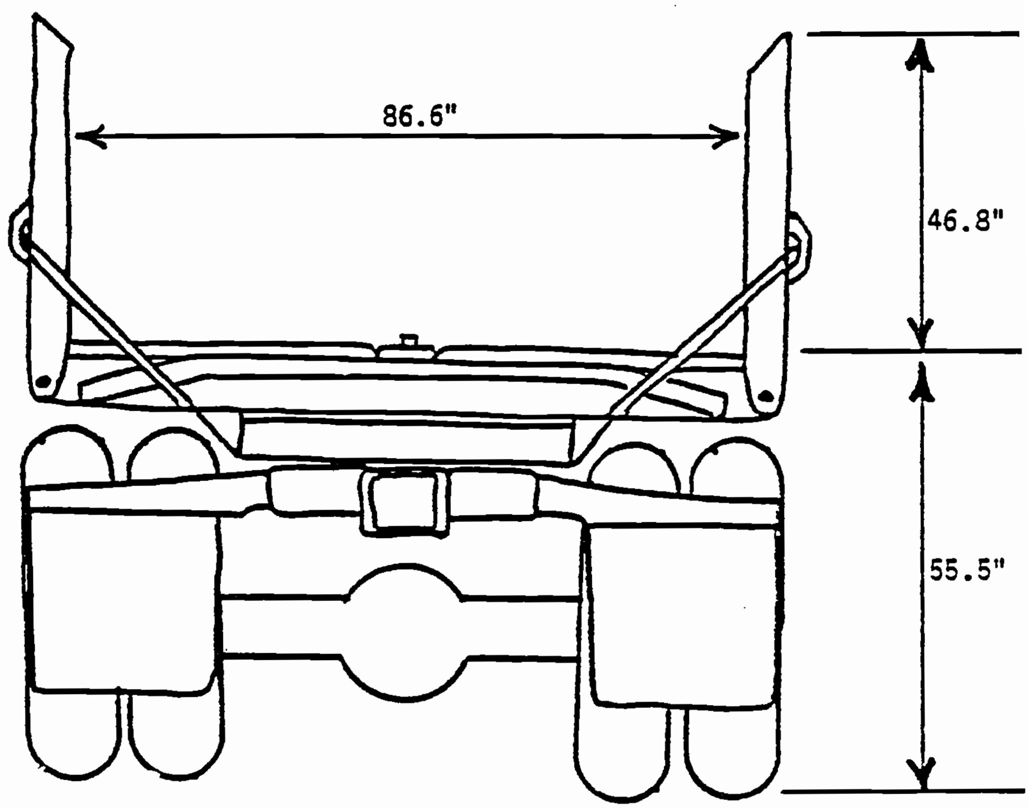
SUPPLEMENTARY DATA FROM MANUFACTURER

NTC-350	MAX TORQUE	1005 ft-lbs @ 1500 rpm
RTO-9513	MAX TORQUE	950 ft-lbs
RTO-9513	LOW GEAR RATIO	12.50:1
DT-380	LOW RANGE GEAR RATIO	6.21:1
TIRES	LOADED TIRE RADIUS	1.742 ft

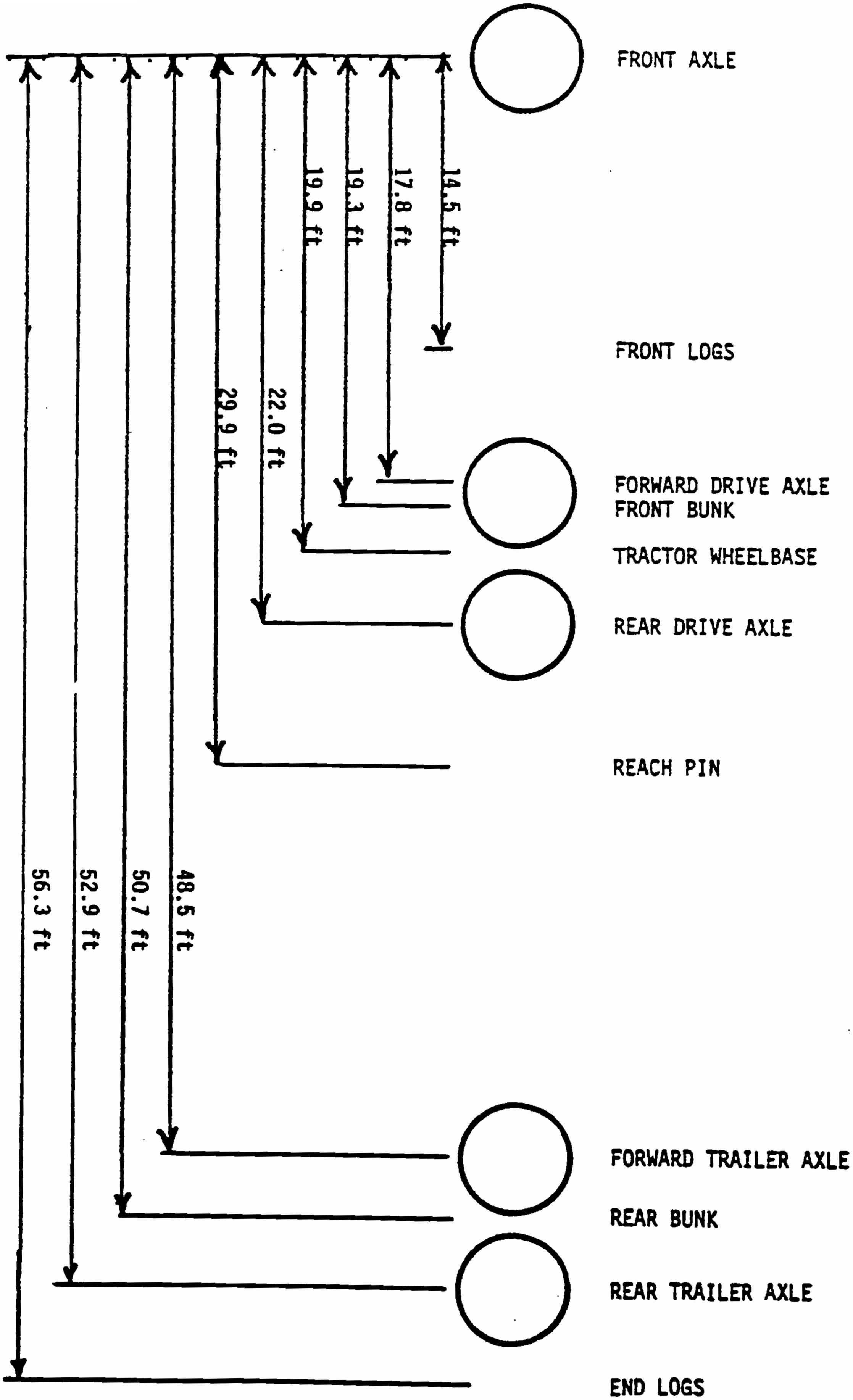
*sampled 26 October, 1976

SAMPLE LOG TRUCK DIMENSIONS

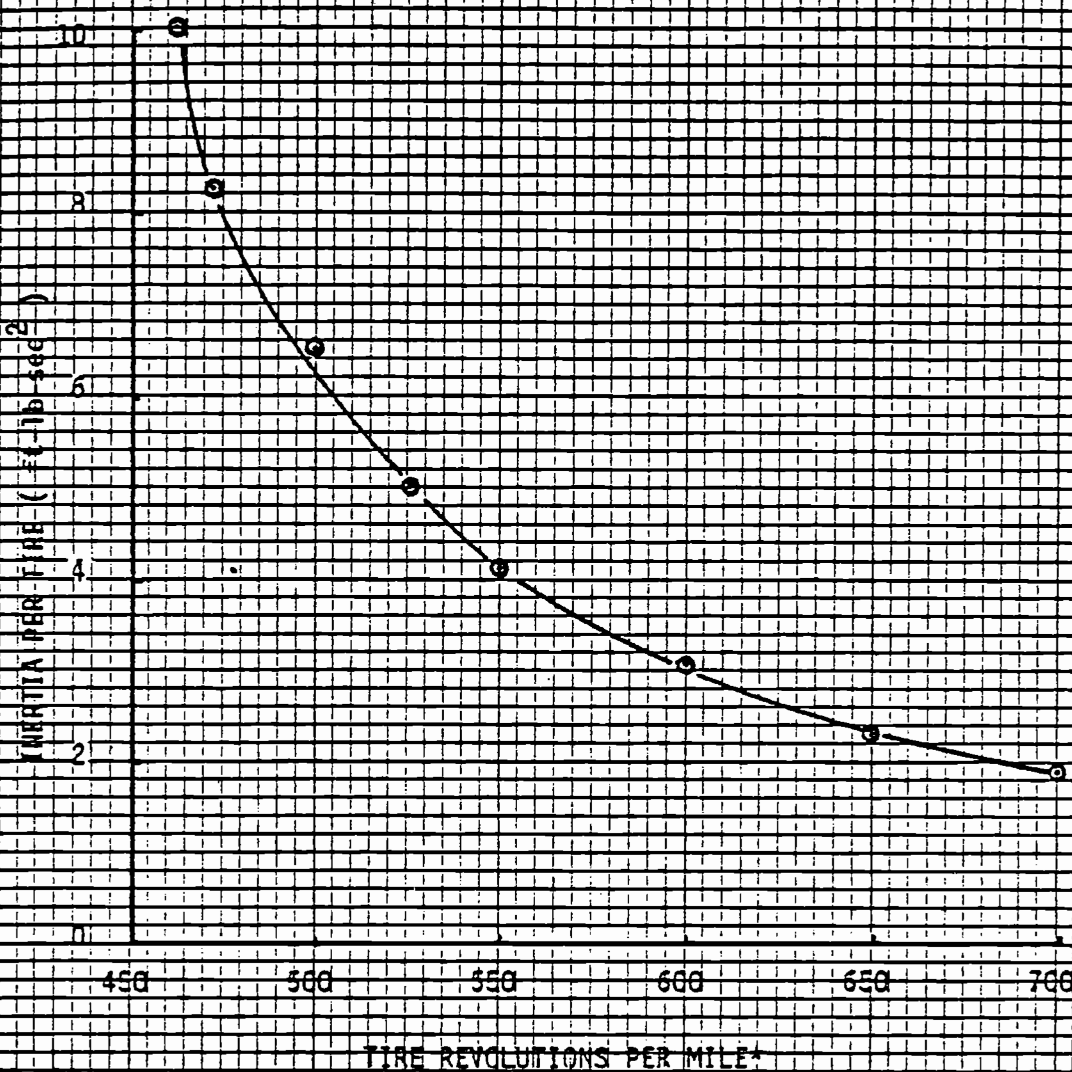




APPENDIX II


FRONT AXLE
14.5 ft
17.8 ft
19.3 ft
19.9 ft
FRONT LOGS
22.0 ft
29.9 ft
FORWARD DRIVE AXLE
FRONT BUNK
TRACTOR WHEELBASE
REAR DRIVE AXLE
REACH PIN
56.3 ft
52.9 ft
50.7 ft
48.5 ft
FORWARD TRAILER AXLE
REAR BUNK
REAR TRAILER AXLE
END LOGS

TIRE INERTIA
(SMITH, 1975)



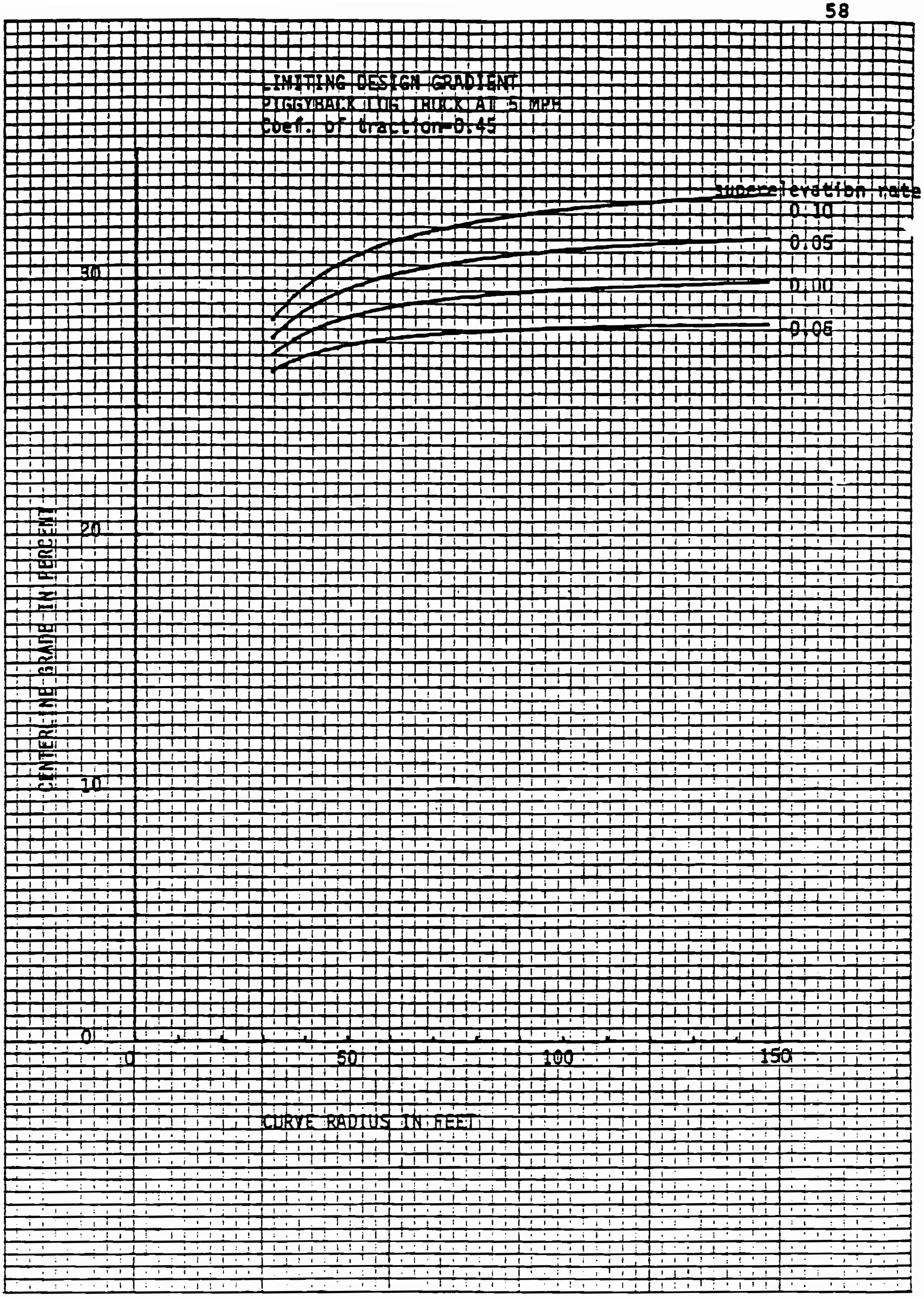
$* \ T_r = \frac{5280}{2\pi Rl}$ where Rl is loaded tire radius in feet

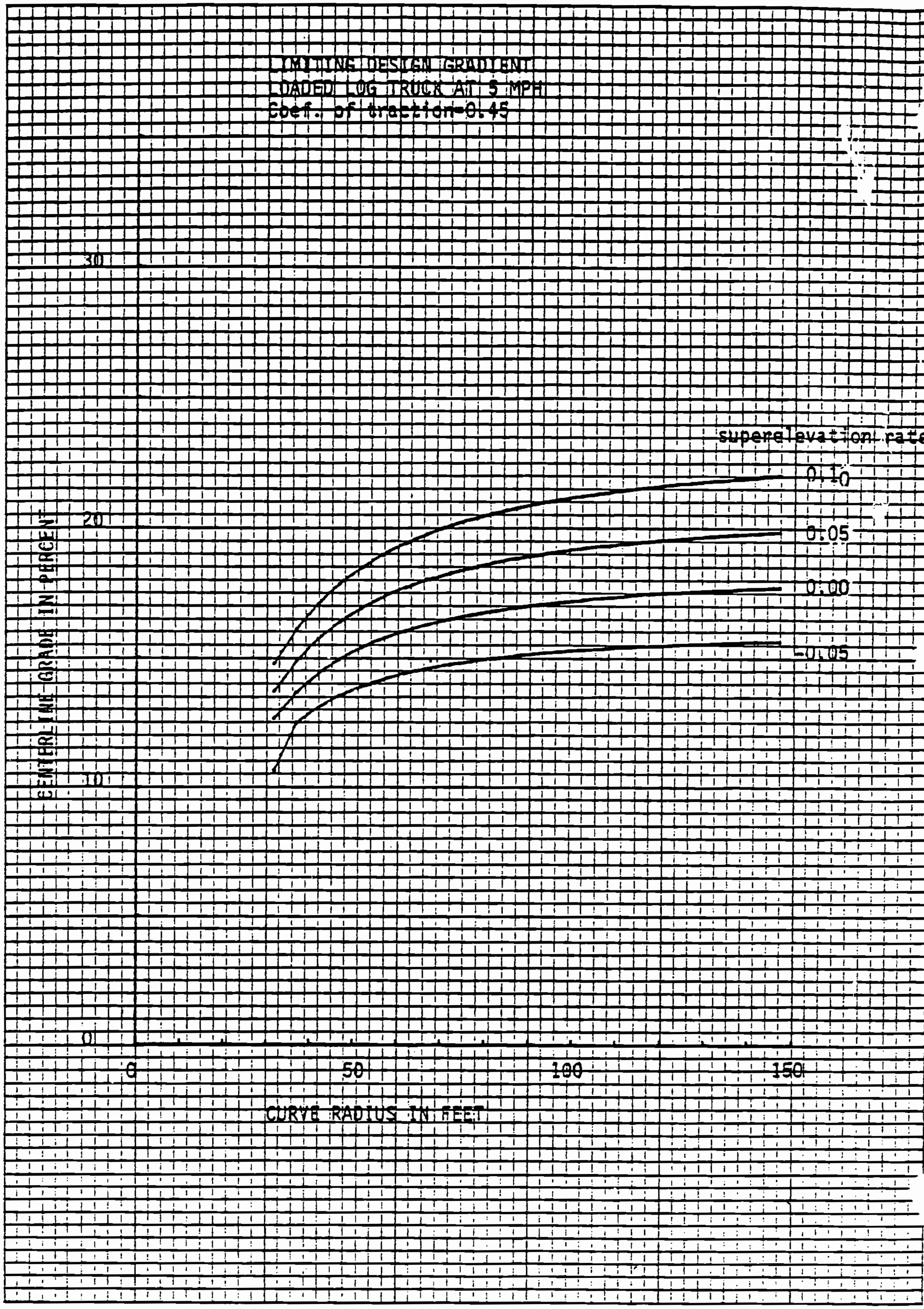
APPENDIX III

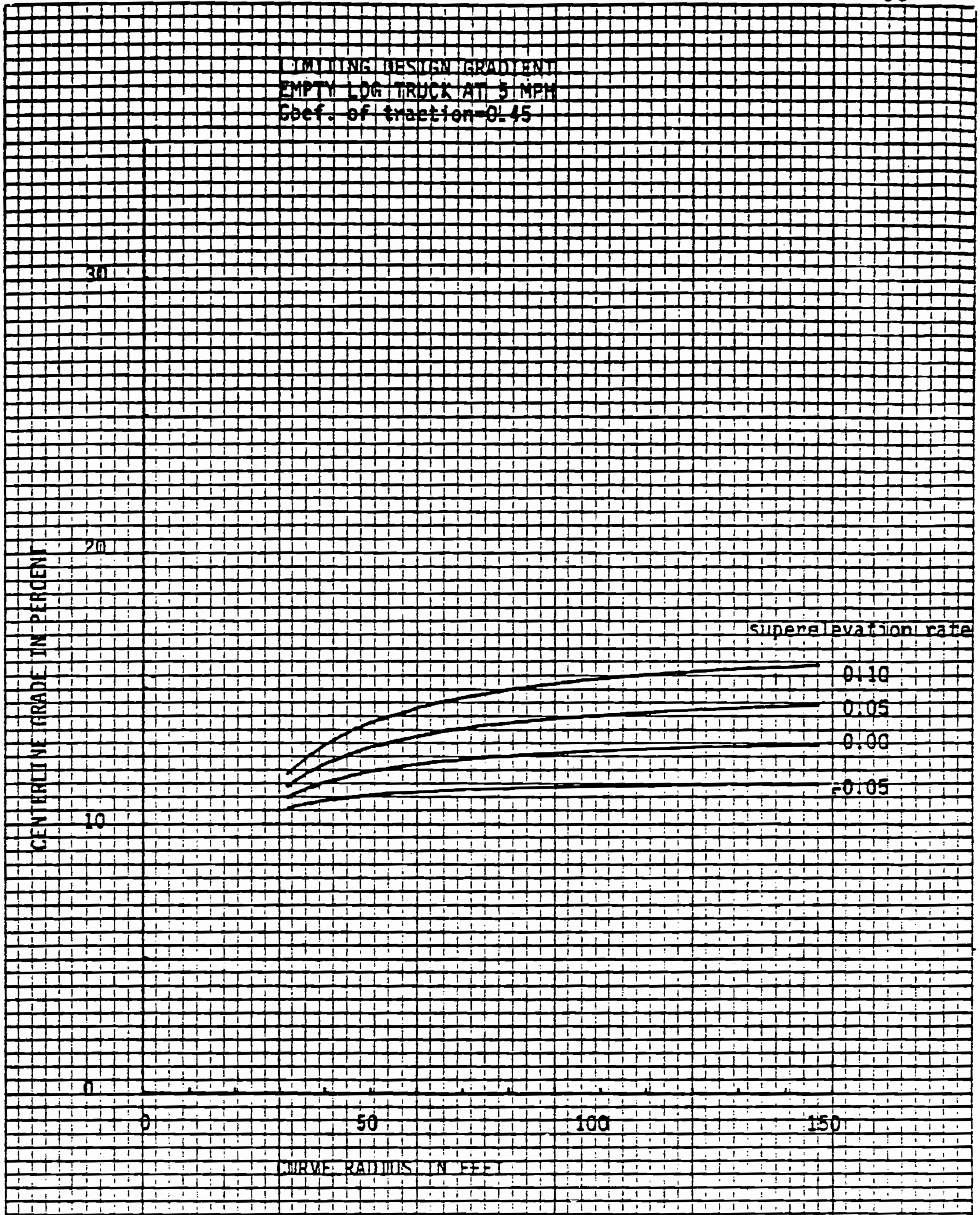
TABLE 3. ROAD-TIRE ADHESION COEFFICIENTS.

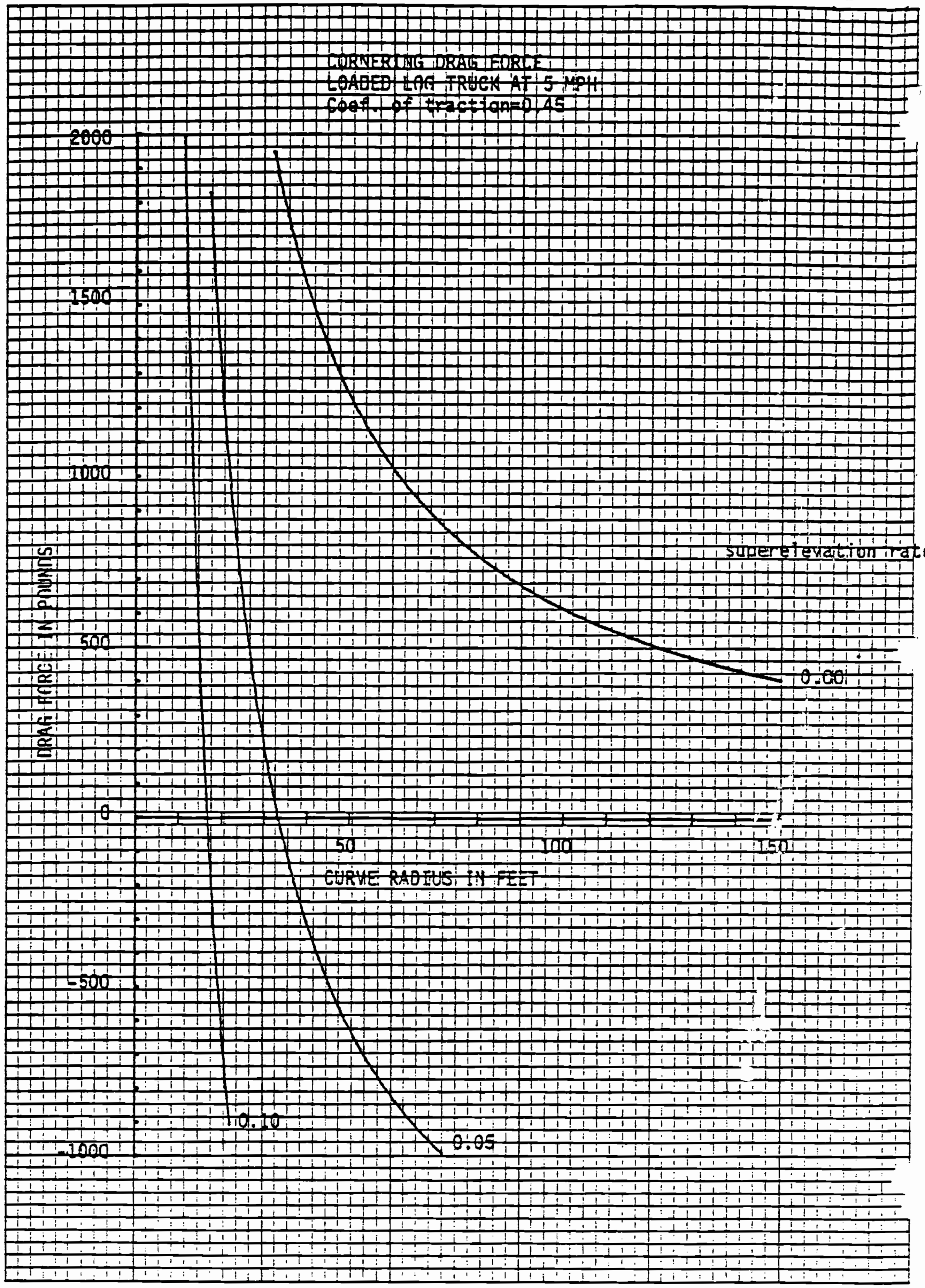
Surface	Condition	Coefficient
Asphalt	dry	.75
Asphalt (traveled)	dry	.55 to .8
Asphalt	wet	.45 to .6
Asphalt (traveled)	wet	.40 to .7
Concrete	dry	.75
Concrete	wet	.70
Cement (traveled)	dry	.60 to .8
Cement (traveled)	wet	.45 to .7
Gravel		.55
Gravel (loose)	dry	.40 to .7
Gravel (loose)	wet	.45 to .75
Gravel (packed, oiled)	dry	.50 to .85
Gravel (packed, oiled)	wet	.40 to .80
Rock (crushed)	dry	.55 to .75
Rock (crushed)	wet	.55 to .75
Earth	dry	.65
Earth	wet	.40 to .5
Snow (packed)		.15
Snow (packed)	dry	.30 to .55
Snow (packed)	wet	.30 to .6
Snow (loose)	dry	.10 to .25
Snow (loose)	wet	.30 to .60
Snow (lightly sanded)		.29 to .31
Snow (lightly sanded)	with chains	.34
Ice		.07

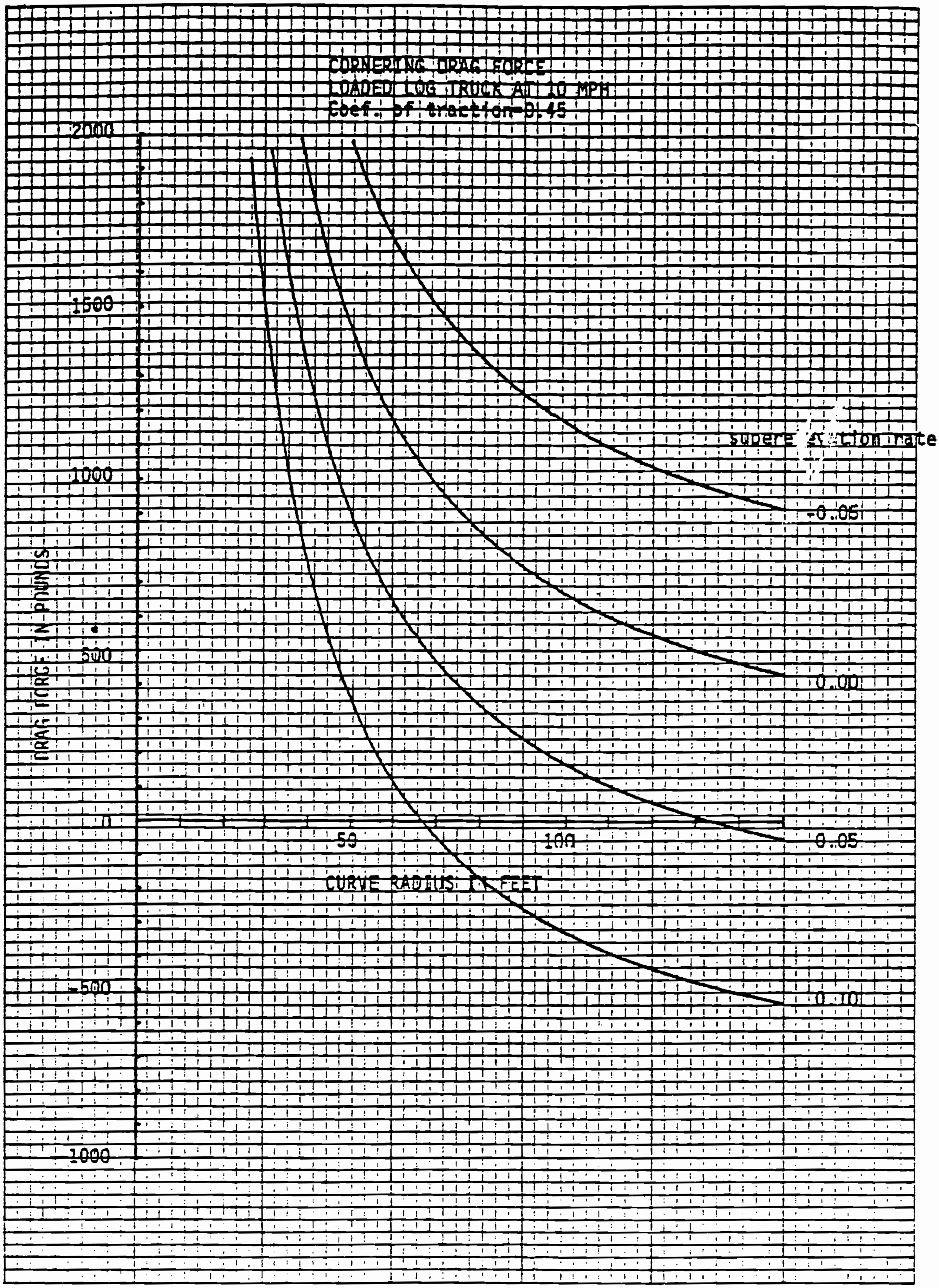
Values are a composite of Taborek (1957) and Western Highway Institute (1976).


LIMITING DESIGN GRADIENT
PIGGYBACK LOG TRUCK AT 5 MPH
Coef. of traction=0.45
superelevation rate
0.10
0.05
0.00
0.05
CENTERLINE GRADE IN PERCENT
30
20
10
0
0
50
100
150
CURVE RADIUS IN FEET

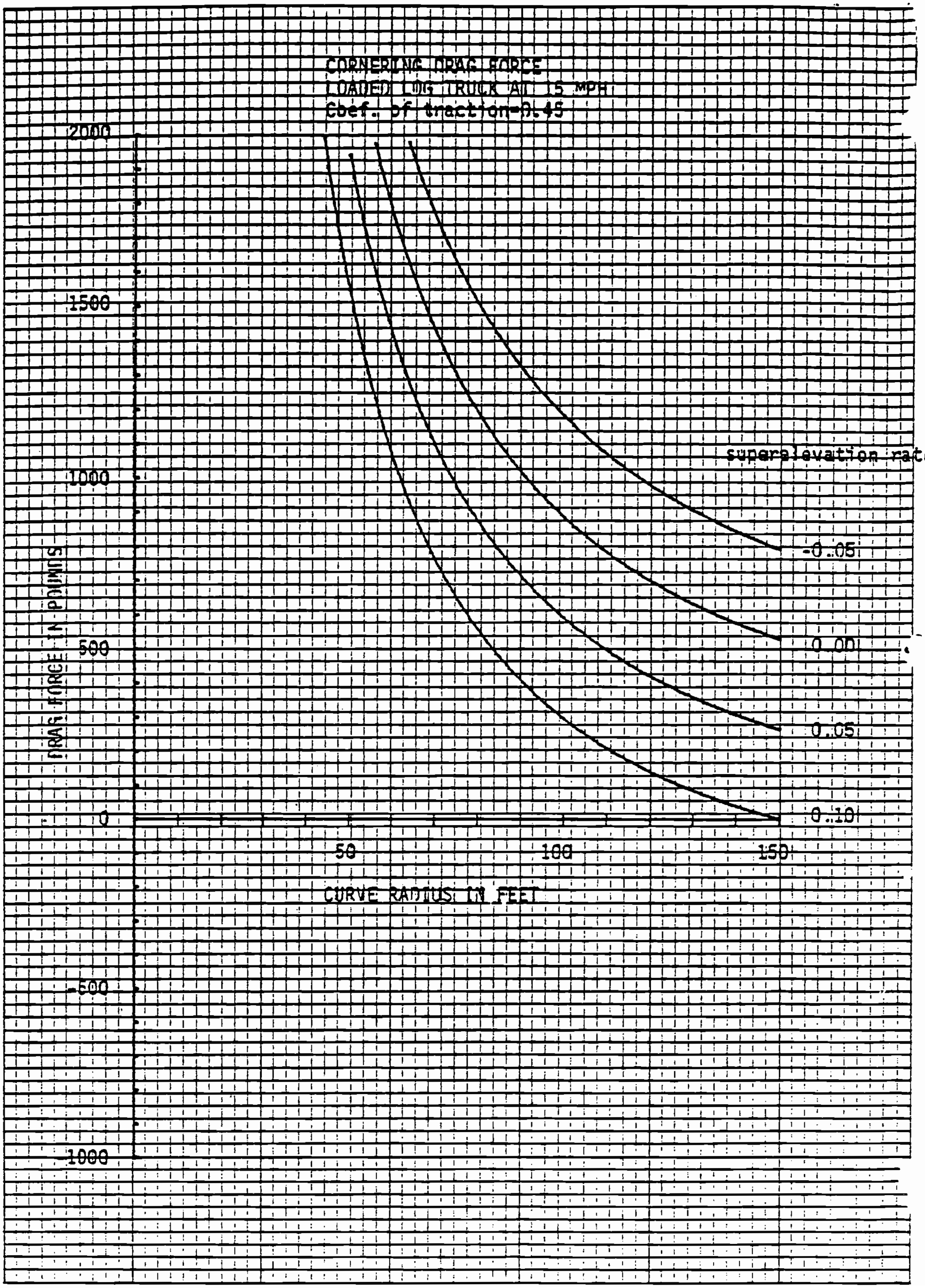

LIMITING DESIGN GRADIENT
LOADED LOG TRUCK AT 5 MPH
Coef. of traction=0.45
superelevation rate
0.10
0.05
0.00
-0.05
CENTERLINE GRADE IN PERCENT
30
20
10
0
0
50
100
150
CURVE RADIUS IN FEET

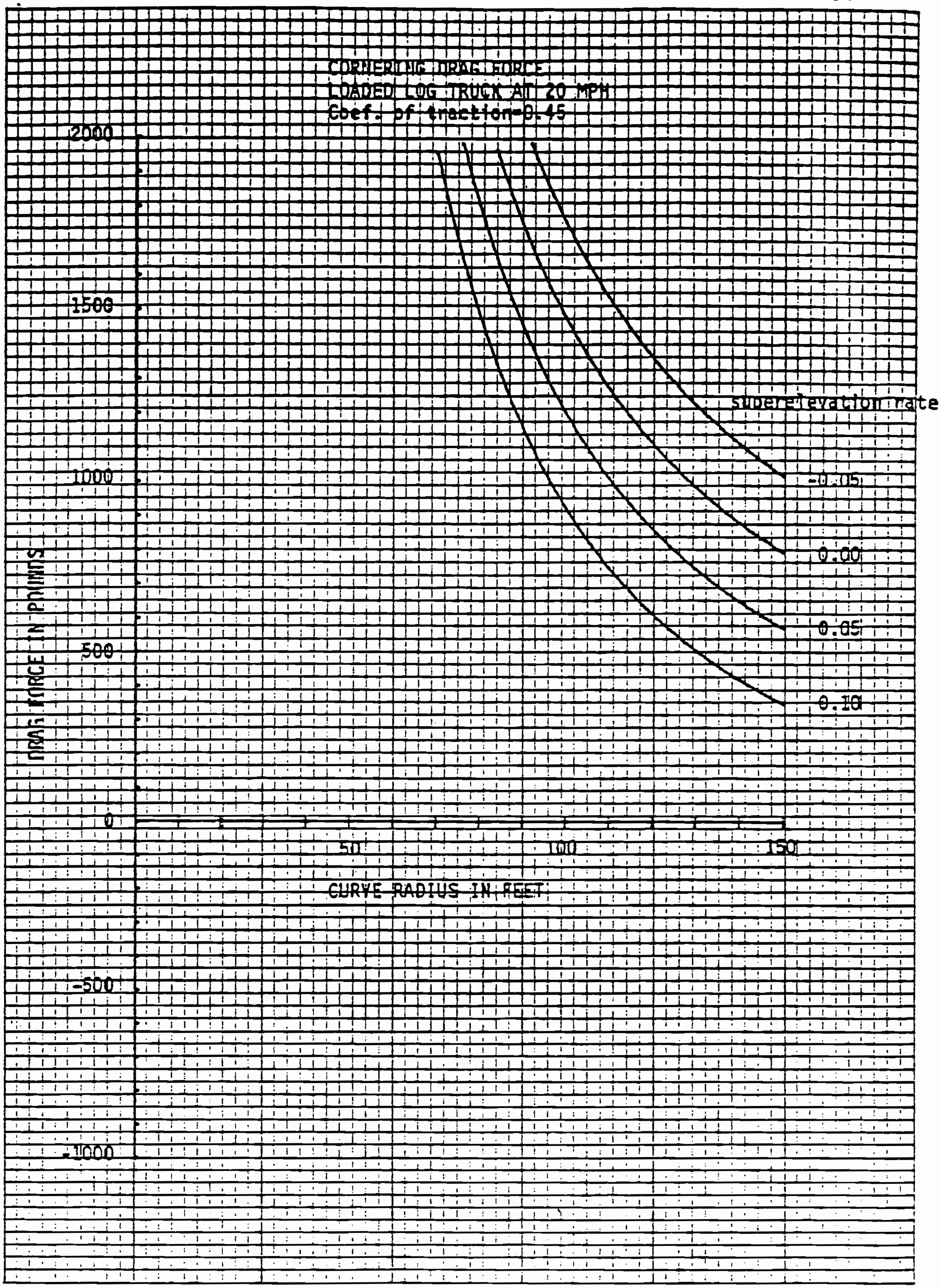

LIMITING DESIGN GRADIENT
EMPTY LOG TRUCK AT 5 MPH
Coef. of traction=0.45
CENTERLINE GRADE IN PERCENT
30
20
10
0
superelevation rate
0.10
0.05
0.00
-0.05
0
50
100
150
CURVE RADIUS IN FEET

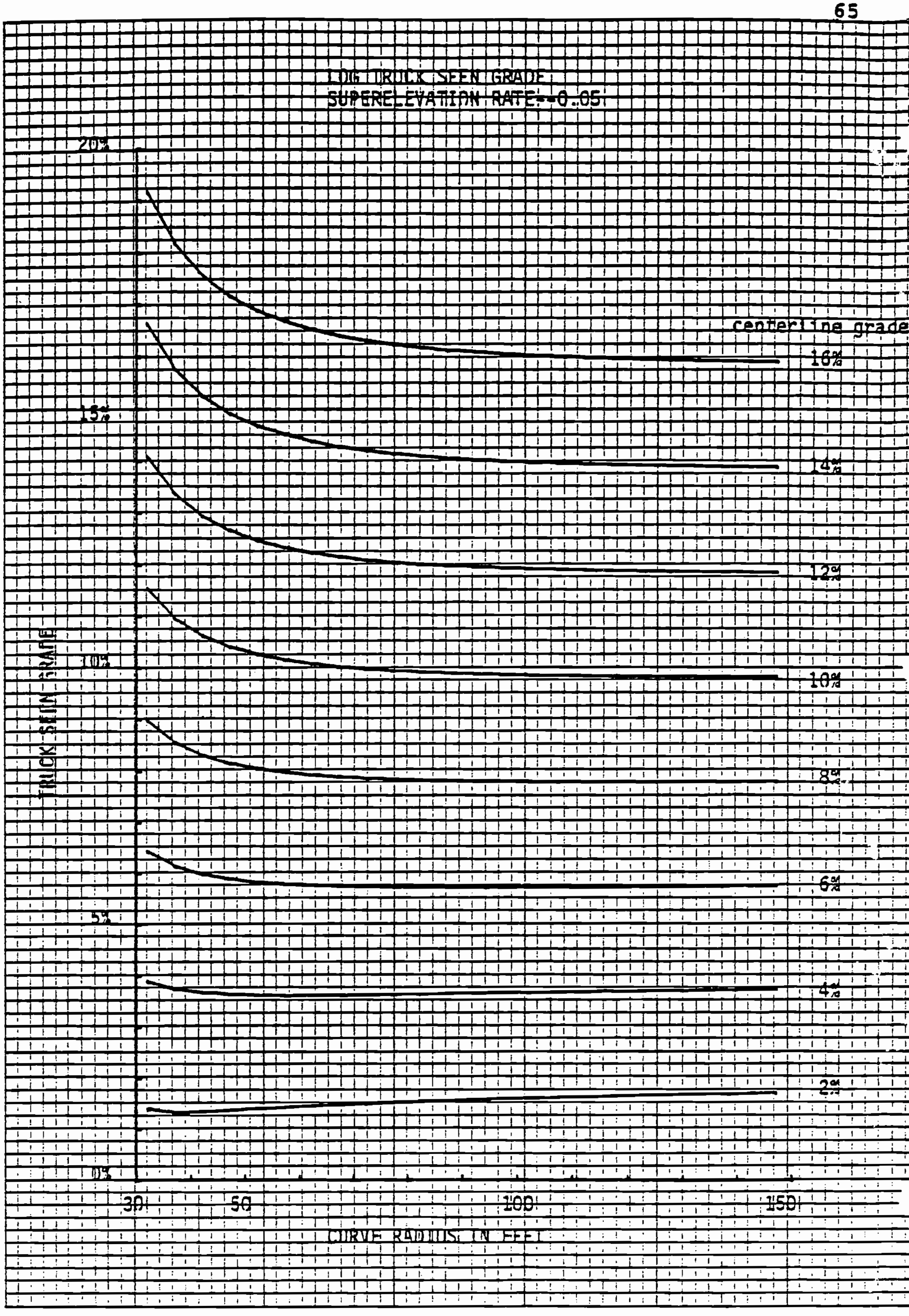

CORNERING DRAG FORCE
LOADED LOG TRUCK AT 5 MPH
Coef. of traction=0.45
2000
1500
1000
500
0
-500
-1000
DRAG FORCE IN POUNDS
50
100
150
CURVE RADIUS IN FEET
superelevation rate
0.00
0.10
0.05

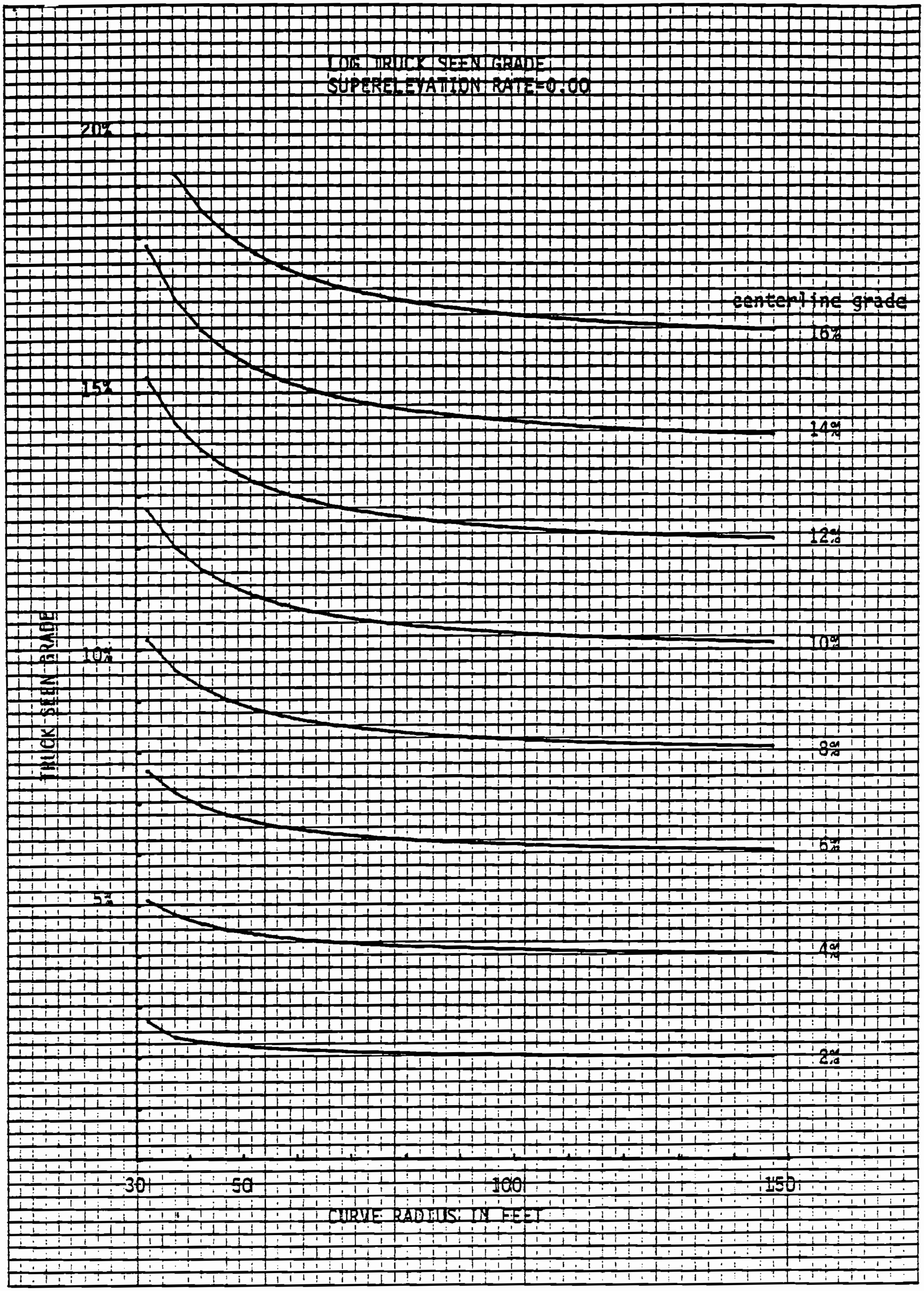


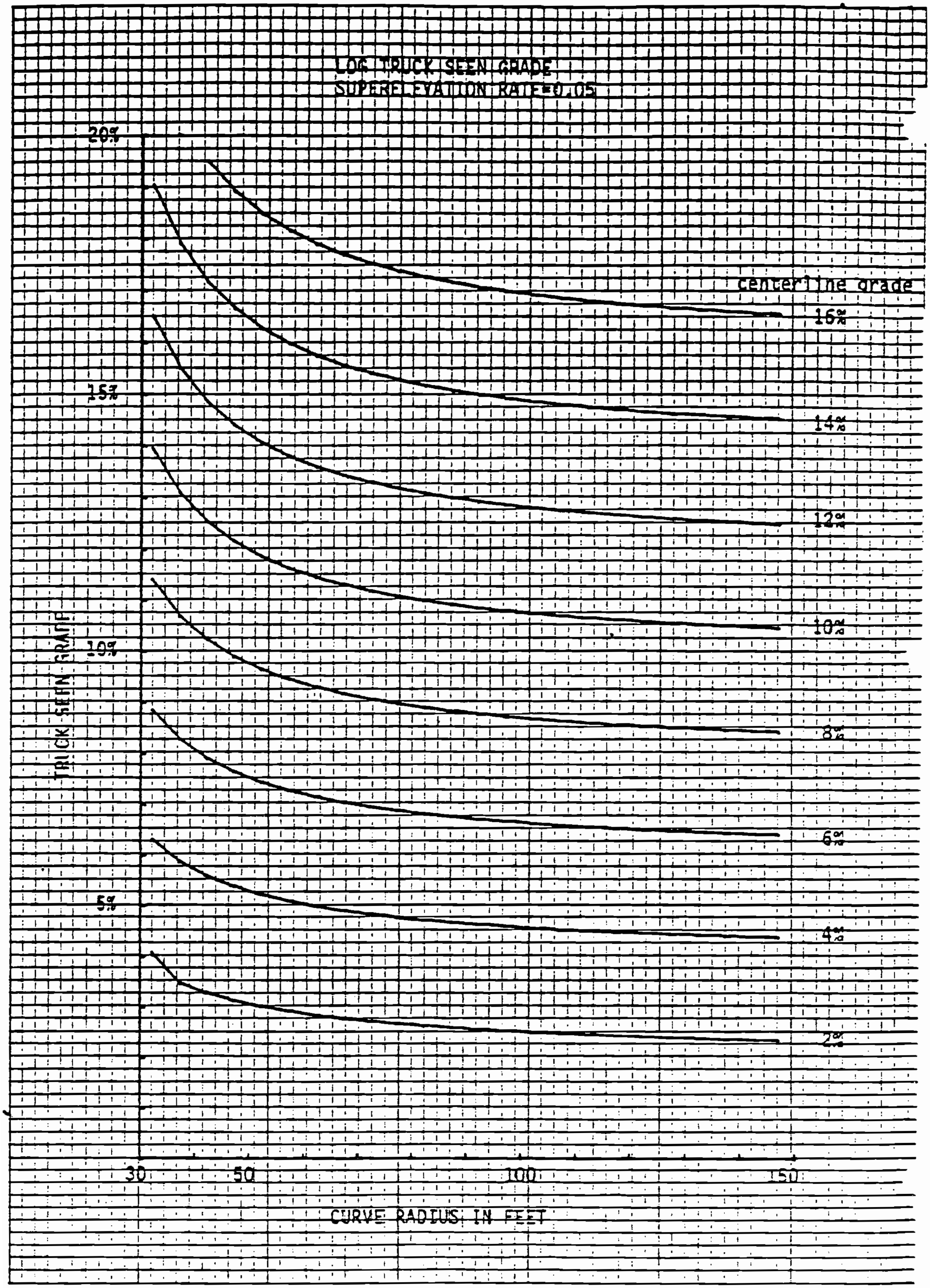
CORNERING DRAG FORCE
LOADED LOG TRUCK AT 10 MPH
Coef. of traction=0.45
DRAG FORCE IN POUNDS
2000
1500
1000
500
0
-500
-1000
50
100
CURVE RADIUS IN FEET
-0.05
0.00
0.05
0.10

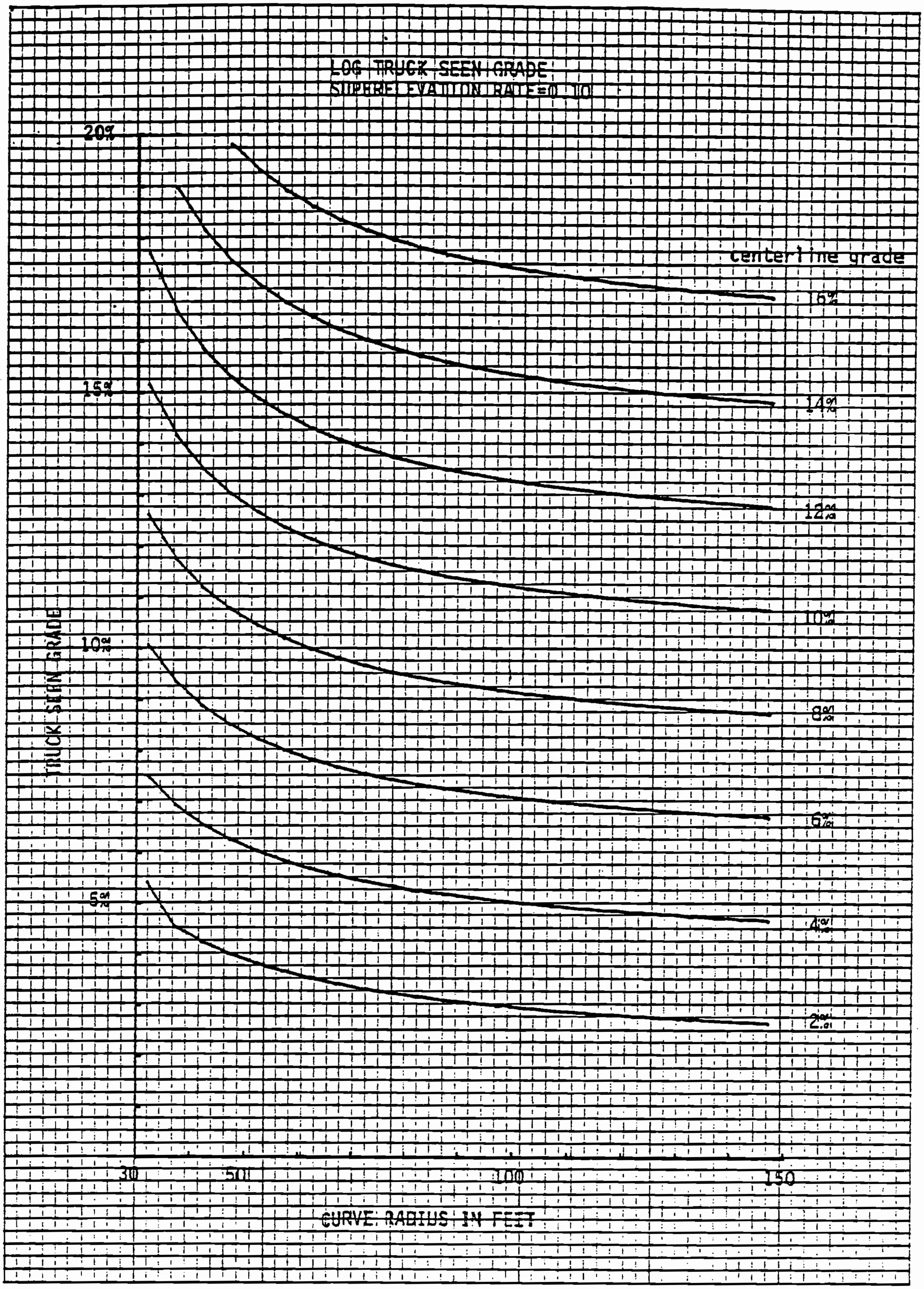

CORNERING DRAG FORCE
LOADED LOG TRUCK AT 15 MPH
Coef. of traction=0.45
DRAG FORCE IN POUNDS
2000
1500
1000
500
0
-500
-1000
50
100
150
CURVE RADIUS IN FEET
superelevation rat
-0.05
0.00
0.05
0.10


CORNERING DRAG FORCE
LOADED LOG TRUCK AT 20 MPH
Coef. of traction=0.45
2000
1500
1000
500
0
-500
-1000
DRAG FORCE IN POUNDS
50
100
150
CURVE RADIUS IN FEET
superelevation rate
-0.05
0.00
0.05
0.10


LOG TRUCK SEEN GRADE
SUPERELEVATION RATE=0.05
20%
15%
10%
5%
0%
TRUCK SEEN GRADE
centerline grade
16%
14%
12%
10%
8%
6%
4%
2%
30
50
100
150
CURVE RADIUS IN FEET


LOG TRUCK SEEN GRADE
SUPERELEVATION RATE=0.00
20%
15%
10%
5%
TRUCK SEEN GRADE
centerline grade
16%
14%
12%
10%
8%
6%
4%
2%
30
50
100
150
CURVE RADIUS IN FEET


LOG TRUCK SEEN GRADE
SUPERELEVATION RATE=0.05
20%
15%
10%
5%
TRUCK SEEN GRADE
centerline grade
16%
14%
12%
10%
8%
6%
4%
2%
30
50
100
150
CURVE RADIUS IN FEET


LOG TRUCK SEEN GRADE
SUPERELEVATION RATE=0.10
20%
15%
10%
5%
TRUCK SEEN GRADE
centerline grade
16%
14%
12%
10%
8%
6%
4%
2%
30
50
100
150
CURVE RADIUS IN FEET