DDS 130-3

STRUCTURAL DESIGN AND ANALYSIS OF

WHEELED AND TRACKED VEHICLE DECKS

DEPARTMENT OF THE NAVY
NAVAL SEA SYSTEMS COMMAND
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PART I: INTRODUCTION

130-3-a. References

(a) DDS 130-2, Structural Design of Aircraft Handling Decks.
(b) DDS 100-4, Strength of Structural Members.
(c) MIL-STD-1399, Interface Standard for Shipboard Systems, Section 301, Ship Motion and Attitude.
(d) Design of Deck Structures Under Wheel Loads, Jackson and Frieze, RINA, 1980.
(e) "Development of Plating Response Methods for Wheel and Track Loads", DTNSRDC Report SD-86-173-18, October 1985.
(f) Individual Ships Specifications, General Specifications for ships of the U.S. Navy or General Overhaul Specifications for Surface Ships, as appropriate.
(g) MIL-HDBK-264 (SH), Properties of Steel Shapes, and Plate-Beam Combinations Used in Shipbuilding.
(h) Structural Design Manual for Surface Ships of the U.S. Navy.
(i) NAVAIR Technical Manual 17-1-537, Aircraft Securing and Handling.
(j) Development of Technical Approach to DDS 130-3. DTNSRDC Report M83, Sept. 1983
(k) Application of Plastic Analysis to U.S. Coast Guard Icebreaker Shell Plating Trans., SNAME 1981.
(l) Design Manual for Orthotropic Steel Plate Deck Bridges, A.I.S.C. 1963.
(m) Structural Design and Analysis of Wheeled and Tracked Vehicle Deck. Review and Development of Methodology and Criteria, NAVSEA 55Y February 1986.

130-3-b. Purpose and Scope

This Design Data Sheet provides uniform standards and simplified methods for the analysis of vehicle handling decks structures. The method incorporated in this Design Data Sheet is to be used to analyze plate deck structures with thin plate, (less than one inch thick), and flexible stiffener supports for longitudinally or transversely framed decks.

This Design Data Sheet deals with only those loads resulting from wheeled and tracked vehicle. Other local loadings and the requirements for hull girder bending, as applicable, should be addressed separately in accordance with References (f) and (h).

The method presented herein is based primarily on Reference (j) and, in addition, References (e), (l) and (m). This version takes advantage of the later and more refined techniques of structural analysis, employing accepted principles of structural mechanics with empirical constants determined from static and dynamic tests. Other documents used in the development of this Design Data sheet are listed in Appendix I.

130-3-c. Symbols, Abbreviations and Definitions

η_{xs} = storm sea ship motion factor, longitudinal to ship
η_{ys} = storm sea ship motion factor, transverse to ship
η_{zs} = storm sea ship motion factor, vertical to ship

X = longitudinal distance from specified center of motion (ft)
Y = transverse distance from specified center of motion (ft)
Z = vertical distance above specified center of motion (tens ft)
η_{xm} = moderate sea ship motion factor, longitudinal to ship
η_{ym} = moderate sea ship motion factor, transverse to ship
η_{zm} = moderate sea ship motion factor, vertical to ship
F_x = ship motion load, longitudinal to ship
F_y = ship motion load, transverse to ship
F_z = ship motion load, vertical to ship
W_p = parked weight of vehicle
W_m = maximum weight of vehicle
W_l = light weight of vehicle
F_w = wind load
a_s = sail area of vehicle
F_f = tire friction force (kips)
F_d = ship motion force, vertical to vehicle (kips)
F_l = ship motion force, longitudinal to vehicle (kips)
F_t = ship motion force, trasverse to vehicle (kips)
X_g = longitudinal distance from main gear to CG of aircraft
Z_g = height of aircraft CG above deck
CP = center of pressure of sail area
CG = center of gravity of vehicle
Z_p = height of CP above deck
Z_t = height of tiedown above deck
Y_t = transverse distance from centerline of vehicle tiedown
T = tiedown force
M = overturning moment (in-k)
Ω = tiedown angle to deck (deg)
R_m = nominal load to left or right side of vehicle
r' = nominal distributed tread load for tracked vehicle
r_f = front distributed load for tracked vehicle in storm sea conditions
r_r = rear distributed load for tracked vehicle in storm sea conditions
R_f = front wheel load for wheeled vehicle in storm sea conditions
R_r = rear wheel load for wheeled vehicle in storm sea conditions
R_f' = nominal front wheel load
R_r' = nominal rear wheel load
R_v' = R_1 or R_2
R_1 = load to left side of vehicle
R_2 = load to right side of vehicle
s = frame spacing (ft)
S = wheel spacing lateral (ft)
P_b = bottoming load of tire (kips)
L_s = stiffener span length (in)
L_b = beam span length
b_e = effective breadth of plating
t = plating thickness
d = depth of stiffener
b_f = breadth of flange

t_w = web thickness
t_f = flange thickness
ψ = dual pitch equivalent load factor, plating
A = patch length of load
B = patch width of load on
a = plate panel length
b = plate panel width
H, J, K_1, K_2, K_3 = terms used in vehicle load balance calculations
E = modulus of elasticity (ksi)
F_y = yield strength of material (ksi)
F_b = allowable working strength of material (ksi)
C_1 = non-dimensionalized bending moment coefficient of plating
C_o = deck function coefficient
P = patch loads (kips)
σ_p = plate allowable bending stress (ksi)
ϕ_1 = patch width load distribution factor, stiffener
ϕ_2 = plating load distribution factor, stiffener
ϕ_4 = beam loading coefficient
I_s = moment of inertia of plate-stiffener combinations (in^4)
I_b = moment of intertia of plate-beam combination (in^4)
γ_{ps} = relative rigidity coefficient of plate-stiffener
γ_{sb} = relative rigidity coefficient of stiffener-beam
M_c = stiffener bending moment correction for elastic supports (in-k)
M_o = stiffener bending moment over rigid supports due to load (in-k)
M_d = stiffener bending moment due to dead load of structure (in-k)
w_d = dead load of plating and stiffener (k/in)
w_s = dead load of stiffener (lb/ft)
w_p = dead load of plating (lb/ft^2)
M_s = total bending moement in stiffener (in-k)
f_{sb} = calculated bending stress in stiffener (ksi)
SM_{min} = minimum section modules of stiffener-plate combination (in^3)
V_o = stiffener shear due to load (kips)
σ_s = stiffener allowable bending stress (ksi)
$\sigma_{primary}$ = hull girder design primary stress (ksi)
V_d = stiffener shear due to dead load (kips)
V_s = total shear in stiffener (kips)
f_{sv} = calculated stiffener shear stress (ksi)
P_o = operational tire pressure (psi)
A_s = shear area of stiffener ($d \times t_w$) (in^2)
e_s = effective span length factor, stiffener
f_p = calculated bending stress in plating (ksi)
e_b = effective span length factor, beam
R_o = unit gear load, beam (k/in)
B_o = unit gear load width, beam (in)
F_w = wind load acting on vehicle (kips)
t_{reqd} = required plating thickness (in)
SM_{reqd} = required minimum section modulus of stiffener-plate combination (in^3)
AS_{reqd} = required shear area of stiffener (in^2)
β = influence line ordnate

130-3-d. General

Parts II and III provide a detailed description of the assumptions, rationale, and methods for determining the loading and load distribution.

Part IV provides a detailed description of the assumptions, rationale, and methods for analyzing the structural response.

Part V provides the design criteria to be used for determining the adequacy of the structure.

Appendices A and B provide the necessary data on existing and notional vehicles.

Appendices C, D, and E provide summaries of the analysis methods for easy reference.

Appendix F provides standard work sheets for documenting the analysis.

Appendix G provides a complete example.

Appendix H provides selected measurement units and conversion factors.

Appendix I provides a list of other documents that are applicable to the design and analysis of wheeled and tracked vehicle handling decks.

PART II: LOADING

130-3-e. General

Vehicle handling decks are exposed to two types of loads in addition to but separate from the standard design loads for all decks. These two types of loads are the parking loads imposed by the vehicle and the inertial loads produced by ship's motion accelerations. The ship specifications specify two conditions, storm seas and moderate seas, and provide equations to determine the ship motion factors for that ship for any point on the ship. Although ship motion factors and their corresponding sea states and chance of exceedance are dependent on ship size and the particular seaway, the following approximations that apply to most Navy ships and operational seaways, are offered as background information. The storm sea factors relate to a sea state 7, which for the average service of a ship, correlates to a 0.05 percent chance of exceedance. The moderate sea factors relate to a sea state 5, which for the average service of a ship, correlates to a 30 percent chance of exceedance.

130-3-f. Vehicle Loads

The nominal vehicle weight, w_m, represents the gross vehicle weight and includes full payload and personnel weight. This conservative weight estimate is necessary to account of initial tie-down forces which will secure the vehicle in a fully loaded state regardless of actual initial vehicle loads.

Large tie-down forces will be exerted on the deck both from initial tie-down of vehicles and from ship motion forces acting on the vehicle. Within this study it is assumed that tie-down cleats will always be located at frames and other structural hard spots on the deck where tie-down induced stresses will be minimized. For frame analysis, tie-down forces oppose normal vehicle loads and are neglected as a conservatism.

Ship motion forces must be calculated for each vehicle and are a function of nominal vehicle weight, type of ship, position of vehicle on ship and sea state. The nominal vehicle weight and ship motion forces are additive for vehicle deck studies. The cyclic nature of ship motion forces requires consideration of a full range of possible load combinations to determine the most critical combination. Each of the three ship motion forces will have three states requiring consideration. These include forces in two opposite directions and a neutral or zero state.

A summary of possible load combinations is included in the following table:

Vehicle Alignments with Respect to Ship					
Longitudinal			Transverse		
Vertical Load	Lengthwise Load	Lateral Load	Vertical Load	Lengthwise Load	Lateral Load
F_d	F_l	F_t*	F_d	F_l	F_t*
$\overline{F_d}$	$-F_l$		F_d	$-F_l$	
$F_d = W_m \times \eta_z$	$F_l = W_m \times \eta_x$	$F_t = W_m \times \eta_y$	$F_d = W_m \times \eta_z$	$F_l = W_m \times \eta_y$	$F_t = W_m \times \eta_x$

*-F_t load is unnecessary because of lateral symmetry

For F_d and $\overline{F_d}$ see equation (1).

This load matrix results in many different load combinations. The following four combinations are most likely to provide the critical load combination:

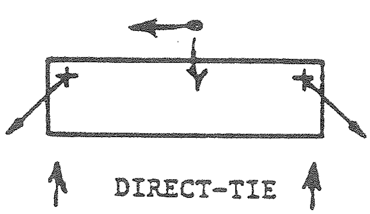
Vehicle Alignment with Respect to Ship	Vertical Load	Lengthwise Load	Lateral Load
Longitudinal	F_d	F_l	F_t
Longitudinal	$\overline{F_d}$	$-F_l$	F_t
Transverse	F_d	F_l	F_t
Transverse	$\overline{F_d}$	$-F_l$	F_t

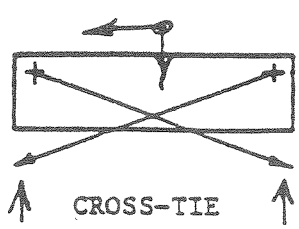
These vehicle loads are assumed to act through the vehicle center of gravity and are resisted by a combination of wheel normal reaction forces, wheel-friction forces and vehicle tie-down forces. For deck design/analysis, the magnitudes of the wheel normal reaction loads are required. These reaction forces are dependent on the distribution of resisting forces from vehicle tie-downs and wheel friction. Because the distribution between tie-downs and wheel friction is statically indeterminate, a simplifying assumption is made to solve the load distribution.

The tie-downs are assumed to act such that the wheel reaction forces at time of initial tie-down are zero. If ship motions act to reduce a wheel load below this miminum level, the tie-down force will act to hold the wheel at the initially position. Consequently, each wheel can only exist in one of two possible states. The first state has a constant reaction force with a variable tie-down force. The second state has a reaction force varying above zero. Both states for wheel friction and are statically determinate.

Tie-Down Considerations

The following two general tie-down arrangements are possible:





Both are typical tie-down arrangements; however, for purposes of analysis the direct-tie leads to a simpler solution because it requires only one unknown tie-down force to maintain the assumed static balance, whereas the cross-tie requires two differing unknow tie-down forces. In order to simplify the analysis procedure, the direct-tie is used for both lengthwise and lateral force balances.

Determination of Vehicle Wheel/Track Loads

Two general vehicle load conditions are significant for vehicle deck design. The first is the nominal vehicle weight which accounts for a motionless or moderate sea condition, fully loaded vehicle and the second is the ship motion loading which accounts for the nominally weighted vehicle tied to the deck in a storm-sea condition subjected to vertical, longitudinal and transverse ship motions forces. These ship motion forces are a function of vehicle type, ship geometry, location of vehicle on ship, and sea state.

Deck design normally will be based on the critical ship motion loading for general vehicle types for the deck. This is based on the assumption that nominal vehicle weight will be constant while the vehicle is on the deck. However, for fork lift vehicles, nominal weight at calm sea (no ship motion forces) and nominal weight at storm sea conditions will vary by the rated capacity of the vehicle based on the assumption that a fork lift will never be operated during storm seas. Hence, both loadings must be considered for design purpose for fork lift vehicles.

The general procedure for obtaining wheel and track loadings is outlined in following discussion and illustrations.

(1) The nominal wheel loading is first calculated based on a simple static balance of the vehicle at rest with known positioning of wheel center and vehicle center of gravity. These wheel loads are established as the minimum nominal wheel loads. If in certain conditions, the load of one wheel is lower than this nominal load, the load of other wheel will be greater.

(2) Ship motion forces are then calculated for the vehicle using storm sea motion factors η_x, η_y, η_z.

(3) Wheel or track loadings are calculated by the method of static force balance accounting for loads, normal wheel reaction forces, friction forces and tie-down forces. Generally, all wheeled vehicle will be described with four wheel loads which equal the reaction forces. Dual or tandem rear wheels are assumed to uniformly distribute the total wheel load amongst all tires on that side of the axle.

(4) Tracked vehicle loads are described by two trapezoidal patch loads. For nominal tracked vehicle loading, a uniform patch load is used.

(5) The static balance procedure is accomplished by balancing lateral and lengthwise loadings separately.

(6) The lateral static balance determies the distribution of loads to the right and left sides of the vehicle due to vertical and laterial forces.

The distribution of lateral loads betweem tie-downs and wheel friction is normally indeterminate; however, by assuming that tie-downs are ineffective unless the side load reduces below zero, the static balance may be calculated by neglecting tie-downs. If the static balance results in a side load less than zero, then tie-down forces are necessary and the static balance is calculated by assuming the zero load at one side and solving for load of the other side load.

A lengthwise balance is performed for each of the side loads to determine the wheel load distribution due to the side load and half of the lengthwise ship motion force. As with the lateral load balance, the tie-downs are assumed ineffective unless the wheel load is less than zero wheel load. This results in two sets of statically determinate balances which will provide wheel or track loadings.

This wheel load determination procedure is outlined in the following pages which include equations for solution of the wheel loads due to the various load conditions. In addition, an example wheel load determination is included in Appendix D.

130-3-f.1 Ship Motion Factors

The ship specifications provide storm and moderate sea ship motion equations in a form such that the dimensions from the motion axes of vessel to the center of gravity of the vehicle are substituted into these equations to obtain the motion factors in each of the directions.

η_x = Fore and aft factor
η_y = Athwartships factor
η_z = Vertical factor

If these equations are not available, Reference (C) can be used to obtain the motion factors.

130-3-f.2 Ship Motion Loads

The loads produced on the vehicle acting at the center of gravity are the product of the ship motion factor and the appropriate weight of the aircraft.

F_i = Ship motion force in i direction
η_i = Ship motion factor in i direction
W_j = Weight of vehicle in j condition
$F_i = \eta_i W_j$

130-3-f.3 Wind Loads

The wind load on the vehicle acts at the center of pressure of the sail area and is defined as follows:

Storm Seas
$F_w = 0.030a_s$
Moderate Seas
$F_w = 0.015a_s$

Where a_s is the sail area in square feet of the vehicle.

130-3-f.4 Ship Motion Forces

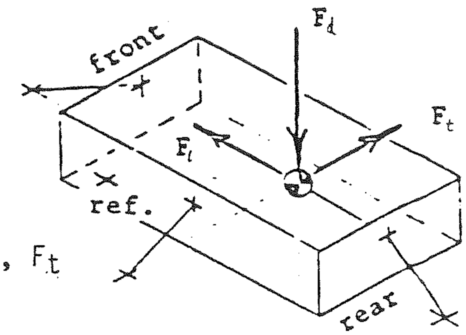
Vehicle Orientation with respect to the ship will also affect the loading imposed on the deck. Therefore, each ship motion load must be oriented to the vehicle for each loading condition.

F_l = Ship motion force longitudinal to vehicle
F_t = Ship motion force transverse to vehicle
F_d = Downward ship motion force
Vehicle oriented athwartships
$F_l = F_y$
$F_t = F_x$
$F_d = F_z$

130-3-f.5 Wheel/Track Load Analysis; Procedure

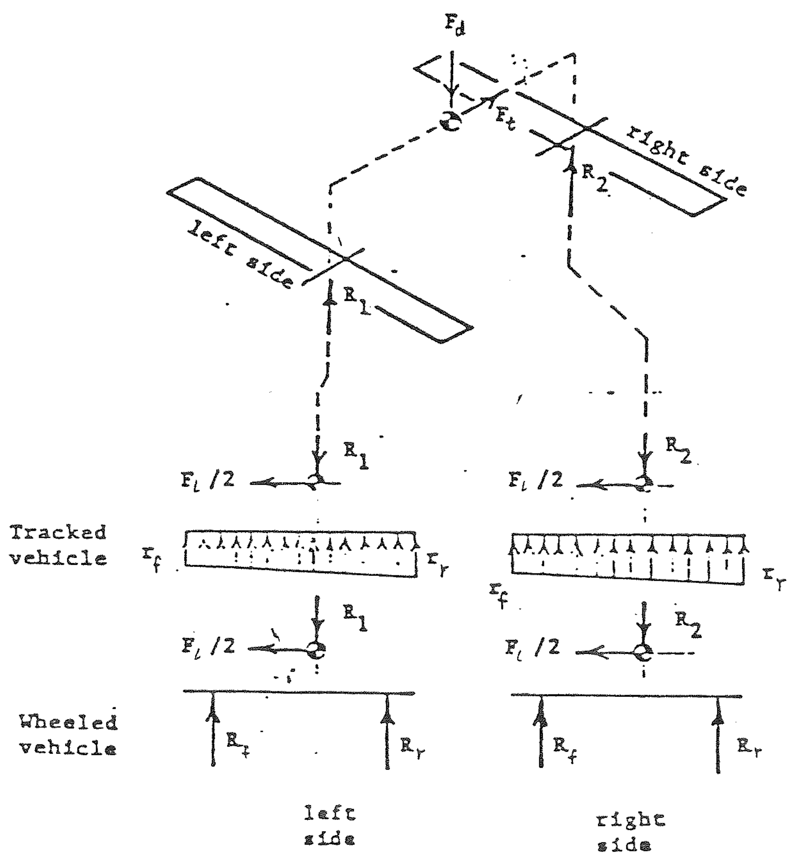
1. Define vehicle particulars:

   a. Vehicle C.G.
   b. Position of Tie-downs
   c. Vehicle Reference Point
   d. Maximum Vehicle Weight, W_m
   e. Vehicle Ship-Motion Forces, F_d, F_l, F_t



2. Determine resultant loads to left and right side of vehicle due to W_m, F_d, and F_t

3. Determine wheel or track loads due to resultant side load R_1 or R_2 and lengthwise ship motion force, F_l



PART III: LOAD DISTRIBUTION

130-3-g. Solution Procedure; Wheeled or Tracked Vehicles

1. Determine maximum vehicle weight, W_m (use of maximum vehicle weight accounts for initial tie down force).

2. Define ship motion factors η_x, η_y, η_z as a function of ship, sea state and vehicle position on ship.

3. Define ship motion forces for vehicle;

   a. If vehicle aligned longitudinally,

   $F_d = \eta_z \times W_m$; $F_l = \eta_x \times W_m$; $F_t = \eta_y \times W_m$

   b. If vehicle aligned transversely,

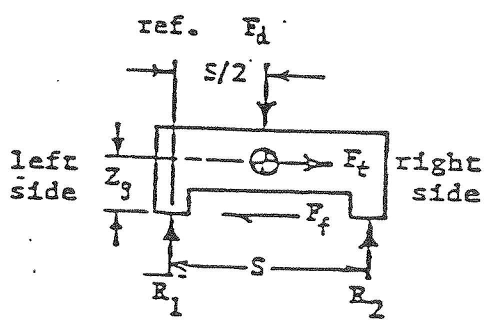
   $F_d = \eta_z \times W_m$; $F_l = \eta_y \times W_m$; $F_t = \eta_x \times W_m$

4. Define:

   a. Location of tie downs on vehicle.
   b. Reference point for wheeled (normally center of left-front tire patch load) or tracked vehicle (normally front-center of left-hand tread patch load).
   c. Location of vehicle CG. (X_g, Y_g, Z_g)
   d. Centers of 4 wheel loads for wheeled or center of tread load for tracked vehicle.
   e. Tie-down angle (one angle assumed for all tie-downs).

5. Determine loading to each side of vehicle by performing static balance of vertical and lateral loads;

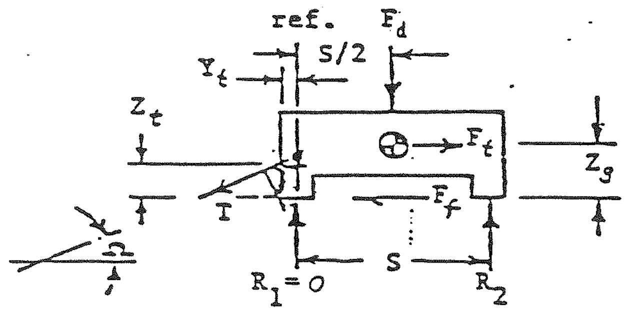
   a. Perform lateral balance assuming only friction force and no tie-down force.



$$R_2 = F_d/2 + F_t \times Z_g/S$$
$$R_1 = F_d - R_2$$

If $R_1 \geq 0$ then R_1 and R_2 valid; however, if $R_1 < 0$ then tie-down forces must be accounted for the balance and R_2 must be determined with following procedure:

b. $R_1 < 0$ in previous analysis; since there no connection between the wheel and the deck $R_1 = 0$



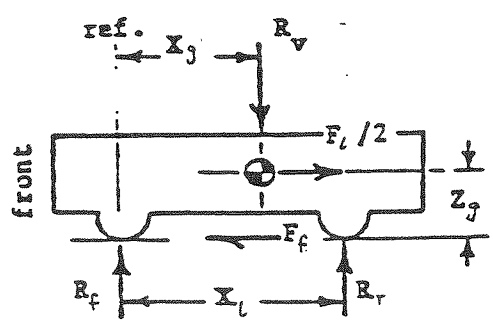
$$
\begin{aligned}
R_1 &= 0 \\
T &= (F_t \times Z_g - F_d \times S/2)/K \\
K &= H + S \sin\Omega, \\
H &= Z_t \times \cos\Omega + Y_t \times \sin\Omega \\
R_2 &= F_d + T \sin\Omega \\
F_f &= F_t - T \cos\Omega
\end{aligned}
$$

6. Proportion one-half of lengthwise motion force to each half of vehicle.
7. Perform lengthwise static balance for each side of vehicle. Procedure is similar to lateral balance; however, 2 cases exist for each vehicle. These 2 cases account for lengthwise force acting toward front of vehicle and toward rear of vehicle.

   a. Perform lengthwise balance assuming only friction force and non tie-down force

Case 1; lengthwise motion force toward rear of vehicle

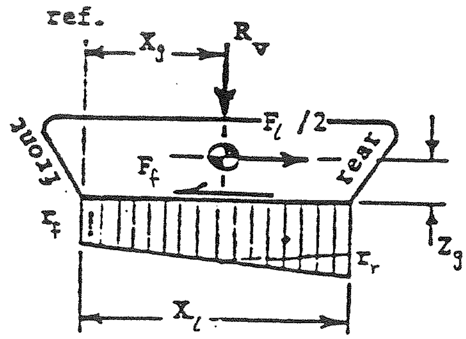
i) Wheeled Vehicle



$$
\begin{aligned}
R_v &= R_1 \text{ (left side)} \\
&= R_2 \text{ (right side)} \\
R_r &= (R_v \times X_g + F_l/2 \times Z_g)/X_l \\
R_f &= R_v - R_r
\end{aligned}
$$

If $R_f \geq 0$ then R_f and R_r valid; otherwise tie-down forces must be accounted for with procedure in 7b.

ii) Tracked Vehicle



$R_v = R_1$ (left side)
$= R_2$ (right side)

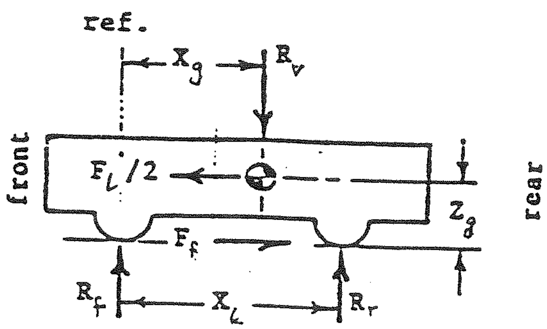
$r_r = 6/X_l^2 \times [R_v \times (X_g - X_l/3) + F_l/2 \times Z_g]$

$r_f = 2 \times R_v/X_l - r_r$.

If $r_f \geq 0$ then r_f and r_r valid, otherwise tie-down forces must be accounted for with procedure in 7b.

Case 2; Lengthwise motion force toward front of vehicle

i) Wheeled Vehicle



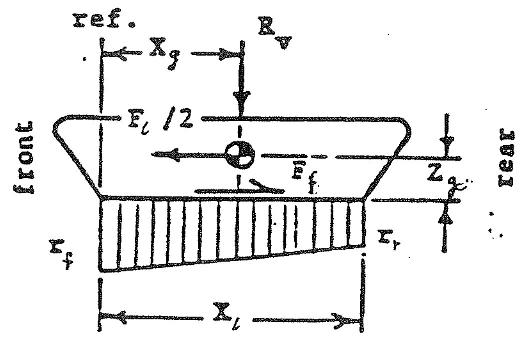
$R_v = R_1$ (left side)
$= R_2$ (right side)

$R_r = (R_v \times X_g - F_l/2 \times Z_g)/X_l$

$R_f = R_v - R_r$

If $R_r \geq 0$ then R_f and R_r valid, otherwise, tie-down forces must be accounted for with procedure in 7b.

ii) Tracked Vehicle



$$R_v = R_1 \text{ (left side)}$$
$$= R_2 \text{ (right side)}$$
$$r_r = 6/X_l^2 \times [R_v \times (X_g - X_l/3) - F_l/2 \times Z_g]$$
$$r_f = 2 \times R_v/X_l - r_r$$

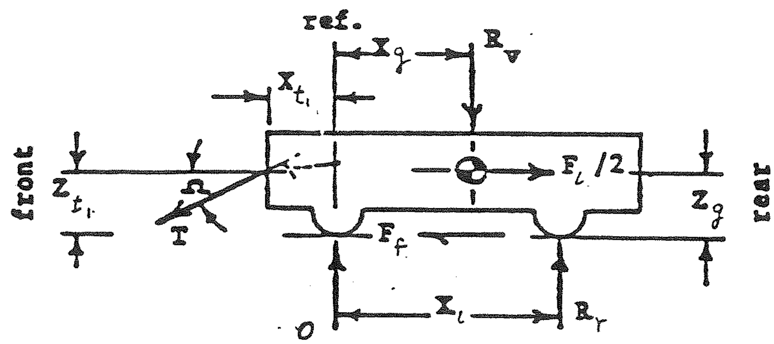
If $r_r \geq 0$ then r_f and r_r valid; otherwise tie-down forces must be accounted for with procedure in 7b.

b. $R_f < 0$, $R_r < 0$, or $r_r < 0$ in 7a; therefore, perform following lengthwise balance where friction and tie-down forces are accounted for with following assumption:

Case	Vehicle Type	Assumption
1	Wheeled	$R_f = 0$
	Tracked	$r_f = 0$
2	Wheeled	$R_r = 0$
	Tracked	$r_r = 0$

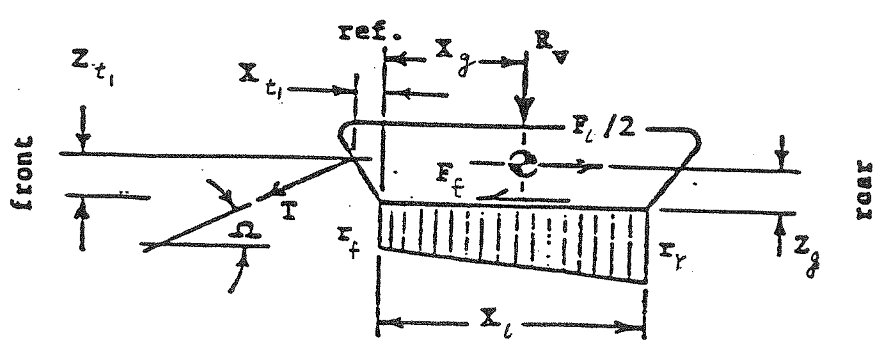
Case 1; lengthwise motion force toward rear of vehicle

i) Wheeled Vehicle:



R_v = R_1 (left side)
= R_2 (right side)
R_f = 0
Where $K = H + X_1 \times \sin\Omega$
$H = Z_{t1} \times \cos\Omega + X_{t1} \times \sin\Omega$
$T = (F_1 \times Z_g/2 - R_v \times (X_1 - X_g))/K$
$R_r = R_v + T \sin\Omega$
$F_f = F_1/2 - T \cos\Omega$

ii) Tracked Vehicle



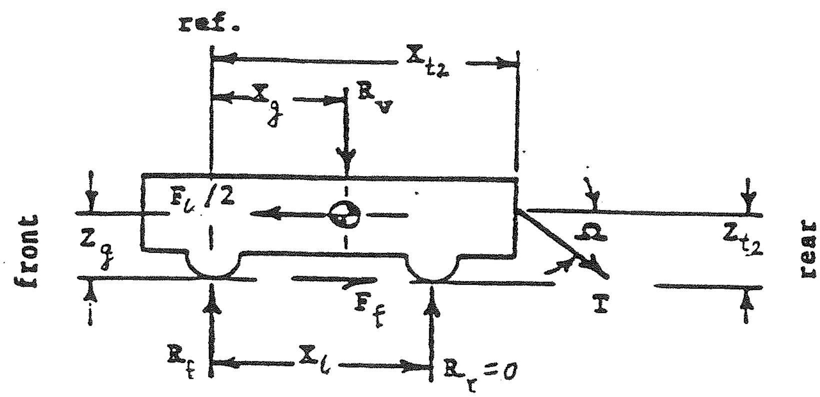
R_v = R_1 (left side)
= R_2 (right side)
$r_f = 0$
$r_r = 2\ (R_v + T)/X_1$
$T = (F_1\ Z_g/2 - R_v\ (2X_1/3 - X_g))/K$
$K = H + 2X_1 \times \sin\Omega\ /3$
$H = Z_{t1} \times \cos\Omega + x_{t1} \times \sin\Omega$

Case 2; lengthwise motion force toward front of vehicle

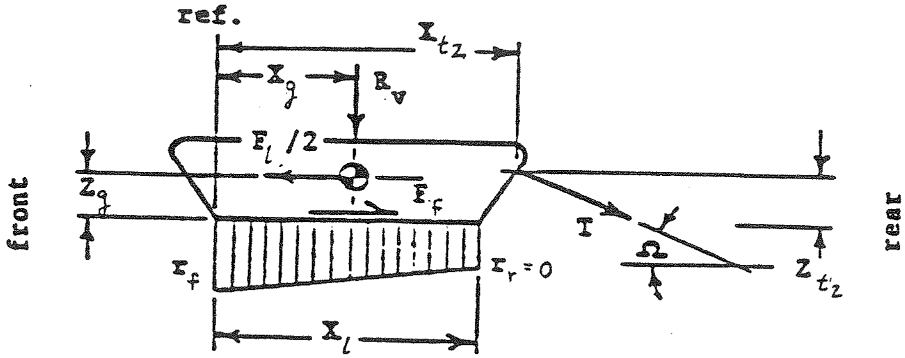
i) <u>Wheeled Vehicle</u>:

$R_v = R_1$ (left side)
$\quad = R_2$ (right side)
$R_r = 0$
$T = (F_1 Z_g/2 - R_v(X_1 - X_g))/H$
$R_f = R_v + T \sin\Omega$
$F_f = F_1/2 - T \cos\Omega$

where, $H = Z_{t2} \times \cos\Omega + X_{t2} \times \sin\Omega$

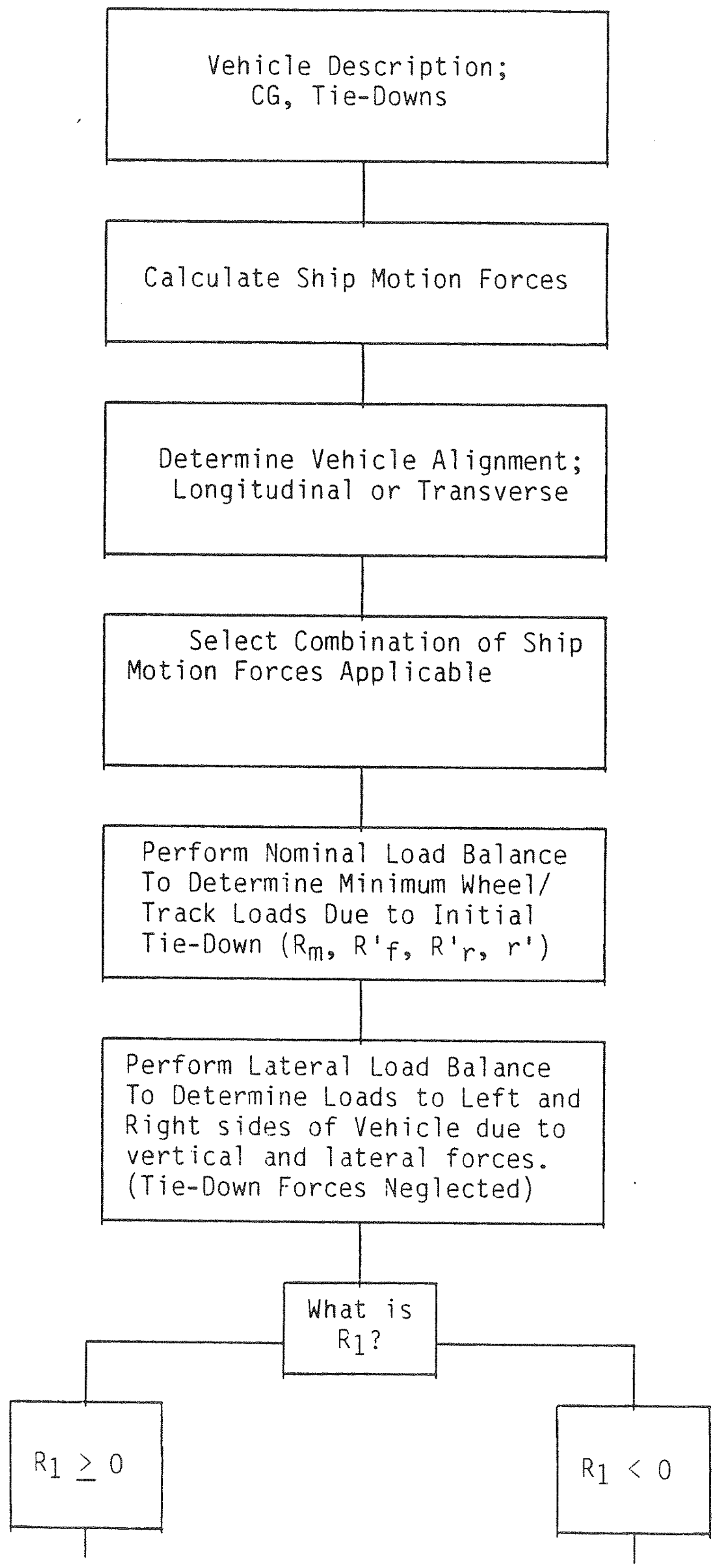


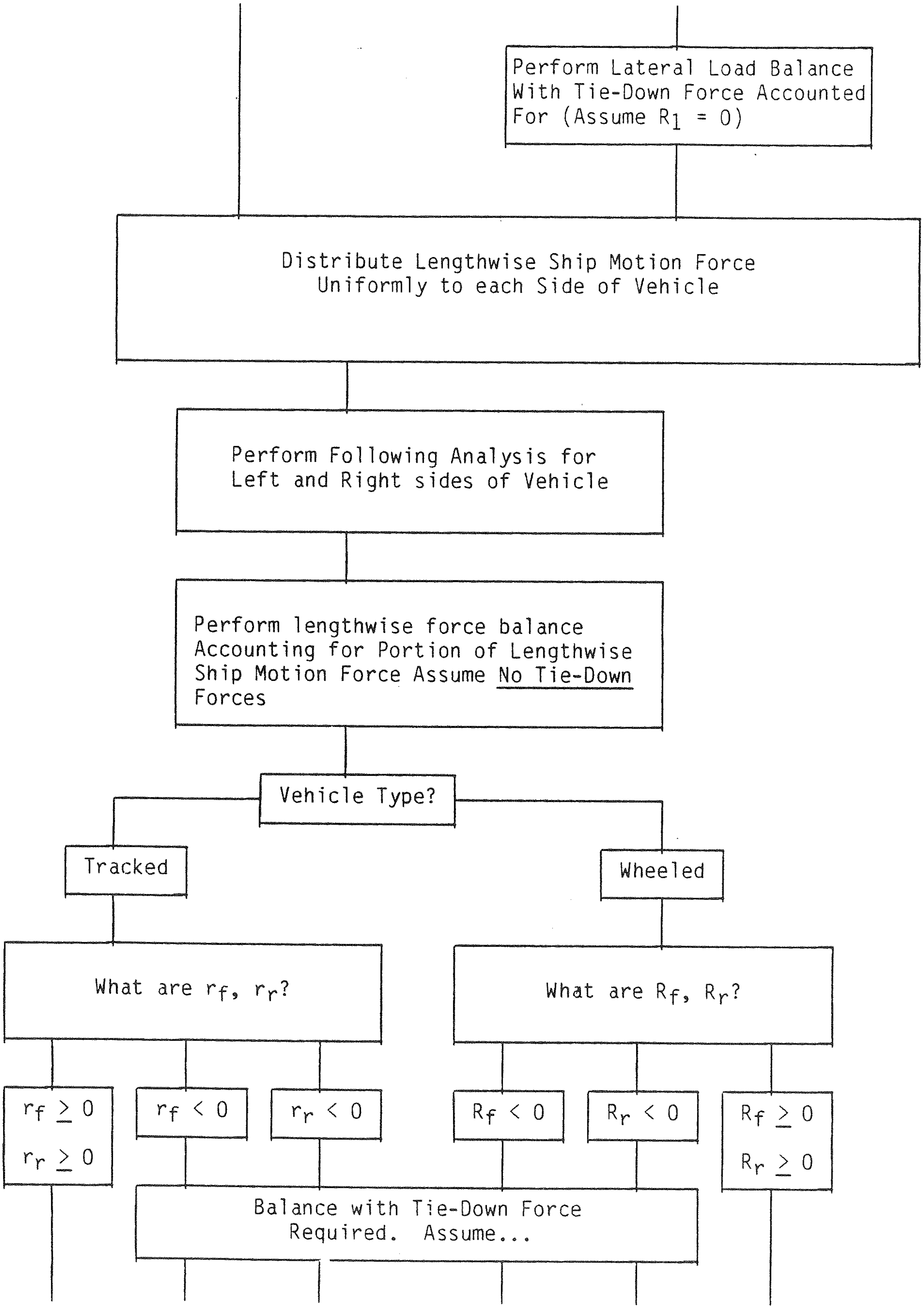
ii) Tracked Vehicle:

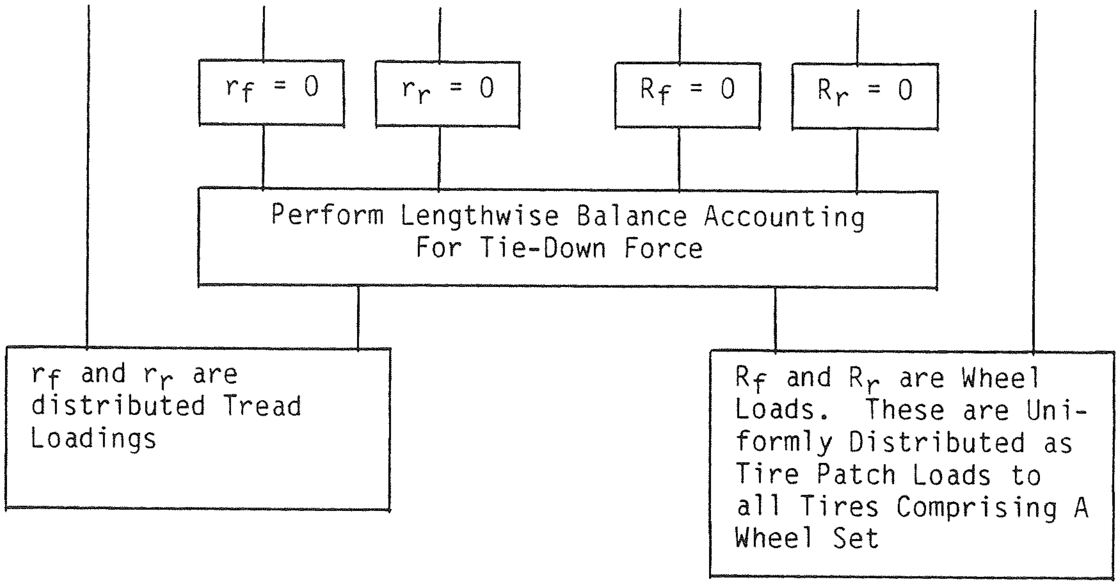


$R_v = R_1$ (left side)
$\quad = R_2$ (right side)
$r_r = 0$
$T = (F_1 Z_g/2 - R_v (X_g - X_1/3))/K$
$r_f = 2 (R_v + T \sin\Omega)/X_1$
$F_f = F_1/2 - T \cos\Omega$
where, $H = Z_{t2} \times \cos\Omega + X_{t2} \times \sin\Omega$
$K = H - X_1 \sin\Omega /3$

130-3-g.1 <u>Vehicle Load Analysis - Procedural Flowchart</u>




Perform Lateral Load Balance With Tie-Down Force Accounted For (Assume $R_1 = 0$)
Distribute Lengthwise Ship Motion Force Uniformly to each Side of Vehicle
Perform Following Analysis for Left and Right sides of Vehicle
Perform lengthwise force balance Accounting for Portion of Lengthwise Ship Motion Force Assume No Tie-Down Forces
Vehicle Type?
Tracked
Wheeled
What are r_f, r_r?
What are R_f, R_r?
$r_f \geq 0$ $r_r \geq 0$
$r_f < 0$
$r_r < 0$
$R_f < 0$
$R_r < 0$
$R_f \geq 0$ $R_r \geq 0$
Balance with Tie-Down Force Required. Assume...


$r_f = 0$
$r_r = 0$
$R_f = 0$
$R_r = 0$
Perform Lengthwise Balance Accounting For Tie-Down Force
r_f and r_r are distributed Tread Loadings
R_f and R_r are Wheel Loads. These are Uniformly Distributed as Tire Patch Loads to all Tires Comprising A Wheel Set

PART IV: ANALYSIS METHOD

130-3-h. Structural Parameters

The typical structure of modern Navy vehicle parking decks is a plate deck grillage of continuously welded construction. Stiffeners are continuous members of uniform size. This causes the individual structural elements to interact; therefore, the structural parameters of each member are altered to account for these effects.

130-3-h.1 Geometry

The vehicle parking deck consists of a flat plate and mutually perpendicular stiffeners supported by beams and/or girders and/or bulkheads. This is a statically indeterminate structural system. For design purposes, it may be treated as a grid, where the deck plate acts as a common top flange for the stiffeners, beams, and girders.

A typical deck scantling arrangement is shown in Figure 1. The structural model of this arrangement for finite element purposes is shown in Figure 2.

130-3-h.2 Effective Spans

The effective span length factor, e_s or e_b, is a function of the number spans of the member.

Where: e_s = effective span length factor of stiffener

e_b = effective span length factor of beam

See Table III for values of factors.

TABLE III - EFFECTIVE SPAN LENGTH FACTOR

Number of Spans	e_s or e_b
1	1.0
2	0.842
3	0.700
4	0.692
5 or more	0.684

130-3-h.3 Members Properties

The properties of the stiffeners, beams and girders are those of the actual member itself plus that portion of deck plating which acts as the upper flange. The effective breadth of the deck plating which may be assumed for this upper flange is a function of the plate thickness and material, span length, or spacing, whichever is less.

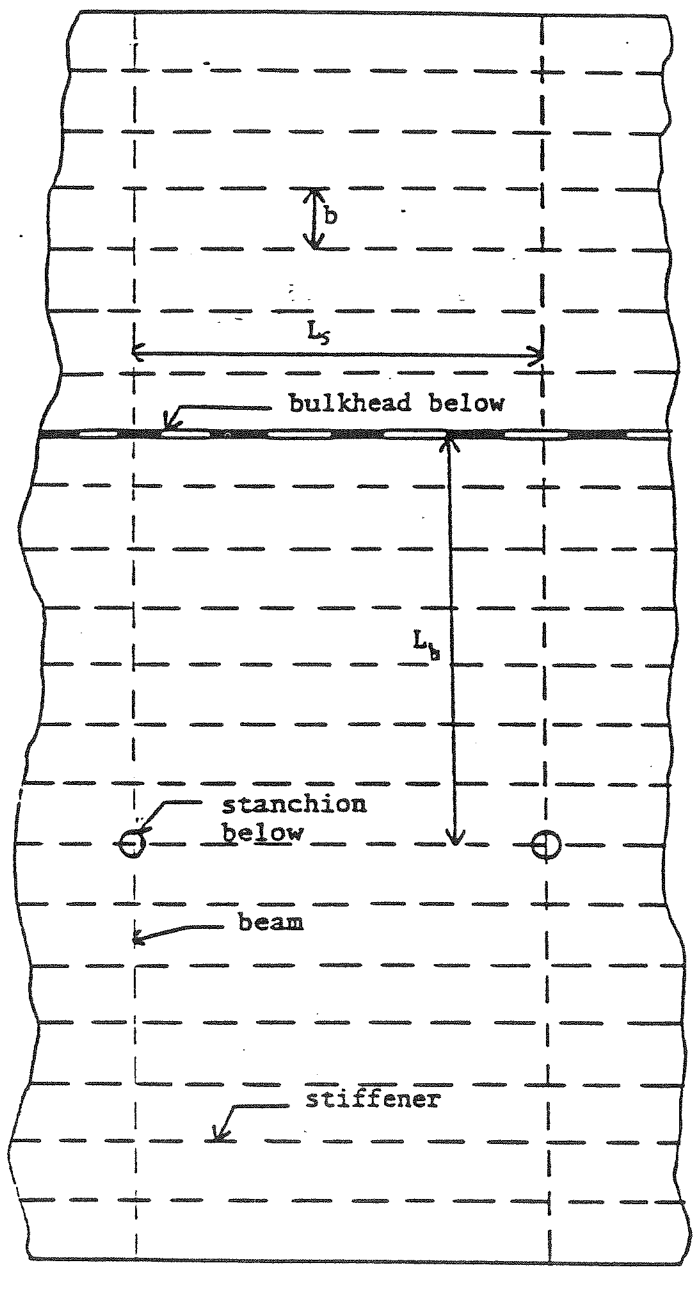


Figure 1 Typical Deck Scantlings

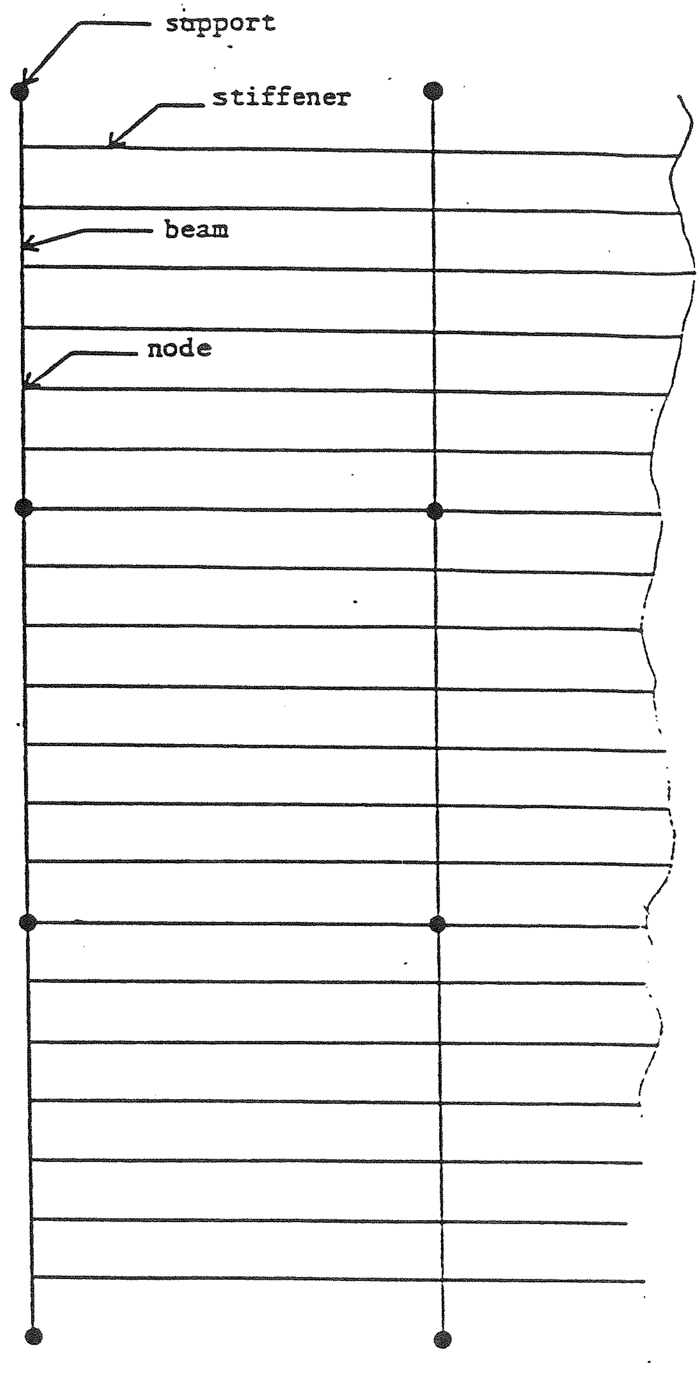


Figure 2 Typical Finite Element Model

$$b_e = \left\{ \begin{array}{l} 2t\sqrt{\frac{E}{F_y}} \\ 1/3\ L_s \\ b \end{array} \right\}, \text{ minimum of}$$

The combined properties of the member and associated plating can be found in Reference (g) or by direct calculation. Figure 3 provides graphical definitions of those dimensions which define a member's size. Figure 4 shows the common structural members used in vehicle parking decks.

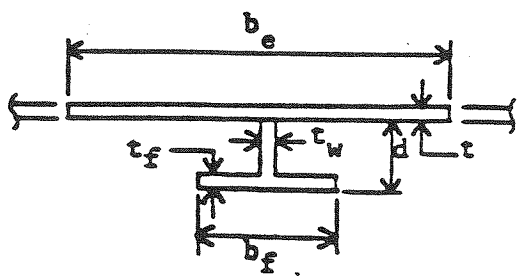
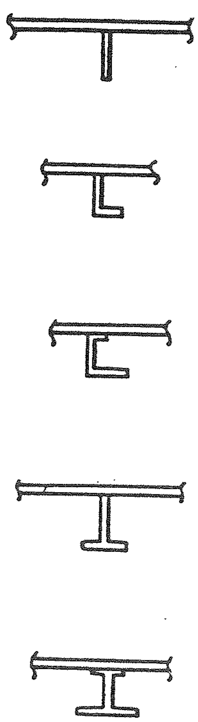


FIGURE 3 - MEMBER DIMENSIONAL VARIABLES

It is assumed that the section modulus at the flange will be the minimum and the section modulus at the plate will be the maximum.



FB , Flat Bar

L , Angle

C - L , Channel cut to Angle

T , Tee

I - T , I Beam cut to Tee

FIGURE 4 - COMMON STRUCTURAL MEMBERS

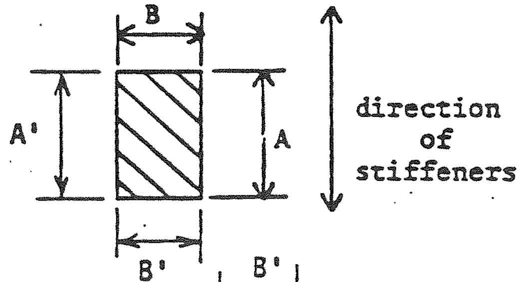
130-3-i. Load Patch Pattern Due to Vehicle Orientation

The orientation of the vehicle with respect to the stiffeners and the number of wheels per gear are resolved into patch(es) with associated load with respect to the stiffeners. The tire contact dimensions, length A, width B, and dual wheel spacing b' are resolved into patch dimensions, length in direction of stiffeners A', wideth perpendicular to stiffeners B', and dual patch spacing b", with a load that is modified as applicable.

When the vehicle is aligned with the stiffeners

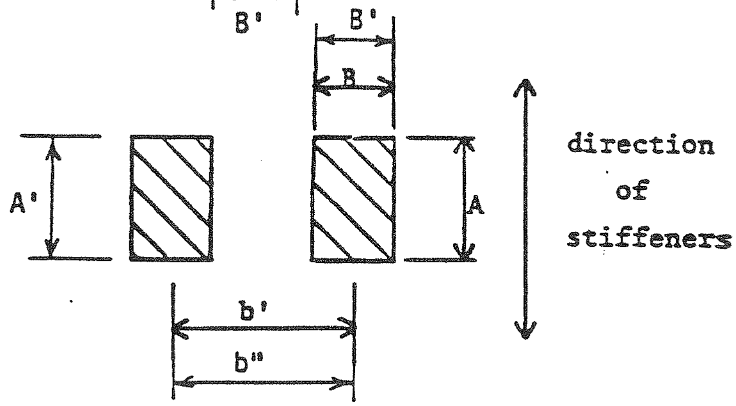
For single wheeled gear

A' = A
B' = B
P = R



For dual wheeled gear

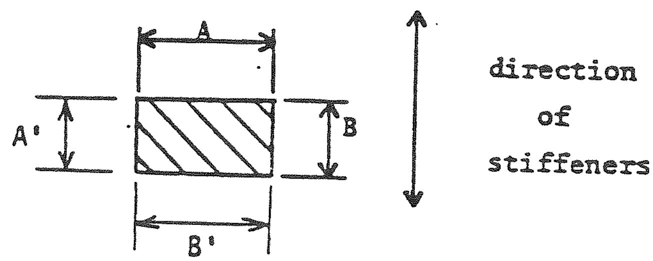
A' = A
B' = B
b" = b'; P = 1/2 R
or
B' = b' + B
P = R
if b' is very small



When the vehicle is aligned perpendicular to the stiffeners

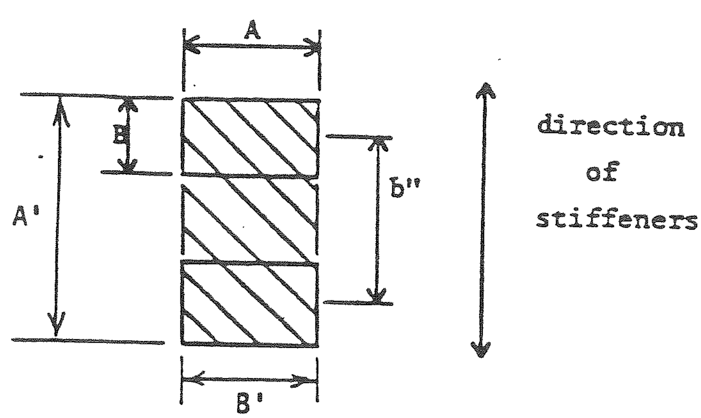
For single wheeled gear

B' = A
A' = B
P = R



For dual wheeled gear

B' = A
A' = b' + B
P = R

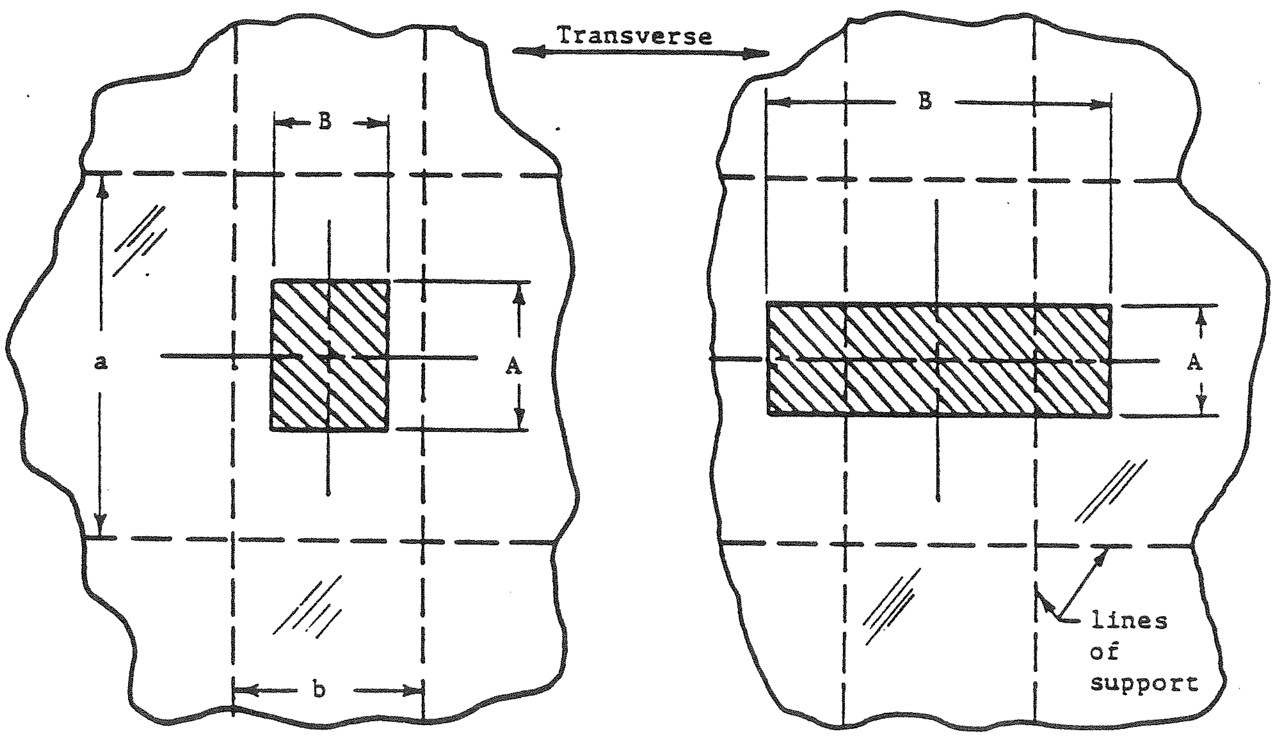


130-3-j. Response of Plating

Elastic response methods rather than plasticity or membrane methods are utilized for analysis of plating loads by these type of vehicles. The following approach is based on the study of Reference (k) and simplified approximation to the complex behavior for design purpose.

General studies have been performed to obtain a series of design curves for plate edge and midspan response which are included in the following graphs. It is presumed that critical stresses will be provided by one of these solutions although it can be shown that maximum respone will occur elsewhere for some assymetric loadings. However, it is assumed that these assymetric responses will never govern design.

Within this study two arrangements of patch loadings are considered and outlined as follows:



A) Load patch within plate panel

B) Load patch extending over onto adjoining panels

Edge moment at the middle of the short edge is not determined as it can be shown that it will never govern design. In addition, longitudinal or lengthwise bending moment at the plate panel midspan is not provided as it will not govern design of the immediate panel although it may be significant for studies involving combining localized stresses resulting from discrete loads with primary ship hull stresses.

Within this study, a series of coefficients were obtained from which midspan transverse plate bending moments, and transverse edge moments can be rapidly and easily determined. This solution is presented in graphical format as shown in Figure 5.1 through Figure 5.6 for β and Figure 6.1 through Figure 6.6 for γ. It is important to note that the solution presented within this report was specifically performed for metallic plating with Poisson's ratio equal to 0.30 and will thus not apply to certain non-metallic materials such as reinforced concrete and composite structures. These coefficients and equations for their use are summarized as follows:

Midspan bending moment

$M = \beta \times P$

where, M = transverse midspan bending moment

β = moment coefficient from graph (Figures 5.1 to 5.6)

Edge bending moment

$M = \gamma \times P$

where, M = transverse edge bending moment

γ = moment coefficient from graph (Figures 6.1 to 6.6)

Response for discrete patch load on continuous plating is used for design purpose. The graphs have been extrapolated for concentrated loads and very small-sized load patches.

The graphical solutions are included for discrete panel aspect ratios, a/b, from 1 to 6. For aspect ratios different from those in the graphs, it is necessary to interpolate values. For panels with a/b greater than 6 it is generally adequate to use the a/b=6 solution.

For single patch loads extending across the central panel or either panel adjoining it along the long edges, moment coefficients can be taken directly from the graphs. The affects of loading on all other panels, including the panels adjoining the central panel along the short edges, are assumed to be negligible and are not covered by this solution procedure. Hence, for excessively large or long patch loads, such as track loads, any component of the load patch that extends onto a panel other than the central or two transversely adjoining panels is simply ignored as it is in a region that has minimal effect on the central panel.

For any arrangement of multiple patch loads, whether symmetric or assymetric, extending across the central plate panel or the two adjoining panels along the long edge, it assumes that the loads located at the central panel and ignores the extended region.

Midspan response to linearly varying loads may be directly determined using the coefficient graphs. In this case, bending moement will be identical to that resulting from a uniform load of the same total magnitude as shown in the Figure 7.

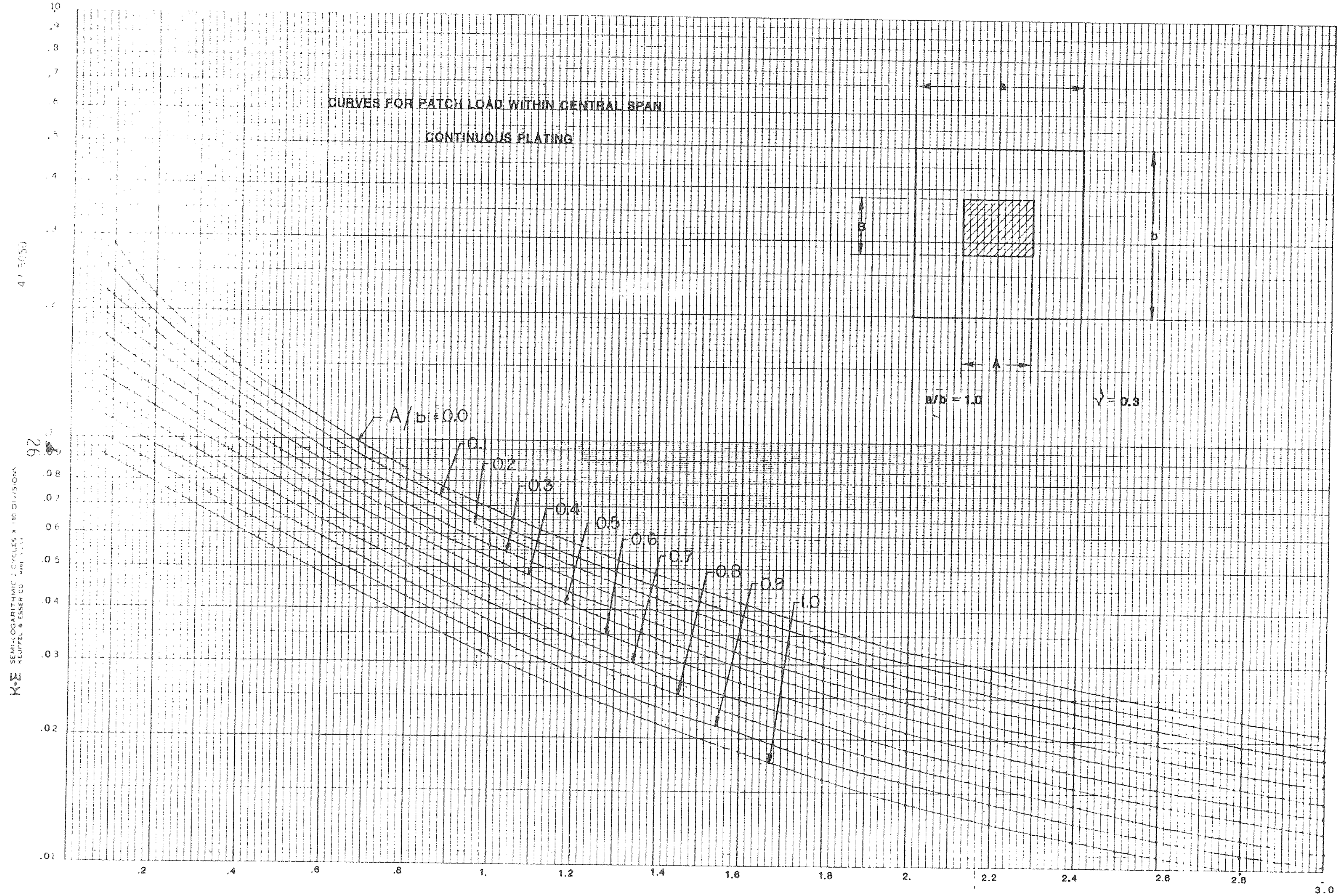


FIGURE 5.1 TRANSVERSE PLATE MIDSPAN BENDING MOMENT COEFFICIENT FOR CENTERED PATCH LOADS

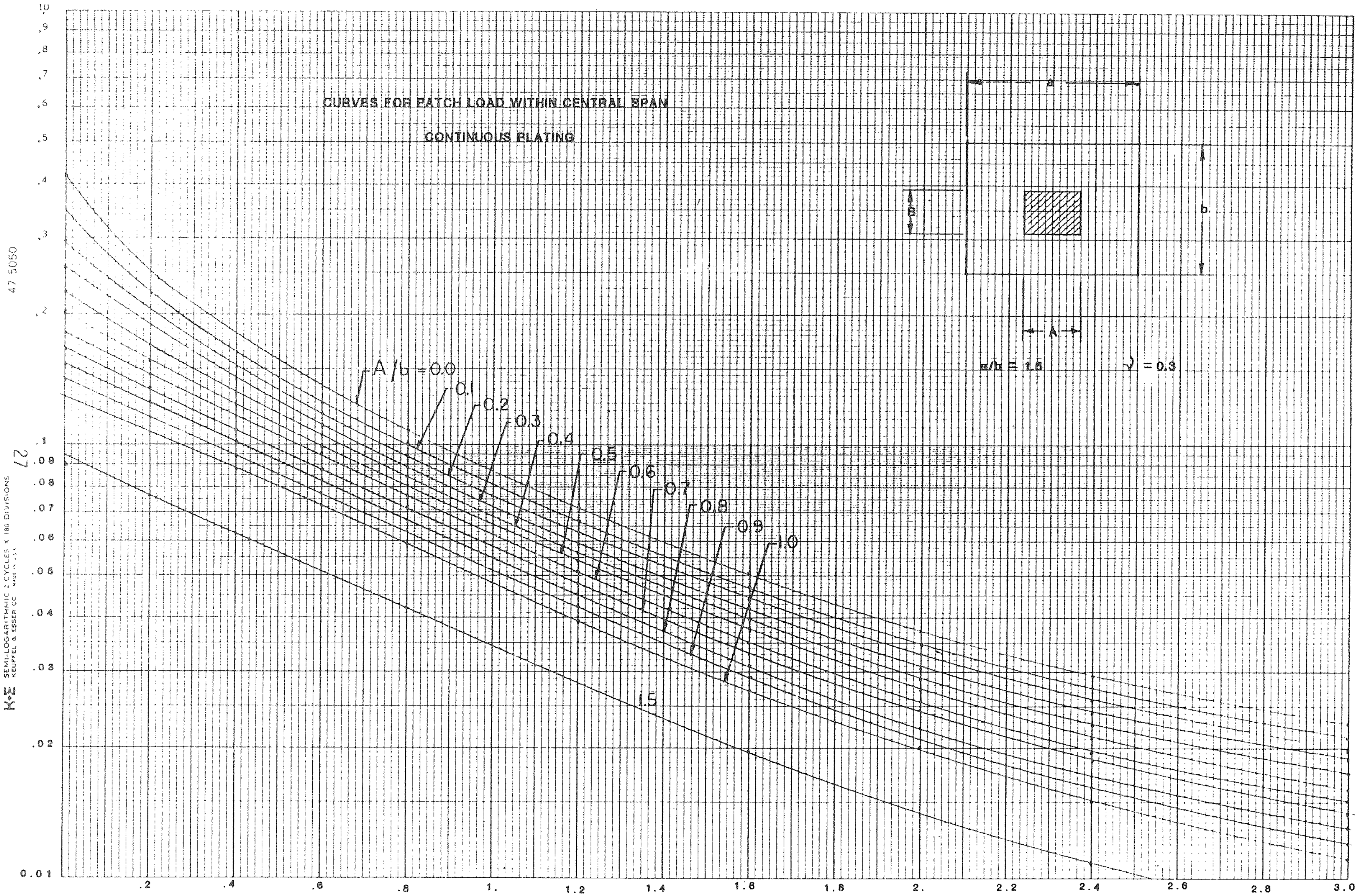


FIGURE 5.2 TRANSVERSE PLATE MIDSPAN BENDING MOMENT COEFFICIENT FOR CENTERED PATCH LOADS

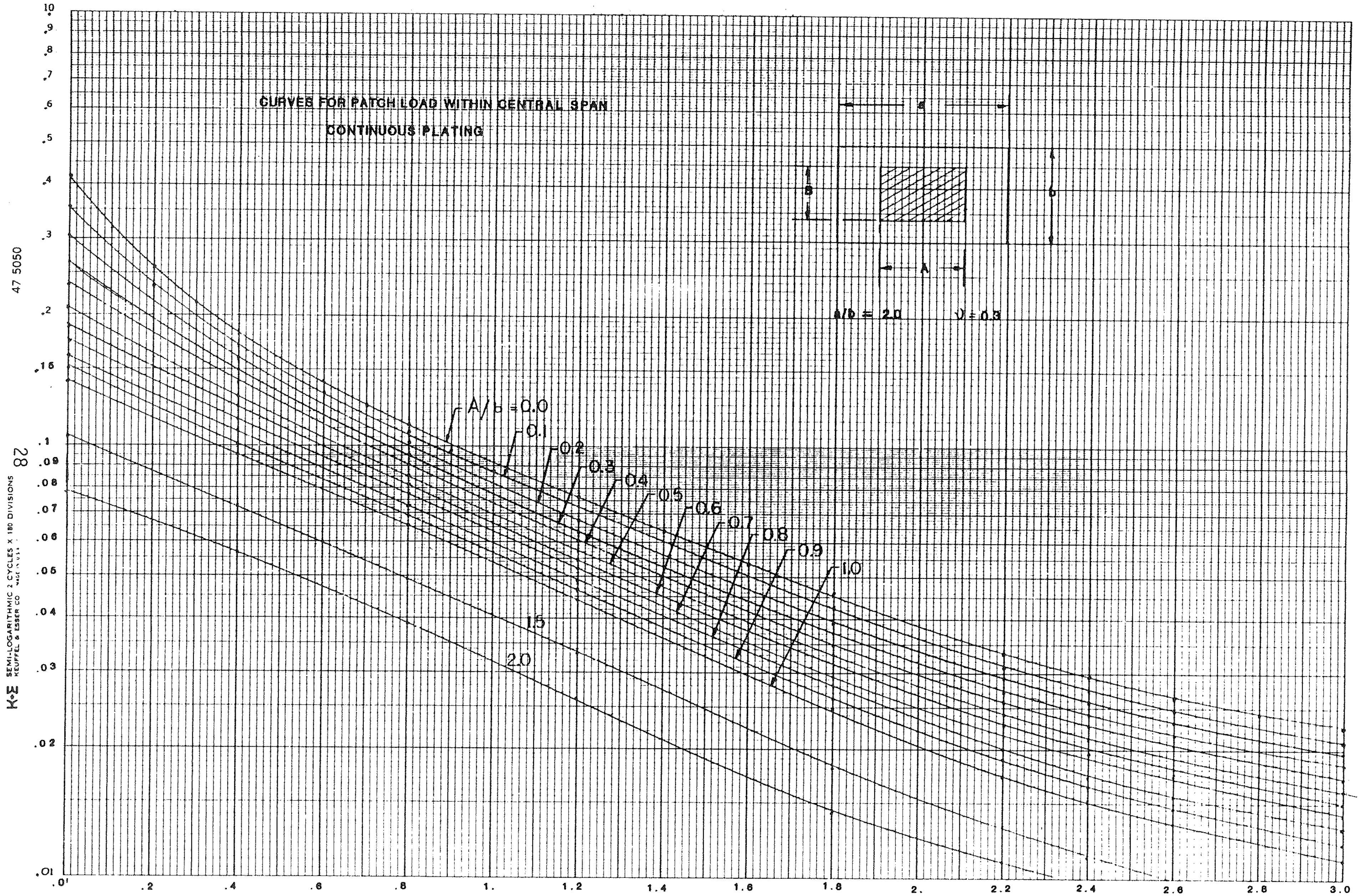


FIGURE 5.3 TRANSVERSE PLATE MIDSPAN BENDING MOMENT COEFFICIENT FOR CENTERED PATCH LOADS

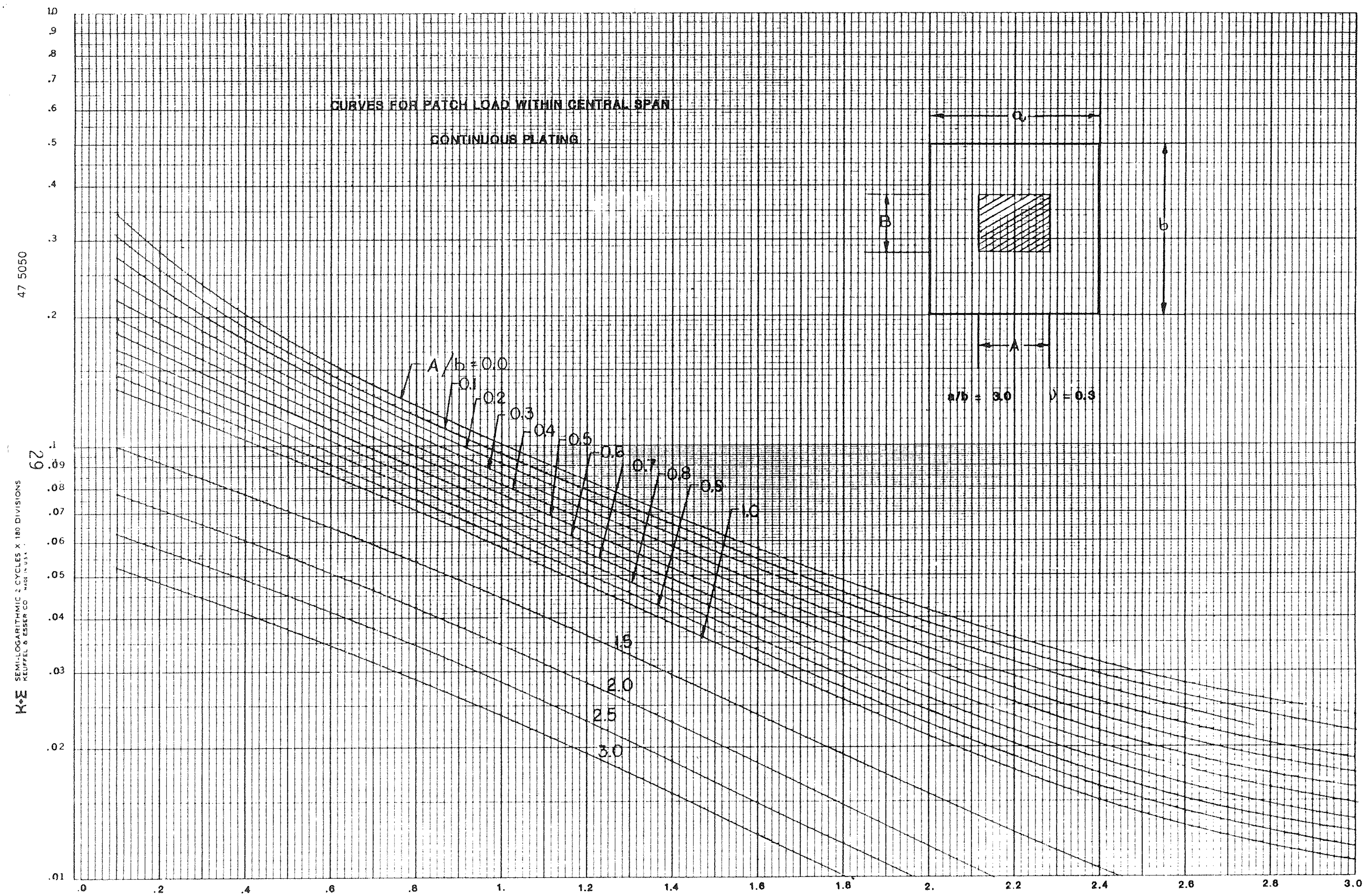


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FIGURE 5.4 TRANSVERSE PLATE MIDSPAN BENDING MOMENT COEFFICIENT FOR CENTERED PATCH LOADS

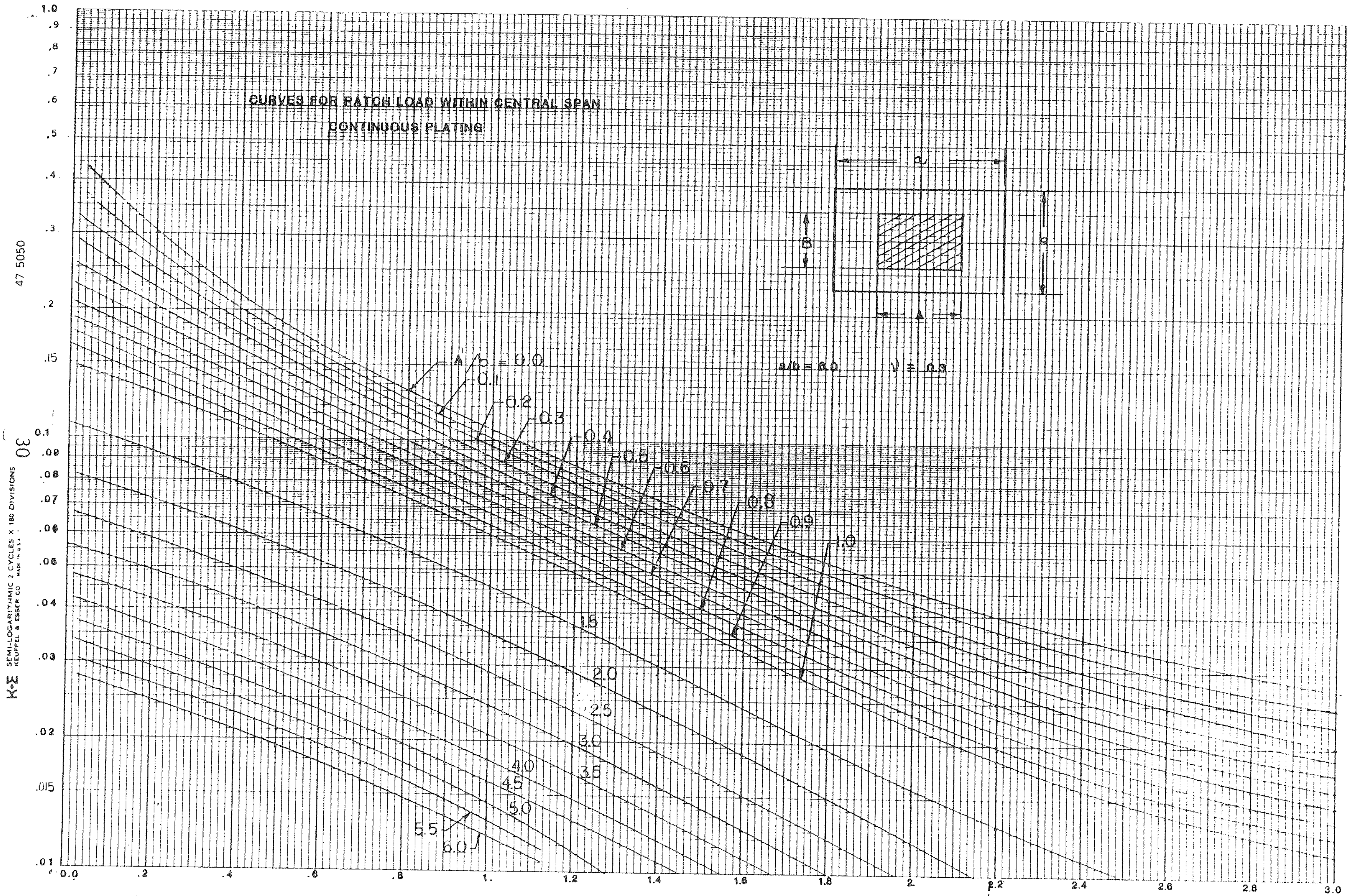
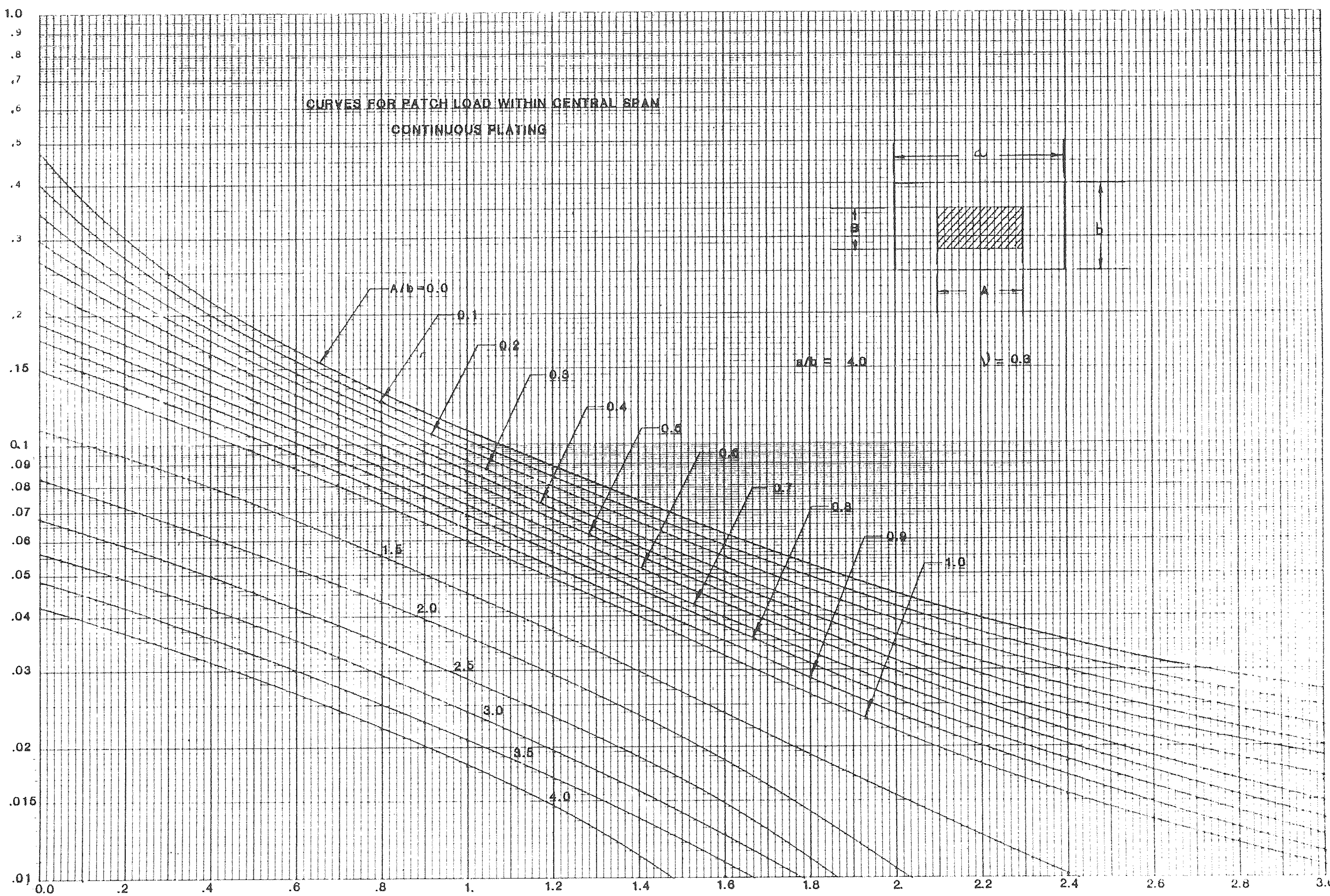


FIGURE 5.5 TRANSVERSE PLATE MIDSPAN BENDING MOMENT COEFFICIENT FOR CENTERED PATCH LOADS



K&E SEMI-LOGARITHMIC 2 CYCLES X 180 DIVISIONS KEUFFEL & ESSER CO. MADE IN U.S.A. 47 5050

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FIGURE 5.6 TRANSVERSE PLATE MIDSPAN BENDING MOMENT COEFFICIENT FOR CENTERED PATCH LOADS

Figure 6.1 TRANSVERSE PLATE EDGE BENDING MOMENT COEFFICIENT, $\gamma = M/P$ FOR EDGE-ALIGNED PATCH LOADS, FOR $\nu = 0.3$

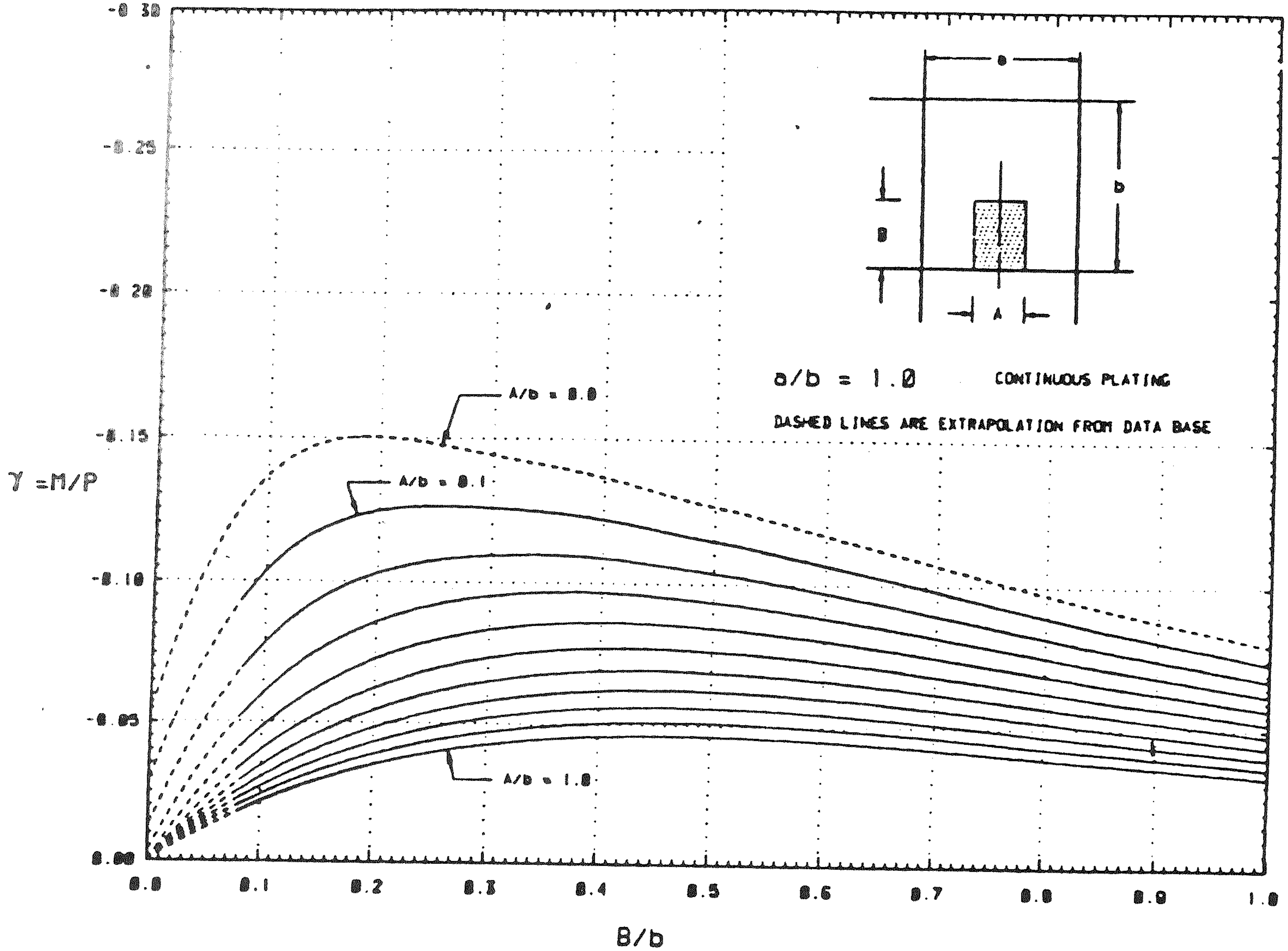

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Figure 6.2 TRANSVERSE PLATE EDGE BENDING MOMENT COEFFICIENT, $\gamma = M/P$ FOR EDGE-ALIGNED PATCH LOADS, FOR $\nu = 0.3$

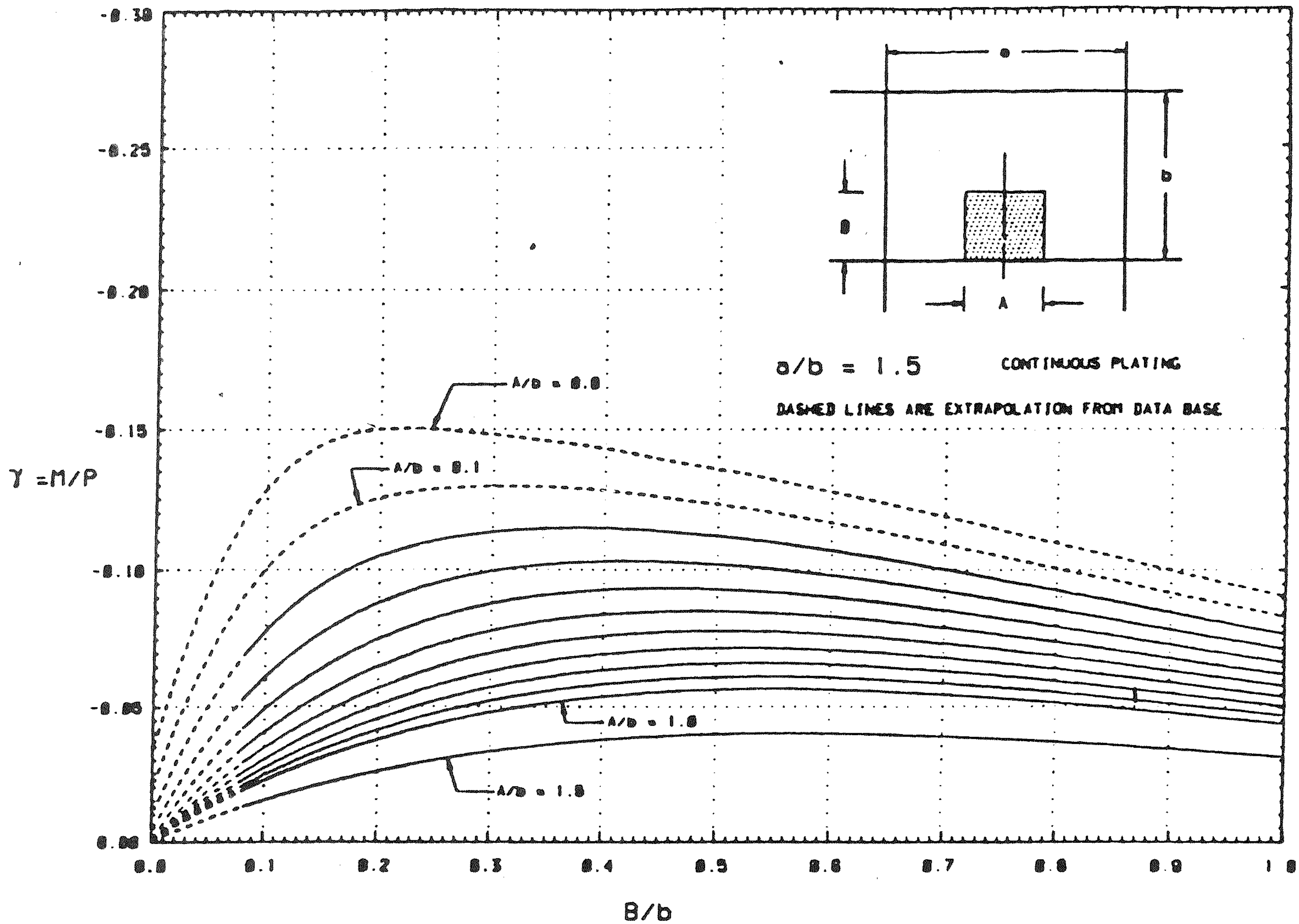


Figure 6.3 TRANSVERSE PLATE EDGE BENDING MOMENT COEFFICIENT, $\gamma = M/P$ FOR EDGE-ALIGNED PATCH LOADS, FOR $\nu = 0.3$

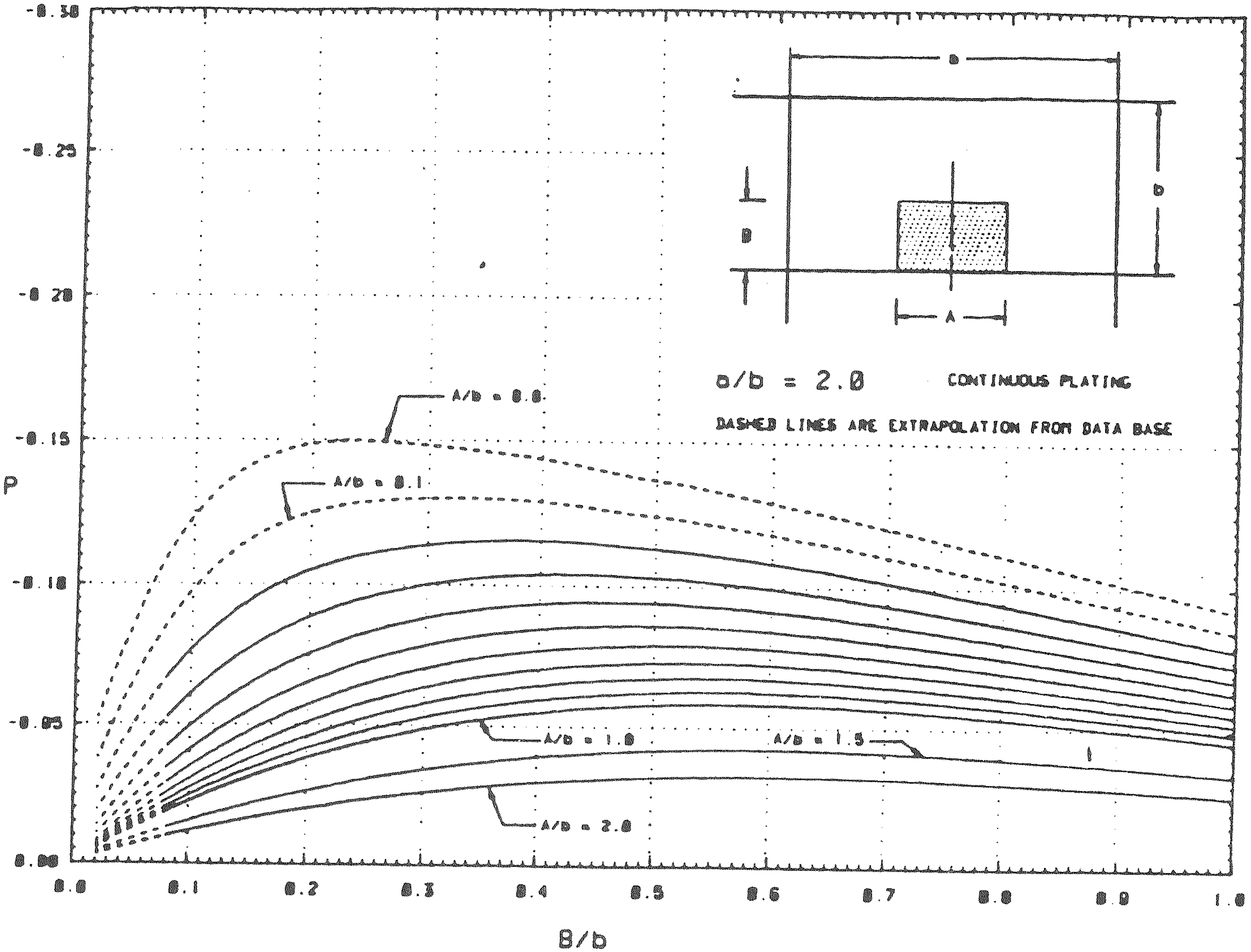


Figure 6.4 TRANSVERSE PLATE EDGE BENDING MOMENT COEFFICIENT, γ = M/P FOR EDGE-ALIGNED PATCH LOADS, FOR ν = 0.3

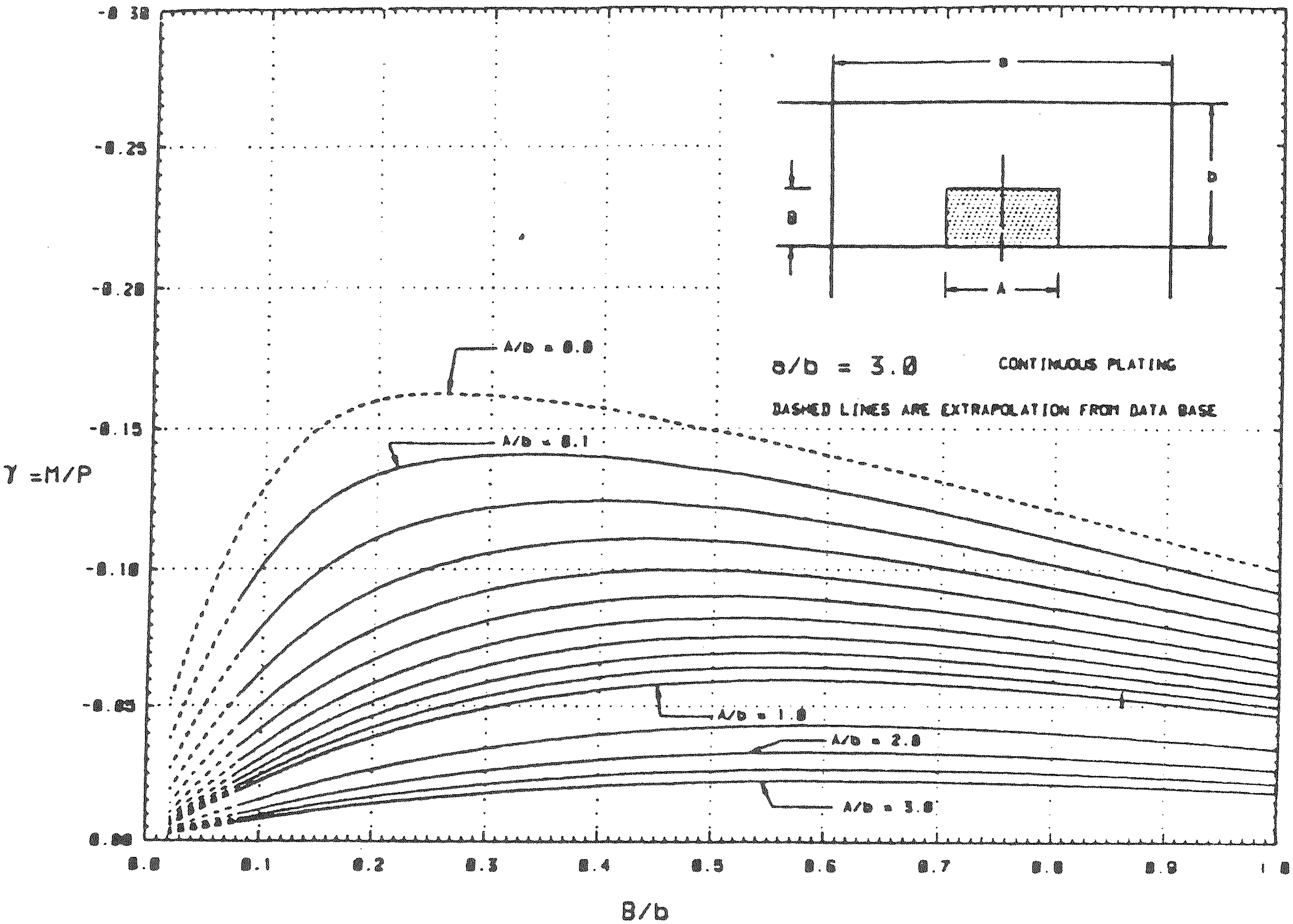


Figure 6.5 TRANSVERSE PLATE EDGE BENDING MOMENT COEFFICIENT, γ = M/P FOR EDGE-ALIGNED PATCH LOADS, FOR ν = 0.3

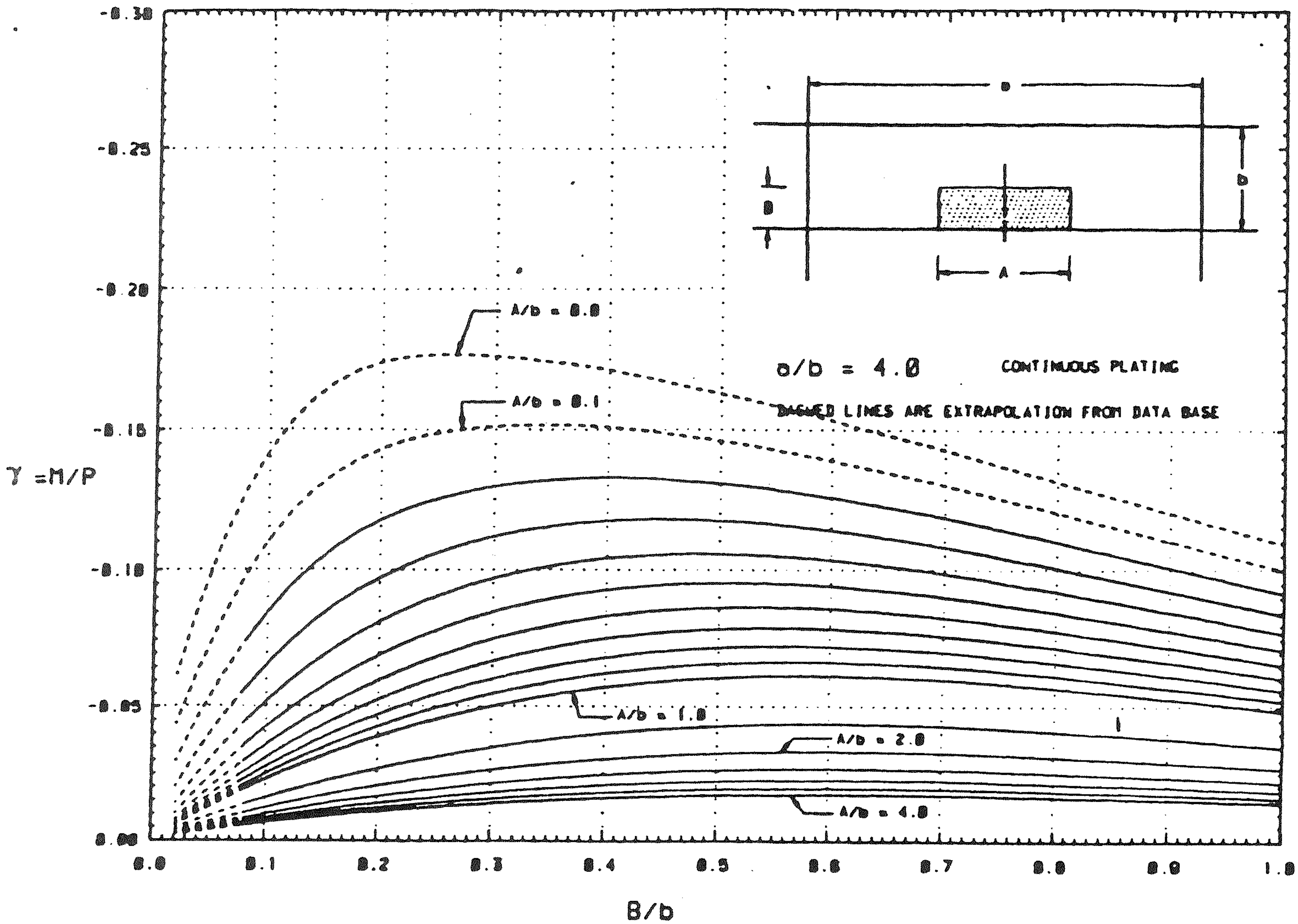
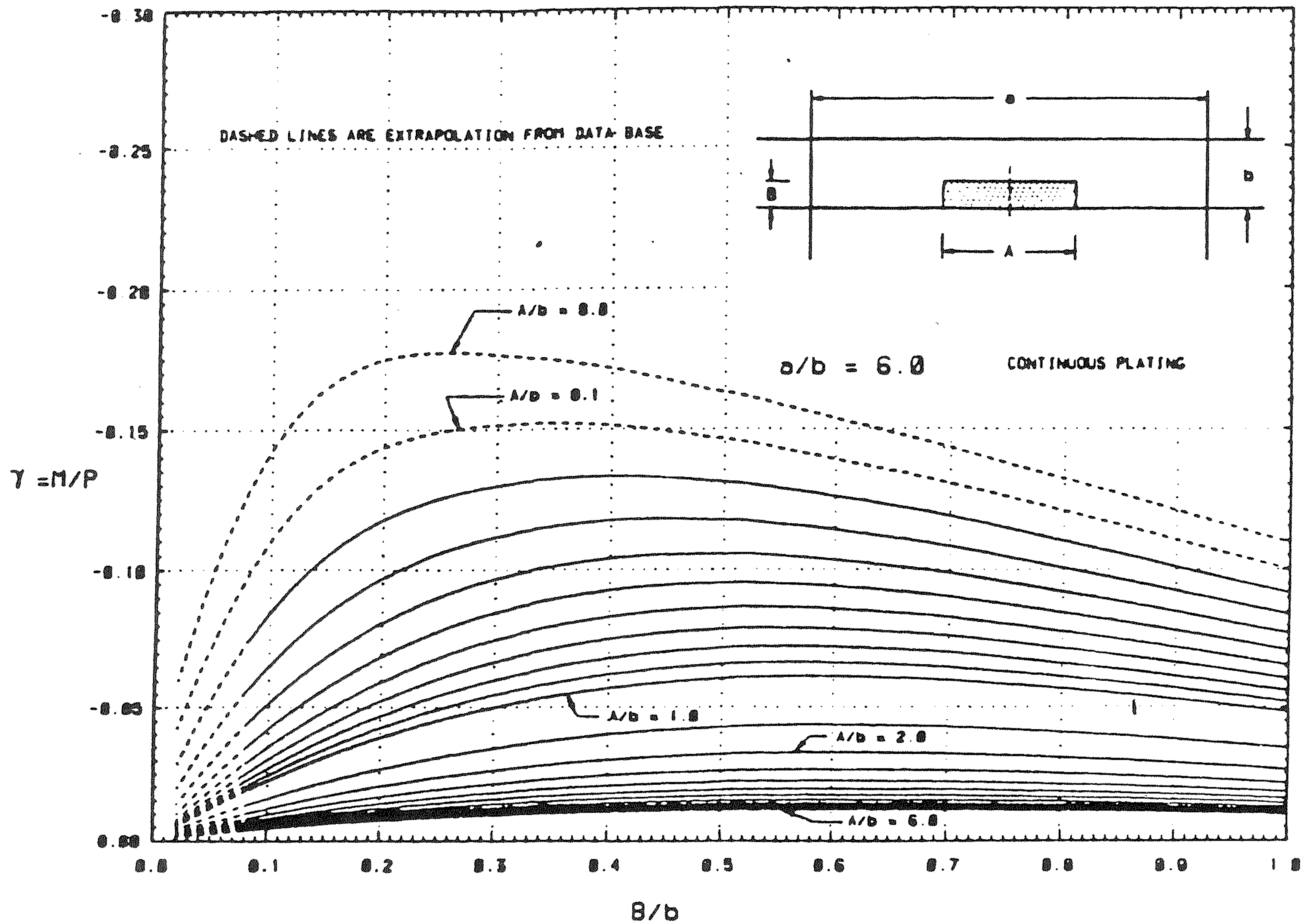


Figure 6.6 TRANSVERSE PLATE EDGE BENDING MOMENT COEFFICIENT, $\gamma = M/P$ FOR EDGE-ALIGNED PATCH LOADS, FOR $\nu = 0.3$



These two load cases produce identical midspan responses

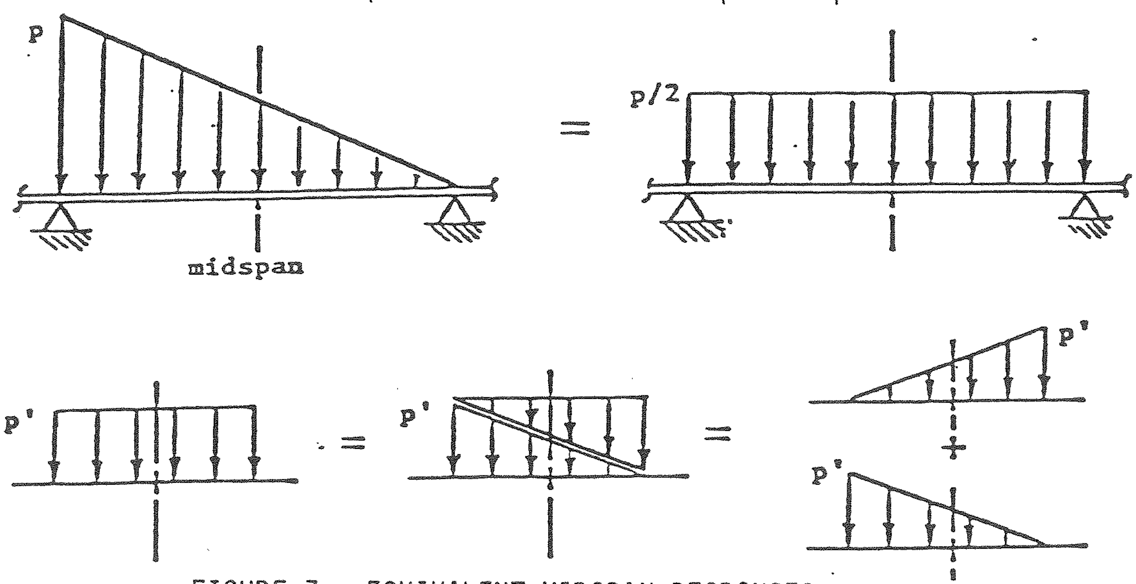
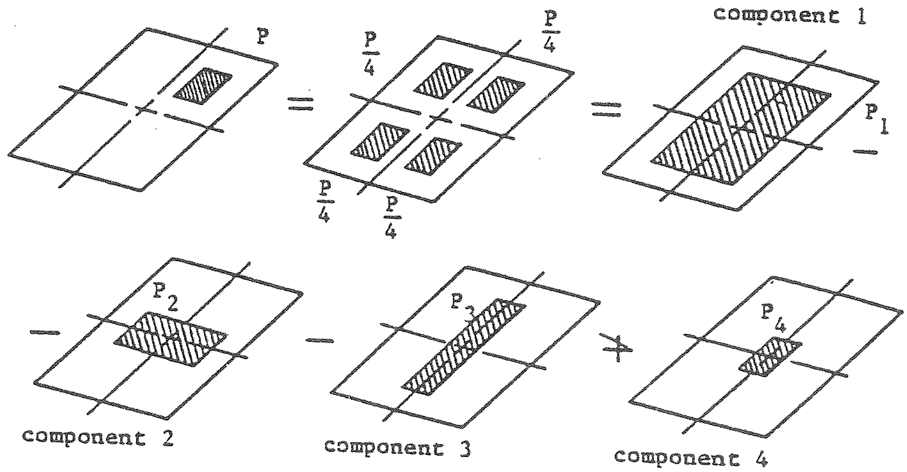
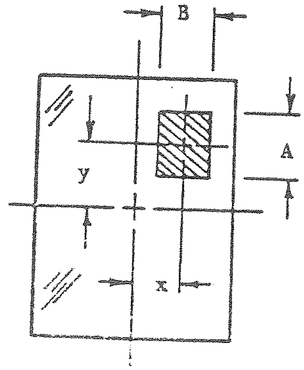


FIGURE 7 - EQUIVALENT MIDSPAN RESPONSES

For any asymmetric multiple patch loads within one panel, the midspan response due to the multiple patch loads will be the sum of each single patch load response within the panel. The general midspan response equation for asymmetric patch load can be obtained from the following illustrated procedures (Worksheets are presented in Appendix F)





where for patch component n,

$$P_n = \frac{A_n B_n}{4AB} P$$

and dimensions A_n and B_n are defined as follows:

component	A_n	B_n
1	2y + A	2x + B
2	2y - A	2x + B
3	2y + A	2x - B
4	2y - A	2x - B

DDS 130-3

General midspan response equation for assymmetric patch P,

Response for P = Response for P_1 - Response for P_2 - Response for P_3 + Response for P_4

The maximum bending moment at midspan of the plate panel.

$$M_{max} = \sum_{i=1} (\beta P)_i$$

The maximum plate stress is then calculated to be

$$f_p = \frac{6}{C_o t^2} \sum_{i=1} (\beta P)_i \quad \text{or} \quad \frac{6}{C_o t^2} \sum_{i=1} (\gamma P)_i$$

where C_o is thickness reduction factor which can be taken from Table IV.

For transversely framed decks subject to parking loads, the plating should be checked with the combined primary and secondary stresses for buckling in accordance with Reference (b).

The required plating thickness, t_{reqd} can be determined as follows:

$$t_{reqd} = \sqrt{\frac{6 \sum_{i=1} (\beta P)_i}{C_o \sigma_p}} \quad \text{or} \quad \sqrt{\frac{6 \sum_{i=1} (\gamma P)_i}{C_o \sigma_p}}$$

where the allowable plating stress, σ_p is as defined in Section 130-3-m.

Appendix D is an abridged description of the plating analysis method and criteria for easier reference. Appendix F contains standard calculation sheets.

A deck function coefficient, C_o is used to provide a relationship between allowable permanent set, load probability, and deck function. Where deck function is determined by the deck's contribute to the hull girder strength and by the vehicle operations performed on the deck. For a particular sea condition and deck function, C_o is given by Table IV.

C_o (Steel or Aluminum)

DECK FUNCTION	MODERATE SEA PARKING AND OPERATION	STORM SEA PARKING
Strength Deck	3.4	2.8
Non Strength Deck	4.2	3.5

TABLE IV Deck Function Coefficient, Co

130-3-k. Responses of Stiffeners

The following analysis procedures incorporate the grillage effects into the stiffener analysis. Acting as continuous members, the stiffeners rest on beams which carry the load to the supports. The beams deflect proportionately to the loads they carry and thus provide elastic supports for the stiffeners.

The loading condition which produces the maximum bending moment is assumed when the critical patch load is placed at midspan directly and superimposed with other patches at the other locations over the stiffener within the same span. This maximum moment is the summation of (1) the moment due to the live load, M_o, from patch load influence lines assuming the stiffener as a continuous beam on rigid supports; (2) the added moment due to the flexibility of the beam supports, M_c; and (3) the moment due to the dead weight of the plating and stiffener, M_d.

130-3-k.1 Regular Structural Scantlings

The following analysis is based on strip theory and the use of influence lines. The strength values determined by this approach are based primarily on References (e) and (j) with support from those documents listed in the bibliography.

The analysis procedure is applicable to longitudinal or transverse framing where the longitudinal or transverse stiffeners are continuous beams over equally spaced supports. For simplicity, the term stiffener will refer to either longitudinal or transverse stiffeners, and the term beam will refer to the members which support the longitudinal or transverse stiffeners.

Bending Moment Influence Lines due to Single Patch

Influence line equations provide a rapid means of estimating bending moments in both stiffened panels and plates due to vehicle loads. By utilizing the principal of superposition for multiple loads, these equations can be used to produce influence lines for general or specific vehicles. These lines graphically describe the deck response to a variably positioned vehicle on the deck. This is useful for determining critical loadings and positioning for critical loadings.

The following diagrams illustrate the basic influence lines for a point load of variable position on a continuous deck with uniformly spaced inelastic supports:

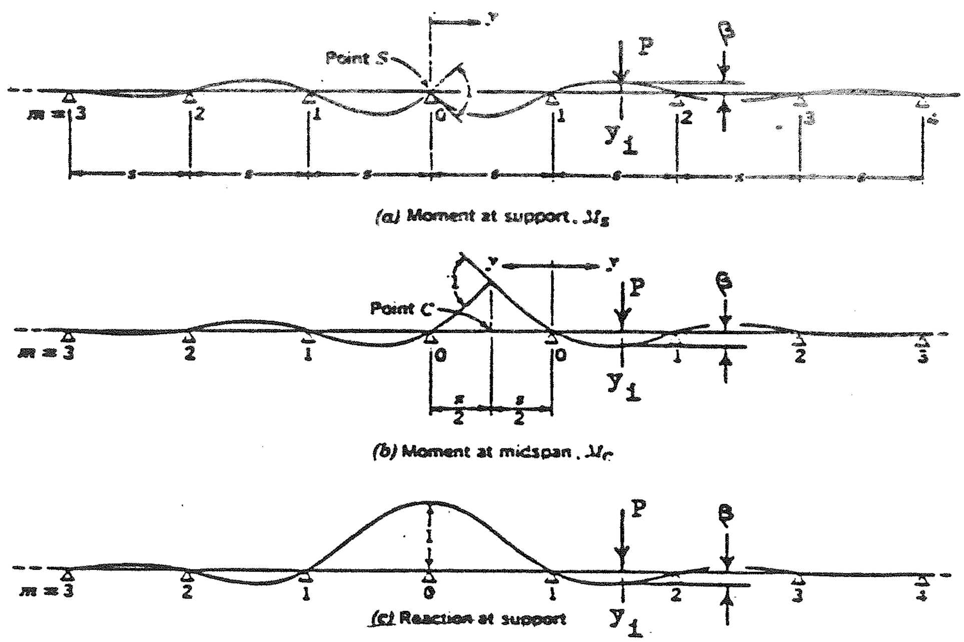


FIGURE 8 - INFLUENCE LINES OF A CONTINUOUS BEAM ON A RIGID SUPPORTS

Figure 8(a) provides bending moment at support S for a load P located at point y_i. At point y_i an ordinate of β may be measured and bending moment at support is given by the following equations:

$$M_S = \beta \times s \times P$$

Similarly, the moment at the midspan may be obtained from Figure (b) and reaction force at support 0 may be obtained from curve (c) using the formulas

$$M_C = \beta \times s \times P$$

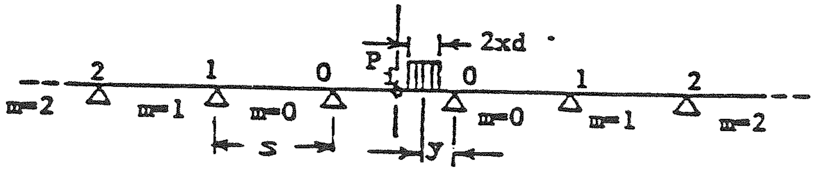
$$R_O = \beta \times P$$

These curves are used to compute moments and reaction forces for a load set consisting of several different loads at various positions along the beam by superimposing the moments and reactions caused by the separate loads. Generally this process can be described as follows:

1. For moments: $M_t = \sum_{i=1} (\beta_i \times s \times P_i)$
2. For reactions: $R_t = \sum_{i=1} (\beta_i \times P_i)$

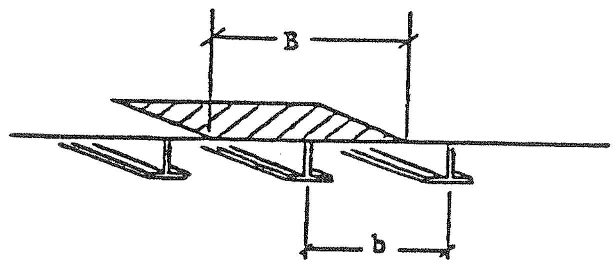
These equations for these unit influence lines are presented graphically for a range of load patch width to span ratios. Influence lines for moment at support, influence lines for moment at midspan and influence lines for reaction at support are presented in Figures 9, 10 and 11, respectively.

These curves are included with a relative ordinate scale rather than an absolute scale for purposes of clarity. The use of these curves is described and illustrated in the following graphs. (Figure 9 thru Figure 11).



Wheel Load Reduction Factor, $\emptyset_1$

Each wheel load is transversely carried by several adjoining stiffeners. For design purposes, the maximum load portion to each stiffener must be determiend as shown in the following illustration:



This critical load distribution to a single stiffener is obtained from the Wheel Load Reduction Curve (Figure 12), by calculating the patch width to stiffener spacing ratio B/b, and reading the load factor, $\emptyset_1$ (i = 0). The required load is then calculated by multiplying the wheel load by the factor.

Plating Distribution Load Factor

To account for the distribution effects of the plating, the plating load distribution factor, $\emptyset_2$, is used. To determine the plating load distribution factor, $\emptyset_2$, calculate the relative rigidity between the plating and stiffener and use in Figure 13. The relative rigidy between the plating and stiffener is calculated by the following equation.

$$\gamma_{ps} = \frac{(e_s L_s)^4 t^3}{3.49\ b^3\ \pi^4\ I_s}$$

Maximum Bending Moment and Stress

The moment due to the live load over rigid supports, M_o is then calculated by applying the appropriate load factors using the following equation.

$$M_o = \left(\frac{M}{PL_s}\right) PL_s\ \emptyset_1\ \emptyset_2$$

FIGURE 9

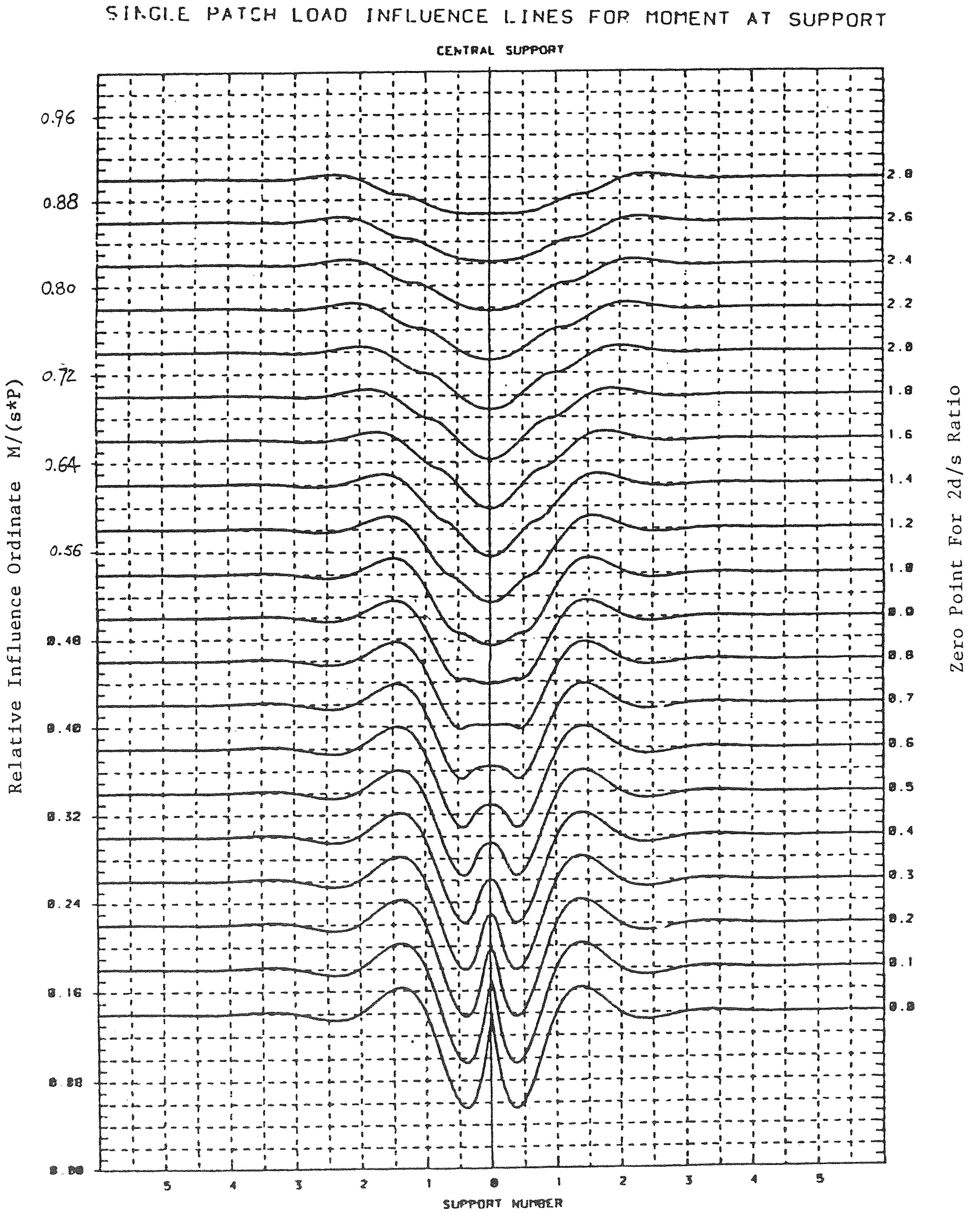

SINGLE PATCH LOAD INFLUENCE LINES FOR MOMENT AT SUPPORT
CENTRAL SUPPORT
Relative Influence Ordinate M/(s*P)
0.96
0.88
0.80
0.72
0.64
0.56
0.48
0.40
0.32
0.24
0.16
0.08
0.00
Zero Point For 2d/s Ratio
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
5 4 3 2 1 0 1 2 3 4 5
SUPPORT NUMBER

FIGURE 10

SINGLE PATCH LOAD INFLUENCE LINES FOR MOMENT AT MIDSPAN

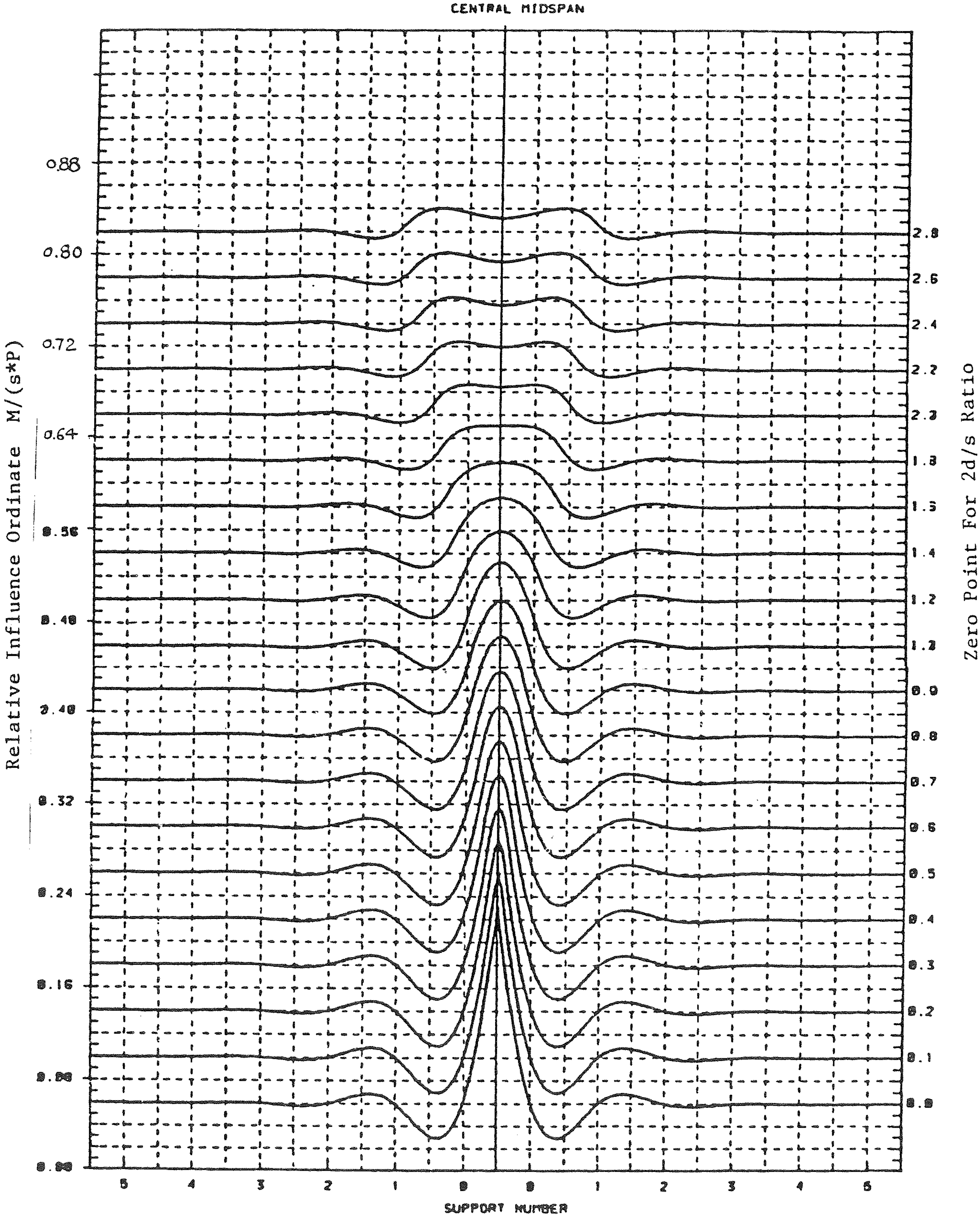
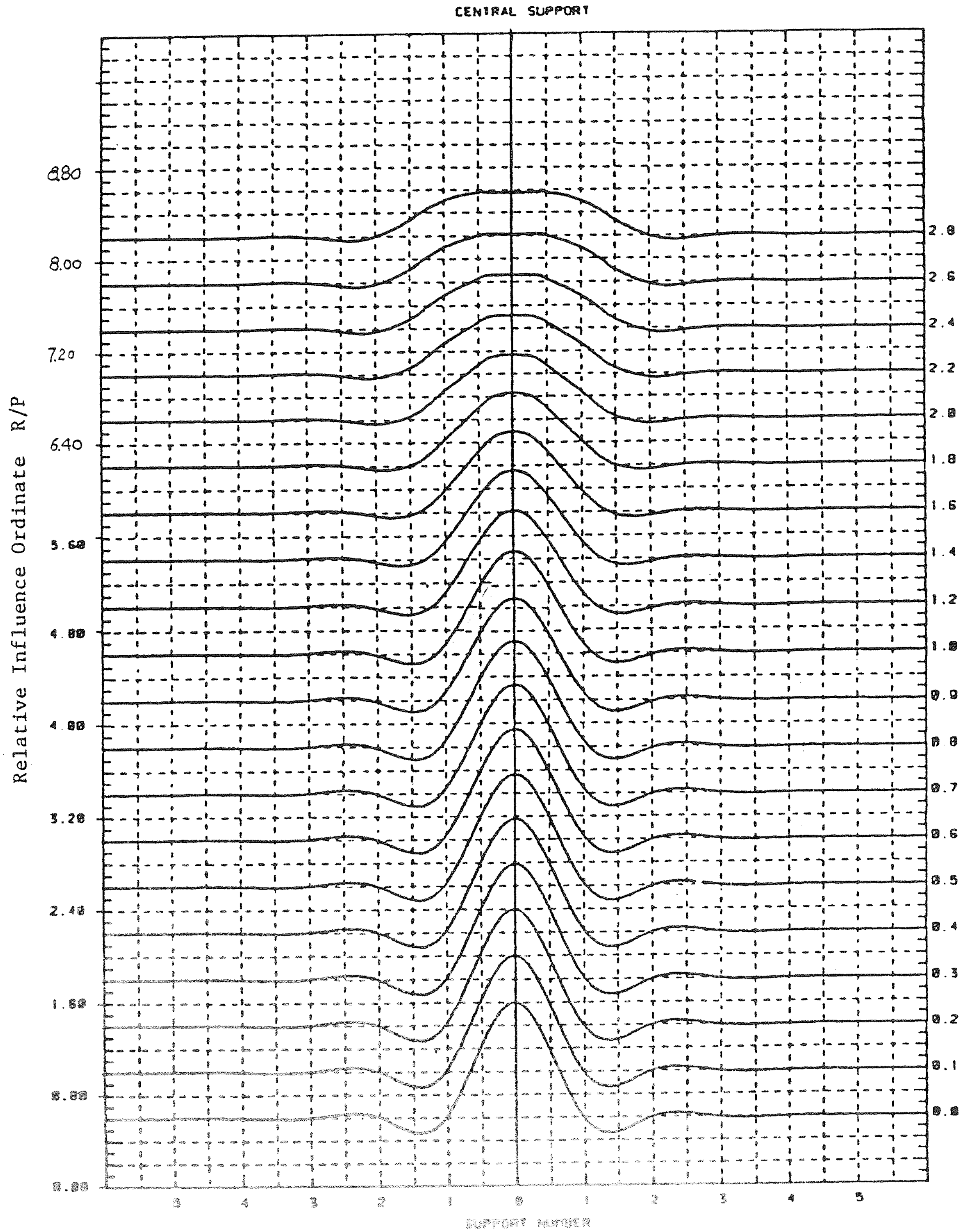


FIGURE 11

SINGLE PATCH LOAD INFLUENCE LINES FOR REACTION AT SUPPORT



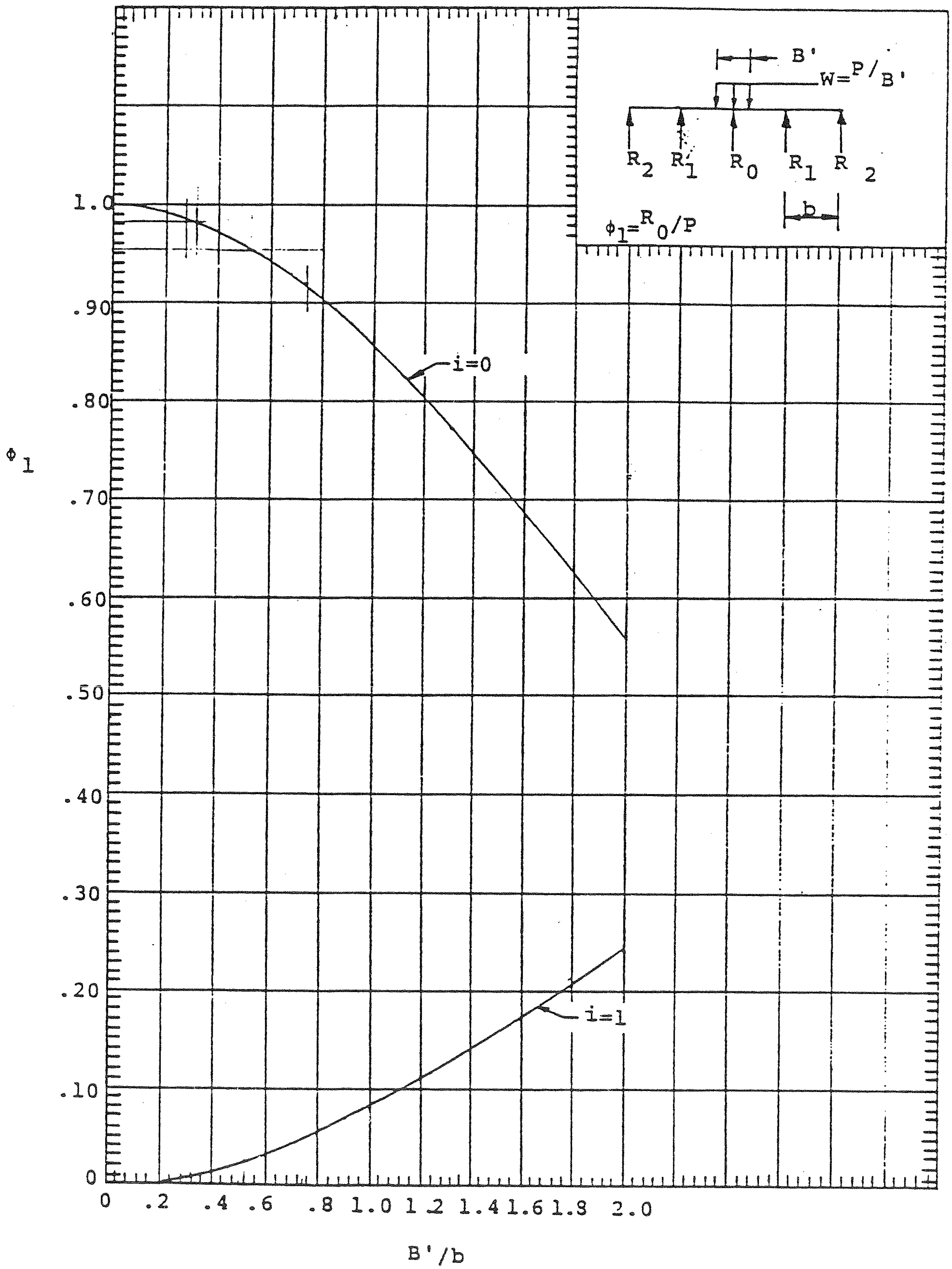


Figure 12 Patch Width Load Distribution Factor, Stiffener, ϕ_1

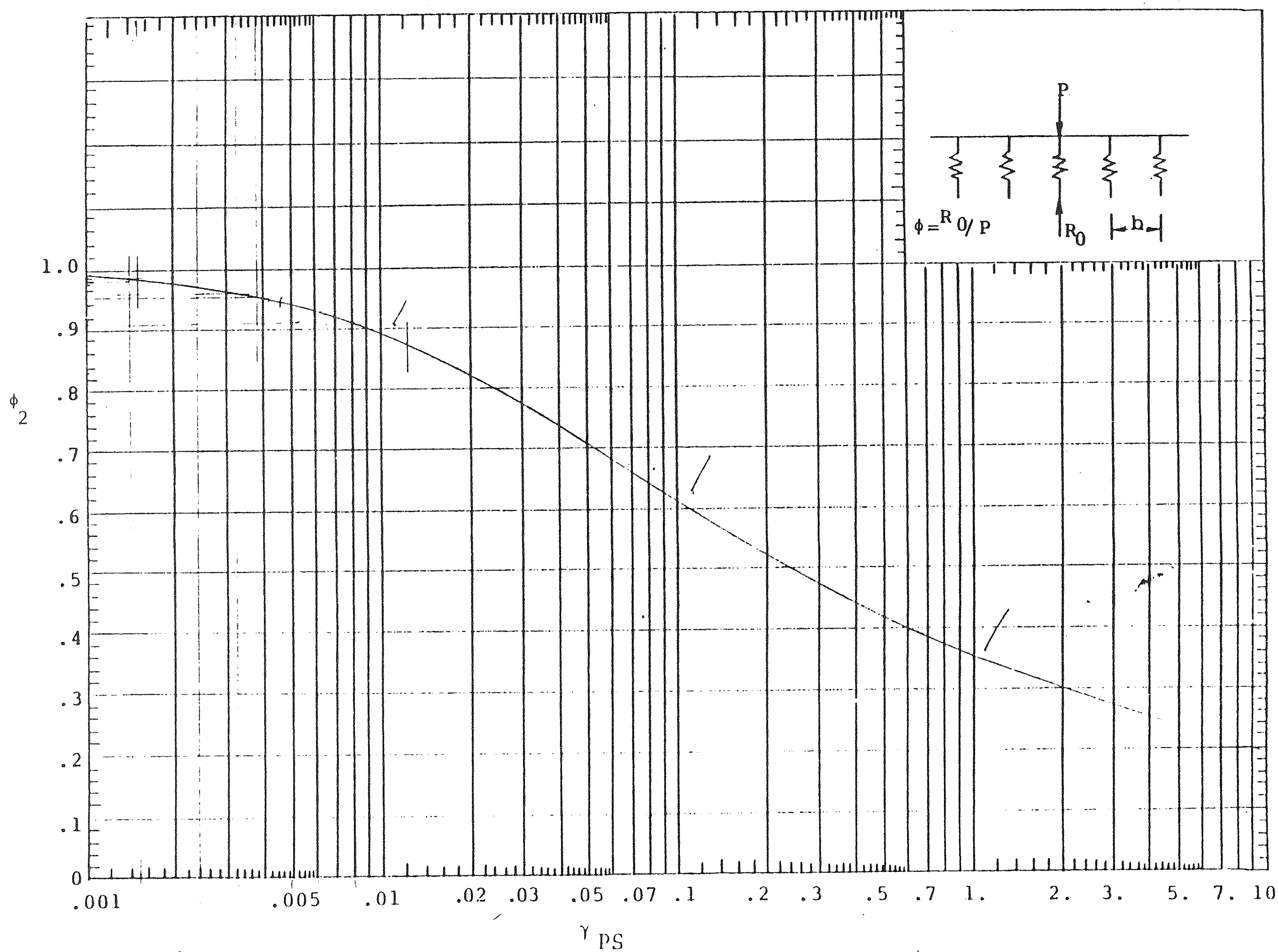


Figure 13 Plating Load Distribution Factor, Stiffener, ϕ_2

For multiple patch loads

$$M_o = L_s \phi_1 \phi_2 \sum_i (\beta P)_i$$

where $\beta = (M/PL_s)$

The moment due to flexibility of the beam supports, M_c, is determined by calculating the relative rigidity between the stiffener and the beam to obtain the moment correction coefficient due to flexure of the beams then applying the appropriate load factors. If bulkheads are supporting stiffeners in lieu of beams, then the stiffener supports are considered rigid; therefore, $M_c = 0$.

The relative rigidity between the stiffener and beam is calculated by the following equation.

$$\gamma_{sb} = \frac{(e_b L_b)^4 I_s}{.684\ b\ (e_s L_s)^3 \pi^4 I_b}$$

The moment correction coefficient due to flexure of the beams, M_c/RL_s is obtained from Figure 14 using the relative rigidity, γ_{sb}.

To account for multiple patch loads, and plating and stiffener load distributions, the load on the transverse beam for determining the moment in the stiffeners due to beam flexture is assumed to be the Fourier Series component representation of the load. This representation consists of a characteristic loading (characteristic load, R_o, and characteristic load width, B_o) and a shape function (beam loading coefficient, ϕ_4).

The beam cnaracteristic load, R_o is calculated based on the total patch load as follows:

$$R_o = \begin{cases} R/B' & \text{, single patch} \\ R/(b'' + B') & \text{, dual patch} \end{cases}$$

The beam characteristic load width, B_o is calculated based on the total gear distribution as follows:

$$B_o = \begin{cases} B'/2 & \text{, single patch} \\ 1/2\ (b' + B') & \text{, dual patch} \end{cases}$$

The notations of width and length are defined in Paragraph 130-3-J.

The beam load coefficient, ϕ_4, is calculated by one of the following equations, depending on the vehicles orientation to the stiffeners.

For the vehicle aligned with the stiffeners and where:

$$L_b \geq 1.5\ s$$

the beam loading coefficient is:

$$\phi_4 = \frac{4}{\pi} \cos\left(\frac{\pi s}{2 L_b}\right) \sin\left(\frac{\pi B_o}{L_b}\right)\left(1 + \cos \frac{\pi s}{2 L_b}\right)$$

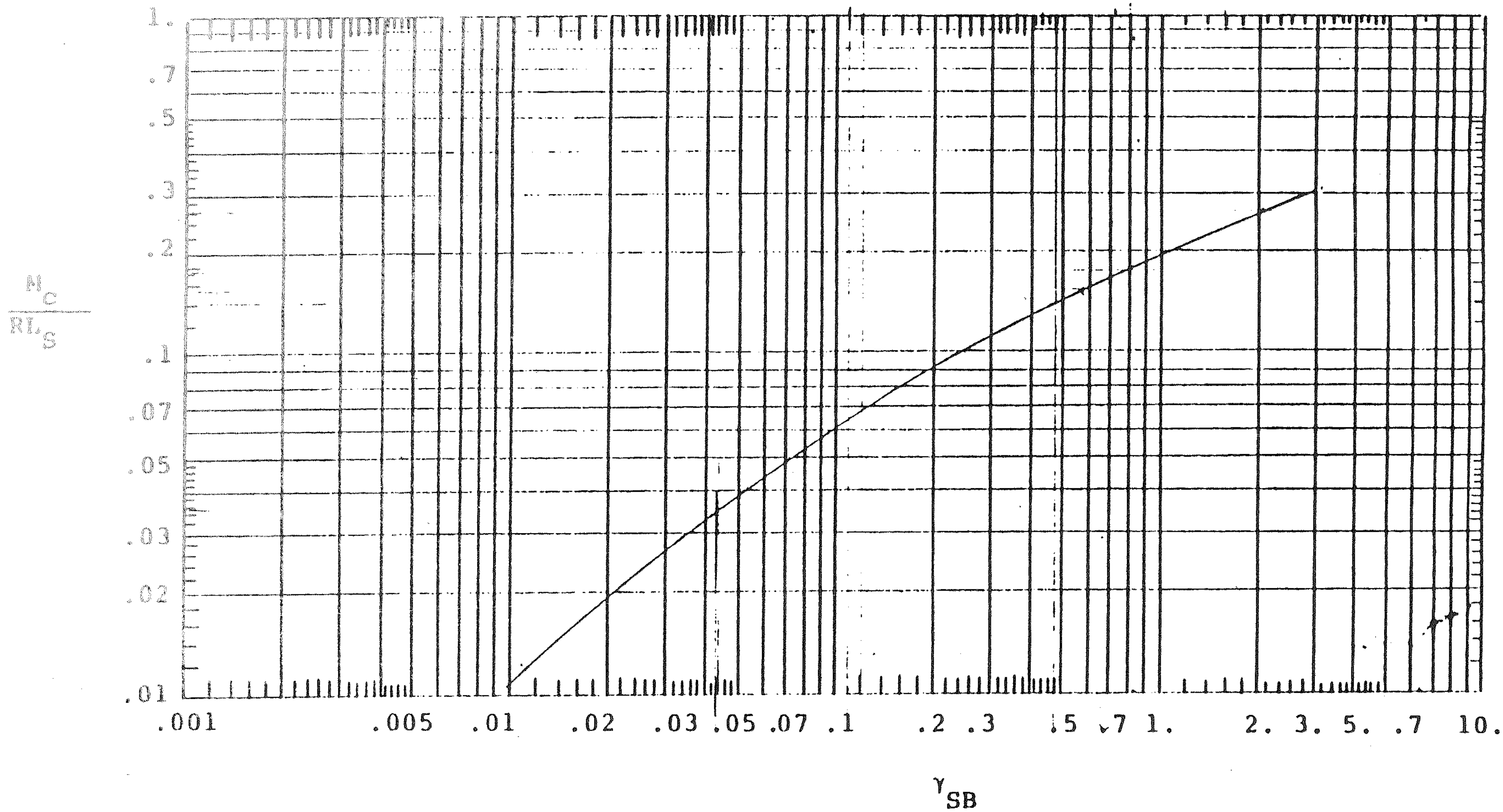


Figure 14 Stiffener Bending Moment Correction for Elastic Supports Coefficient, $\frac{M_C}{RL_S}$

And for the vehicle aligned perpendicular to the stiffeners, or, for the vehicle alinged with the stiffeners where:

$$L_b < 1.5\ s$$

the beam loading coefficient is:

$$\phi_4 = \frac{4}{\pi} \sin\left(\frac{\pi B_\ell}{L_b}\right)$$

The moment due to flexibility of the beam supports, M_c, is then calculated by applying the appropriate load factors using the following equation.

$$M_c = \left(\frac{M_c}{RL_s}\right) R_o b L_s\ \phi_4$$

The moment M_c due to multiple patch loads may be neglected.

The moment due to the dead weight of the plating and stiffener, M_d, is determined by calculating the midspan moment of a single span beam with fixed supports under that distributed load. The distributed load due to the weight of the plating and stiffener is calculated using the following equation.

$$w_d = \frac{1}{12{,}000} \left(w_s + \left(\frac{b}{12}\right) w_p\right)$$

The moment due to the dead weight of the plating and stiffener, M_d, is then calculated using the following equation.

$$M_d = \frac{\eta_z w_d L_s^2}{12}$$

The maximum bending moment in the stiffener, M_s, is the sum of these components, and is calculated using the following equation.

$$M_s = M_o + M_c + M_d$$

The bending stress is then calculated using one of the following equations depending on type of vehicle operations and deck function.

For the parking condition on a non-strength deck, the bending stress is calculated using the following equation.

$$f_{sb} = \frac{M_s}{SM_{min}}$$

For the parking condition on a strength deck, the bending stress is calculated using the following equation.

$$F_{sb} = \frac{M_s}{SM_{min}} + \sigma_{primary}$$

The value of the design primary stress ($\sigma_{primary}$) for the storm sea condition and its variation along the length of the vessel may be taken from the detail specifications for that vessel. The primary stress values to be used in the various loading conditions are as shown in Table V.

TABLE V - DESIGN PRIMARY STRESS VERSUS SEA CONDITIONS

	% of Design Primary
Storm Sea Parking	100
Moderate Sea Parking	50

In addition, it may be necessary to determine ($M_s/SM_{max} + \sigma_{primary}$) where high hull girder deck compressive stresses occur. Check the plate stiffener combination, for buckling as a column in accordance with Reference (b).

An estimate of the required section modulus, SM_{reqd}, can be determined as follows:

$$SM_{reqd} = M_s \; / \; (\sigma_{sb} - \sigma_{primary})$$

where the allowable bending stress, $_{sb}$, is as defined in Section 130-3-o.

Maximum Shear Force and Stress

The loading condition which produces the maximum shear force is when the patch load is placed adjacent to the stiffener support directly over the stiffener. This maximum shear force is the summation of (1) the shear due to the live load, V_o, (assuming the stiffener as a continuous beam on rigid supports) and (2) the shear due to the dead weight of the plating and stiffener, V_d.

The load applied to the stiffener is a single or dual patch load of length, A', (along the stiffener), width B', (perpendicular to the stiffener), magnitude, P, and dual patch spacing, b", as described in this Section.

To account for the distribution effects of the patch width, B', the patch width load factor, ϕ_1, calculate B'/b and use Figure 12.

The shear due to the live load over rigid supports, V_o, is determined by using the influence line coefficient for the shear at the support of a continuous beam over equally spaced rigid supports for a patch load adjacent to the support are presented in Figures 11.1 and 11.2.

The shear due to the live load over rigid supports, V_o, is then calculated by applying the appropriate load factors using the following equation.

$$V_o = \left(\frac{R}{P}\right) P \; \phi_1$$

For multiple patch loads

$$V_o = \phi_1 \sum_{i=1} (\gamma P)_i \qquad \text{where } \gamma = (R/P)$$

To account for the combined effects of patches, the linear superposition method described in 130-3-k.1 will be used.

The shear due to the dead weight of the plating and stiffener, V_d, is determined by using the distributed load due to the weight of the plating and stiffener, and then calculating the shear at the support of a single span beam under the distributed load. The shear due to the dead weight of the plating and stiffener, V_d, is calculated using the following equation.

$$V_d = \frac{\eta_z w_d L_s}{2}$$

The maximum shear force in the stiffener, V_s, is the sum of these components and is calculated using the following equation.

$$V_s = V_o + V_d$$

The shear stress is then calculated using the following equation.

$$f_{sv} = \frac{V_s}{A_s}$$

An estimate of the required shear area, $A_{s_{REQD}}$, can be determined as follows:

$$A_{s_{REQD}} = \frac{V_s}{\sigma_{sv}}$$

Where the allowable shear stress, σ_{sv}, is as defined in Section 130-3-o.

Appendix E is an abridged description of the stiffener analysis methods and criteria for both bending and shear for easier reference. Appendix F contains standard calculation sheets.

130-3-k.2 Irregular Structural Scantlings

Where the structural arrangement of the deck scantlings cannot be characterized as equally spaced or sized stiffeners or beams due to unique design considerations,, a grid analysis of the deck may be performed using an accepted Finite Element Program. Figure 2 shows a typical grid model.

The modeling of the deck is left to the discretion of the engineer, but some suggestions follow:

Plating - Generally it is not necessary to model the plate. Panels are usually regular in size or a given set of panels will control the design. If this is not the case, and analysis is necessary, the analysis must allow for the membrane as well as the bending behavior of the plating.

Stiffeners - Beam elements are adequate. The effective plate should be included in the beam properties. Generally, five spans are sufficient to achieve the effects of continuous beams.

Beams and Girders - Beam elements are adequate. The effective plate should be included in the beam properties. Generally three spans are sufficient to achieve the continuity effects for these beams.

Stanchions and Bulkheads - Generally it is not necessary to model the supports other than rigid. If sway of the deck is critical, then modeling may be necessary.

Loading - (1) Live Load.

If only the stiffeners will be designed or analyzed based on the results obtained, then only one wheel load need be applied. If the beams and girders will be designed or analyzed based on the results obtained, then the other wheels should be included and the critical loading conditions for these members should be used. Only those loading conditions pertaining to the stiffener design or analysis will be discussed.

Two loading conditions will be needed, (1) to determine the maximum bending moment and stress and (2) to determine the maximum shear force and stress. As described before, the maximum shear force occurs when the load is adjacent to the support.

To properly distribute the loads so that the effects of patch width distribution, plating distribution, multiple patches, and beam flexibility are accurately modeled, the wheel load should be distributed between the three stiffeners nearest midspan of the beams. The loads are distributed using the patch width load factor, ϕ_1, the plating distribution load factor, ϕ_2, as described in Section 130-3-k.1. The loads applied to each stiffener should be distributed along the length of the stiffeners for a distance equal to the patch length, A'. Figure 15 shows the proposed distribution of the wheel load, where L_0 is the load on the stiffener in question, L_{1l} is the load on the stiffener directly to the left, and L_{1r} is the load directly to the right.

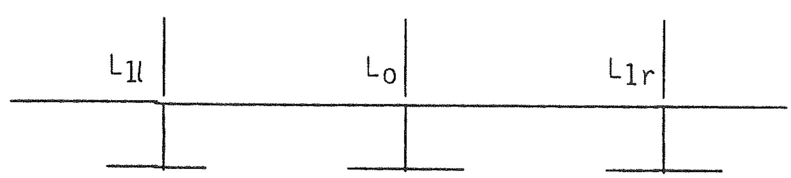


FIGURE 15

The magnitude of these loads should be shown in Table VI.

LOADING FOR	BENDING MOMENT	SHEAR FORCE
L_0	$P\ \phi_1\phi_2$	$P\phi_1$
L_{1l}	$1/2P\ \phi_1\ (1-\phi_2)$	$1/2P\ \phi_1$
L_{1r}	$1/2P\ \phi_1\ (1-\phi_2)$	$1/2P\ \phi_1$

TABLE VI - F.E.M. MODEL LOADINGS

Note that the patch width load fator, $\emptyset_1$, is different for L_0 than for L_{1l} or L_{1r}. For L_0 the $\emptyset_1$, curve marked i = 0 should be used and for L_{1l} and L_{1r} the $\emptyset_1$, curve marked i = 1 should be used.

(2) Dead Load.

The dead load of the plating and stiffeners can be modeled as distributed loads of magnitude, w_d, applied to each of the stiffeners over the entire length.

(3) Primary Stress.

The effects due to the hull girder bending (primary stress) can be either added as an additional stress or as an axial load in the longitudinal direction.

130-3-1. Beams, Girders and Stanchions

Beams, girders, and stanchions supporting vehicle handling decks must be designed to withstand the maximum bending, shear, or compressive stress induced by the vehicle loads or any other loading requirements of the deck. It is essential that the most critical loading condition for each of these members be determined, since the vehicle could be at almost any location on the deck.

Longitudinal beams or girders must also be designed or analyzed with primary stress considered, if applicable.

For both beams and girders, any acceptable linear analysis method such as moment distribution may be utilized. The method chosen should be based on the structural geometry and the engineers discretion.

Stanchions provide intermediate support for beams or girders where their spans would otherwise be excessive. Likewise, where deck stiffeners, beams, or girders are supported by bulkheads, the vertical stiffener under the beams may be considered as a column using the appropriate plate-stiffener combination.

The maximum reaction into a stanchion or bulkhead support must be obtained. Reference (b) should be used to determine the adequacy or required size of the stanchion or bulkhead support.

Parametric Vehicle Load Influence Lines

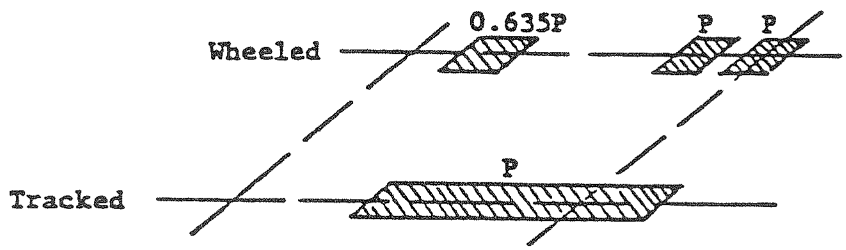
A series of influence lines for specific representative vehicles have been developed for a range of possible span lengths from 2 to 21 feet. These lines, presented in Appendix C, where developed through the process of superimposing the individual patch loads that make up the nominal vehicle loading. These vehicle influence lines are given in terms of a single vehicle reference point which is normally taken as the center of the front-left tire patch. The curves are presented with a relative ordinate axis for clarity. The principal use of these curves is for determining critical moments and load positions of nominal vehicle loading.

These influence lines are presented in Appendix B for the following four representative vehicles which may be used for design purposes as generic vehicle types.

VEHICLE
M54 5 Ton Cargo Trunk
M715 1 1/4 Ton Truck
M88 Tracked Vehicle
6000-lb Fork Lift

Influence lines for bending moment at support and midspan are presented for each vehicle (nominal loading) for both longitudinal and transverse alingment of the vehicle. The maximum bending moment is calculated at the vehicle position which causes the maximum influence ordinate.

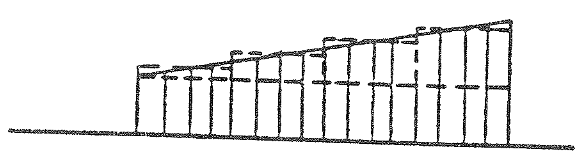
Influence lines for a given vehicle are based on the ratio of load between the vehicle patches under the nominal loading condition (see figure below).



Example vehicle load patches

The influence lines presented here are based on the nominal load ratio (the load ratio when the truck and ship are motionless). The load ratio varies as the ship moves. For a vehicle loading with a load patch ratio differing from that of the nominal vehicle load, superposition is used to describe the loading in terms of the nominal load ratio and a single patch load.

The load patch for a tracked vehicle is modeled as a rectangular patch with a superimposed triangular patch load. The triangle loading is approximated by several single patch loads (see figure below).



Use of the vehicle influence lines for determining critical stiffener moment is illustrated in Appendix G.

For multiple vehicles, the vehicle influence ordinates are calculated from the superposition of single patch influence lines ordinates.

PART VI: DESIGN CRITERIA

130-3-m. General

The yield stress, F_y, and allowable working stress, F_b, for each material are as provided in the ship specifications, corresponding to Section 100 (Section 9110-0 for older ships) of Reference (f).

130-3-n. Plating

The allowable stress levels for the plating are a function of the probability of the occurence of the loading. In deck parking areas, when severe ship motions (storm sea) are assumed, the plating allowable stress is taken as the welded yield strength of the material, that is,

$$\sigma_p = F_y$$

For moderate sea parking, (the most common and frequent load magnitudes), the allowable stress is the allowable working strength of the material or

$$\sigma_p = F_b$$

The calculated stress in the plating, f_p, must be less than or equal to the allowable stress or

$$f_p \leq \sigma_p$$

130-3-o. Stiffeners

The allowable bending stress level for the stiffeners is the allowable working strength of the material or

$$\sigma_{sb} = F_b$$

The calculated bending stress in the stiffener, f_{sb}, must be less than or equal to the allowable bending stress or

$$f_{sb} \leq \sigma_{sb}$$

The allowable shear stress level for the stiffeners is sixty percent (60%) of the allowable working strength of the material or

$$\sigma_{sv} = 0.6\ F_b$$

The calculated shear stress in the stiffener, f_{sv}, must be less than or equal to the allowable shear stress or

$$f_{sv} \leq \sigma_{sv}$$

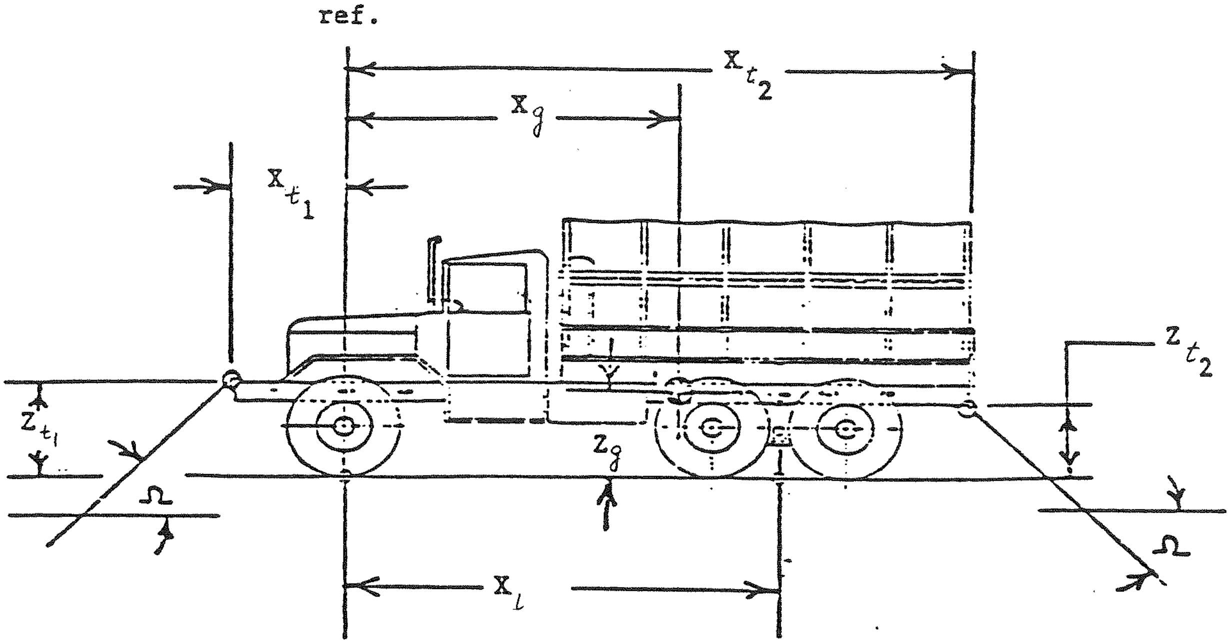
130-3-p. Beams, Girders and Stanchions

The allowable stress levels for beams, girders and stanhions are as per the design criteria in the ship specifications, corresponding to Section 100 (Section 9110-0 for older ships) or Reference (f) or Reference (b).

APPENDIX A

DATA SHEETS FOR EXISTING VEHICLES & TIRE LOAD PATCHES

M54 5-TON CARGO TRUCK



X_l = 179"

X_g = 135.9"

X_{t_1} = 37"

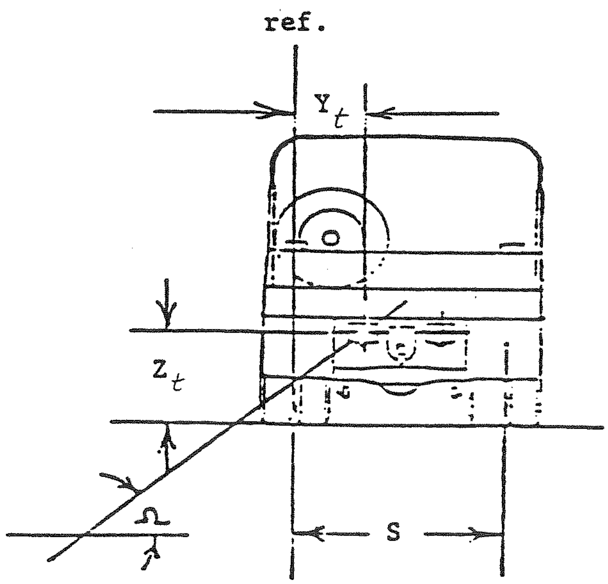
X_{t_2} = 261"

Z_g = 40.5"

Z_{t_1} = 44"

Z_{t_2} = 38"

Ω = 45°



S = 72"

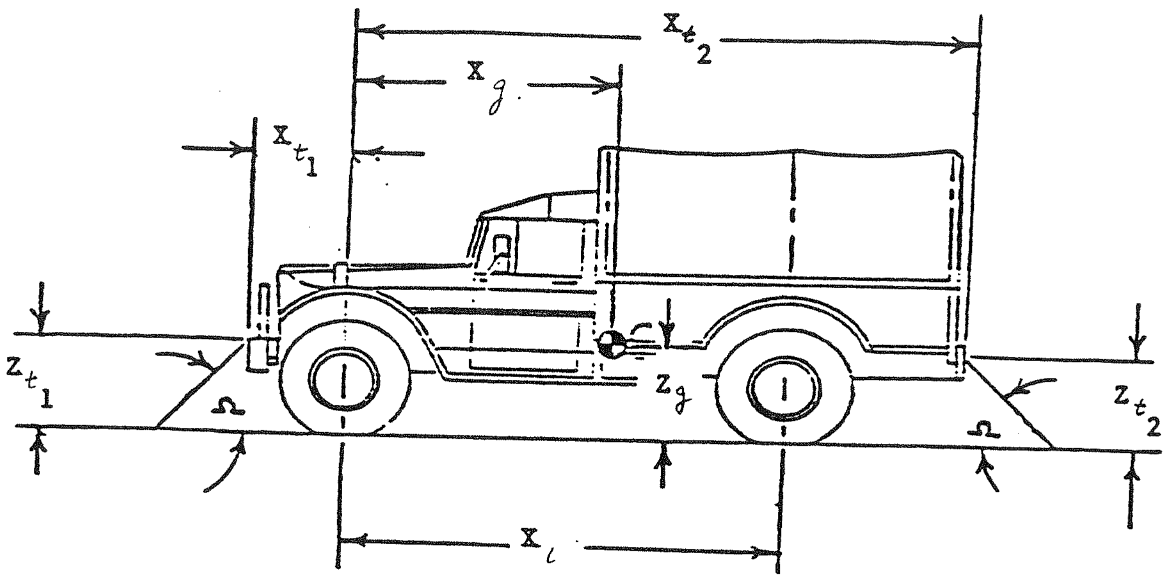
Z_t = 40"

Y_t = -20"

Ω = 45°

Nominal vehicle weight = 41100 lbs

M715 1-¼ TON TRUCK



X_l = 129"

X_g = 78"

X_{t_1} = 29..5"

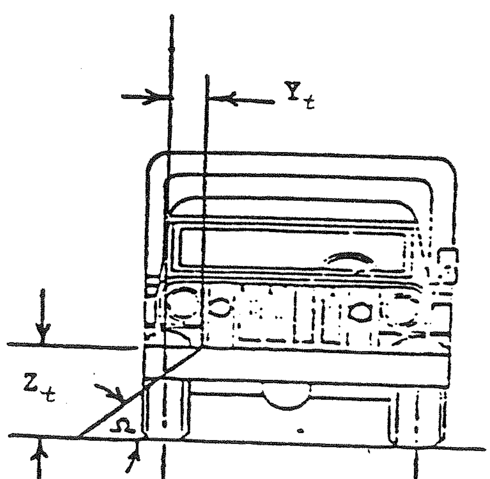
X_{t_2} = 180"

Z_g = 31"

Z_{t_1} = 30"

Z_{t_2} = 30"

Ω = 45°



S = 67"

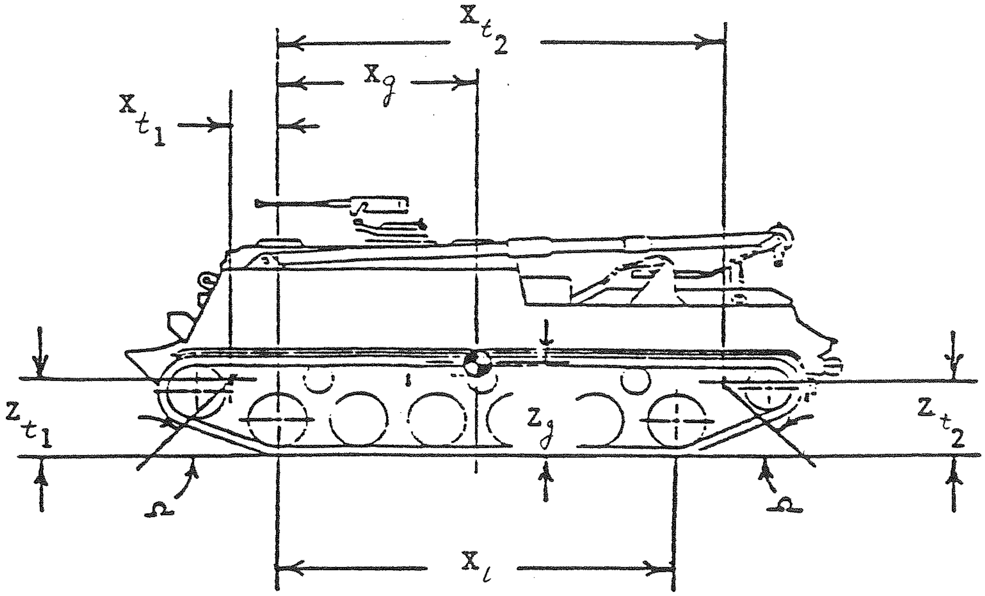
Z_t = 30"

Y_t = -12"

Ω = 45°

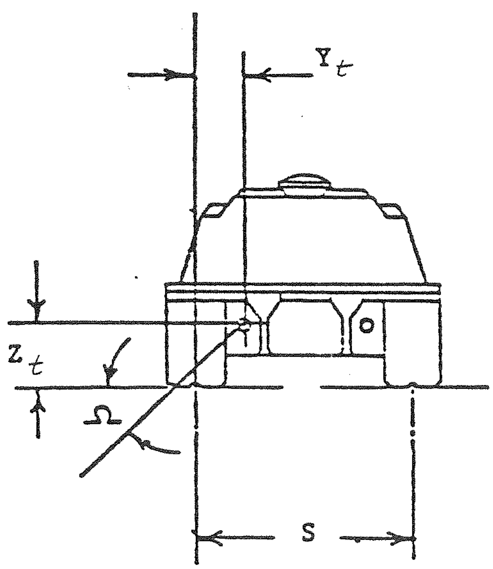
Nominal vehicle weight = 8950 lbs

M88 TRACKED VEHICLE



X_l = 181.5"
X_g = 91"
X_{t_1} = 23"
X_{t_2} = 202"

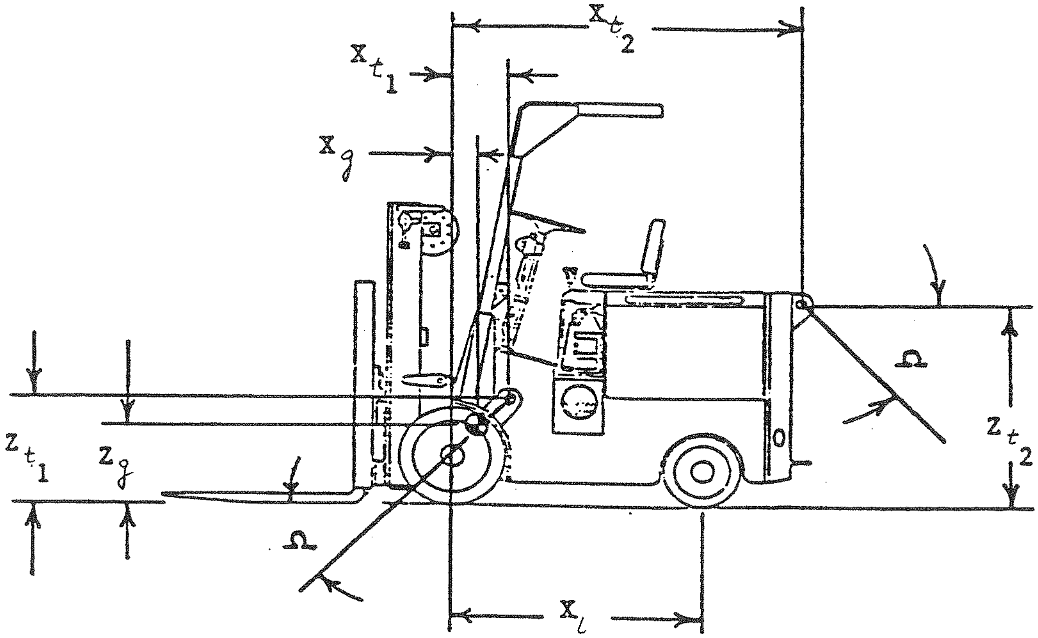
Z_g = 45"
Z_{t_1} = 37"
Z_{t_2} = 37"
Ω = 45°



S = 107"
Z_t = 37"
Y_t =-23"
Ω = 45°

Nominal vehicle weight = 112000 lbs

6000 lb FORK LIFT



X_l = 48"

X_g = 5" (fully loaded)

X_g = 18" (unloaded)

X_{t_1} = -12"

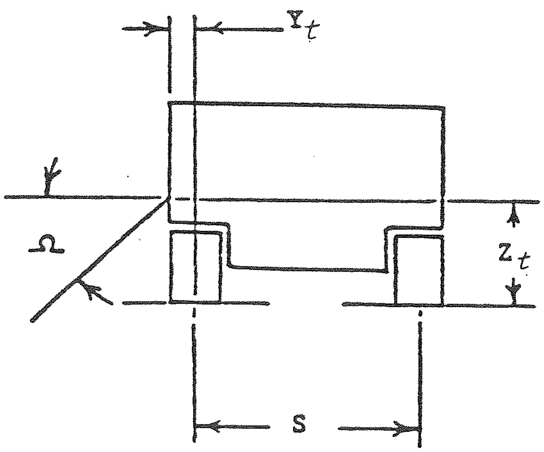
X_{t_2} = 74"

Z_g = 20" (assumed)

Z_{t_1} = 21"

Z_{t_2} = 41"

Ω = 45°



S = 35"

Z_t = 21"

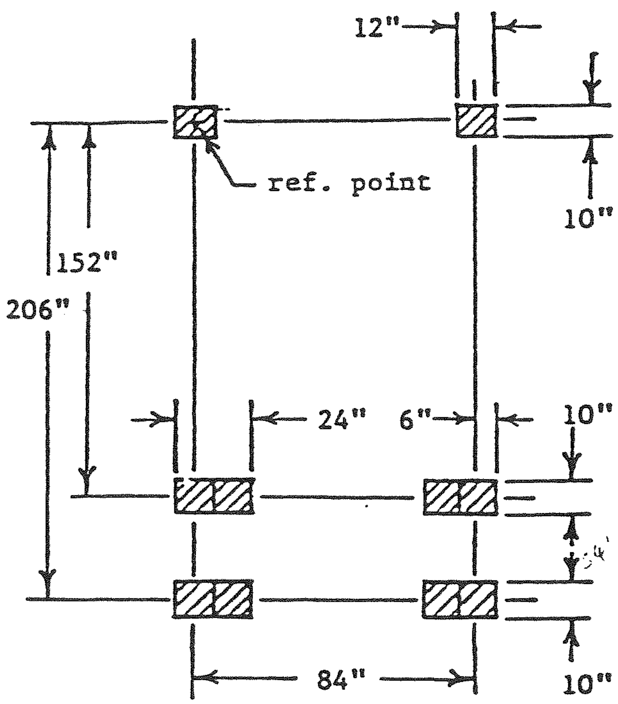
Y_t = 4"

Ω = 45°

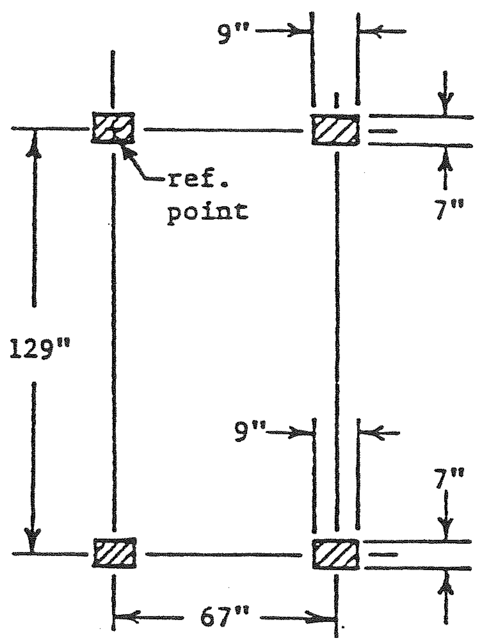
Nominal vehicle weight = 17500 lbs (fully loaded)
= 11500 lbs (unloaded)

REPRESENTATIVE VEHICLE TIRE LOAD PATCHES

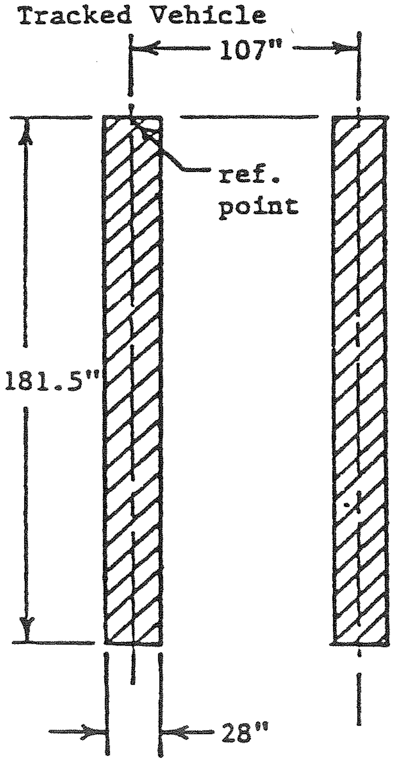
M54 5-ton Truck



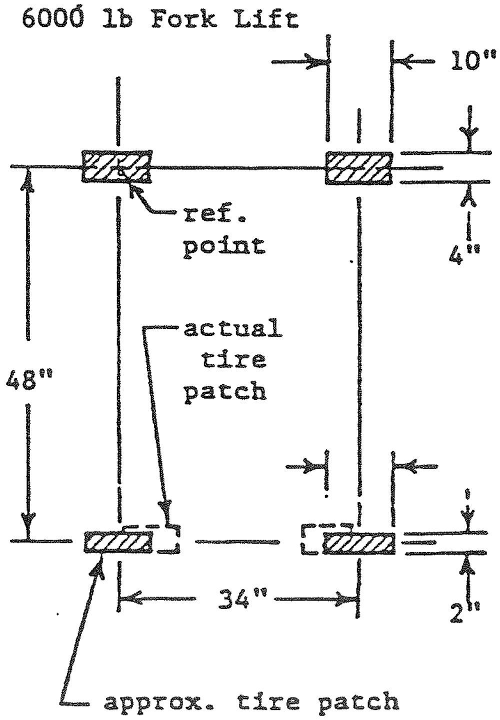
M715 1-¼ ton Truck



M88 Tracked Vehicle



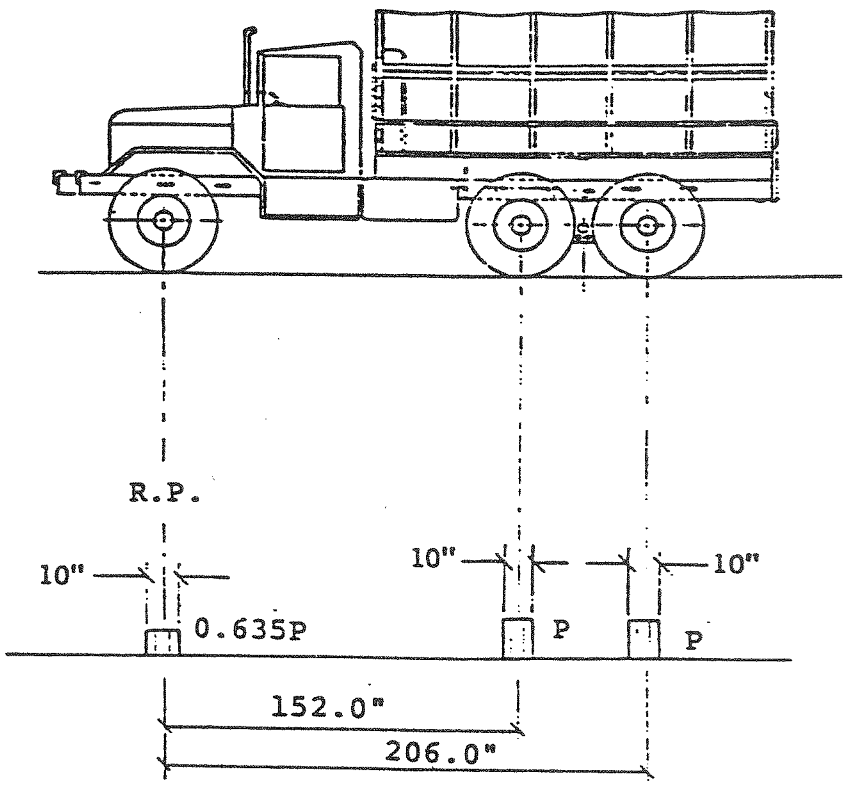
6000 lb Fork Lift



APPENDIX B

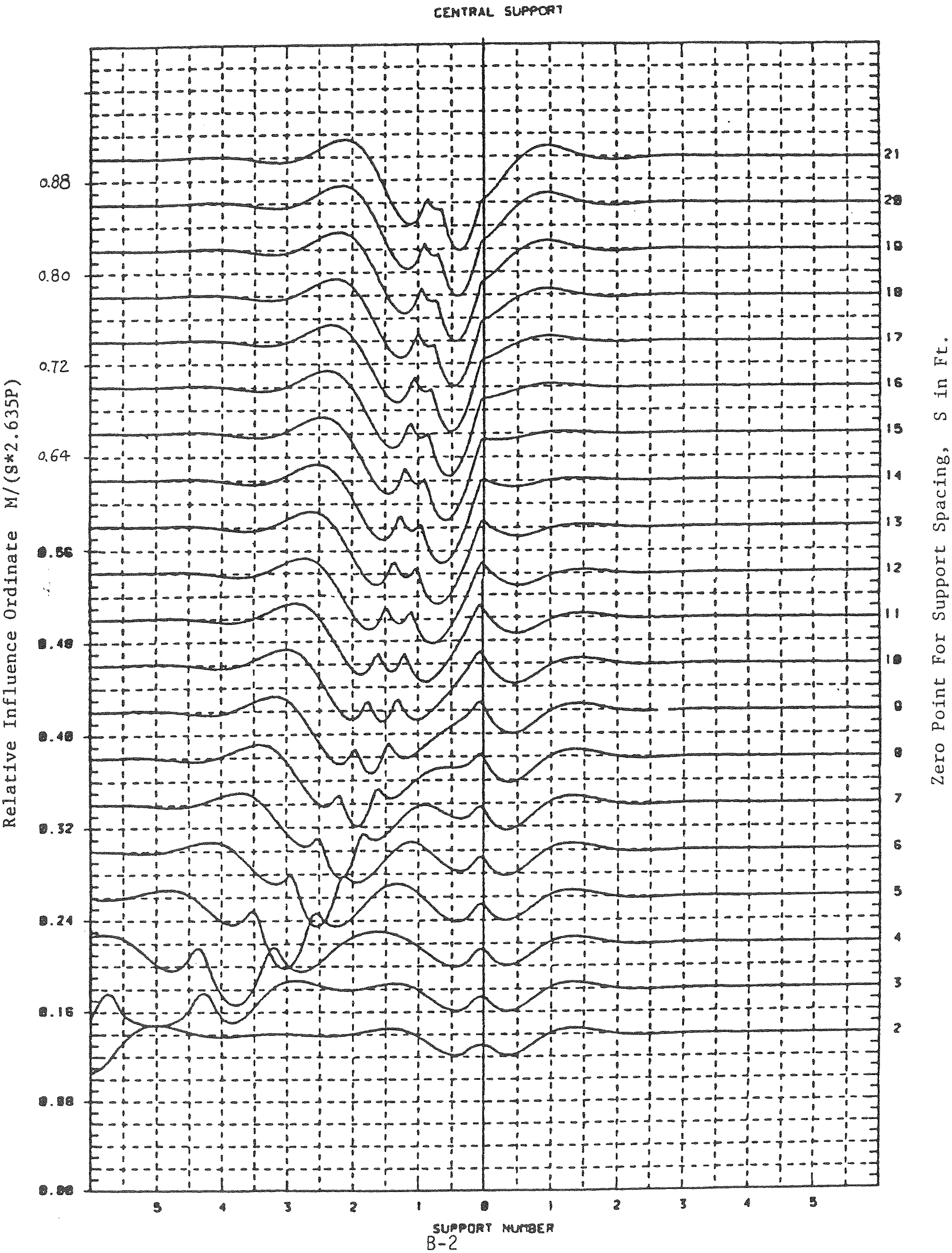
NOMINAL VEHICLE INFLUENCE LINES FOR REPRESENTATIVE VEHICLES

M54 5-TON TRUCK (LONGITUDINAL)

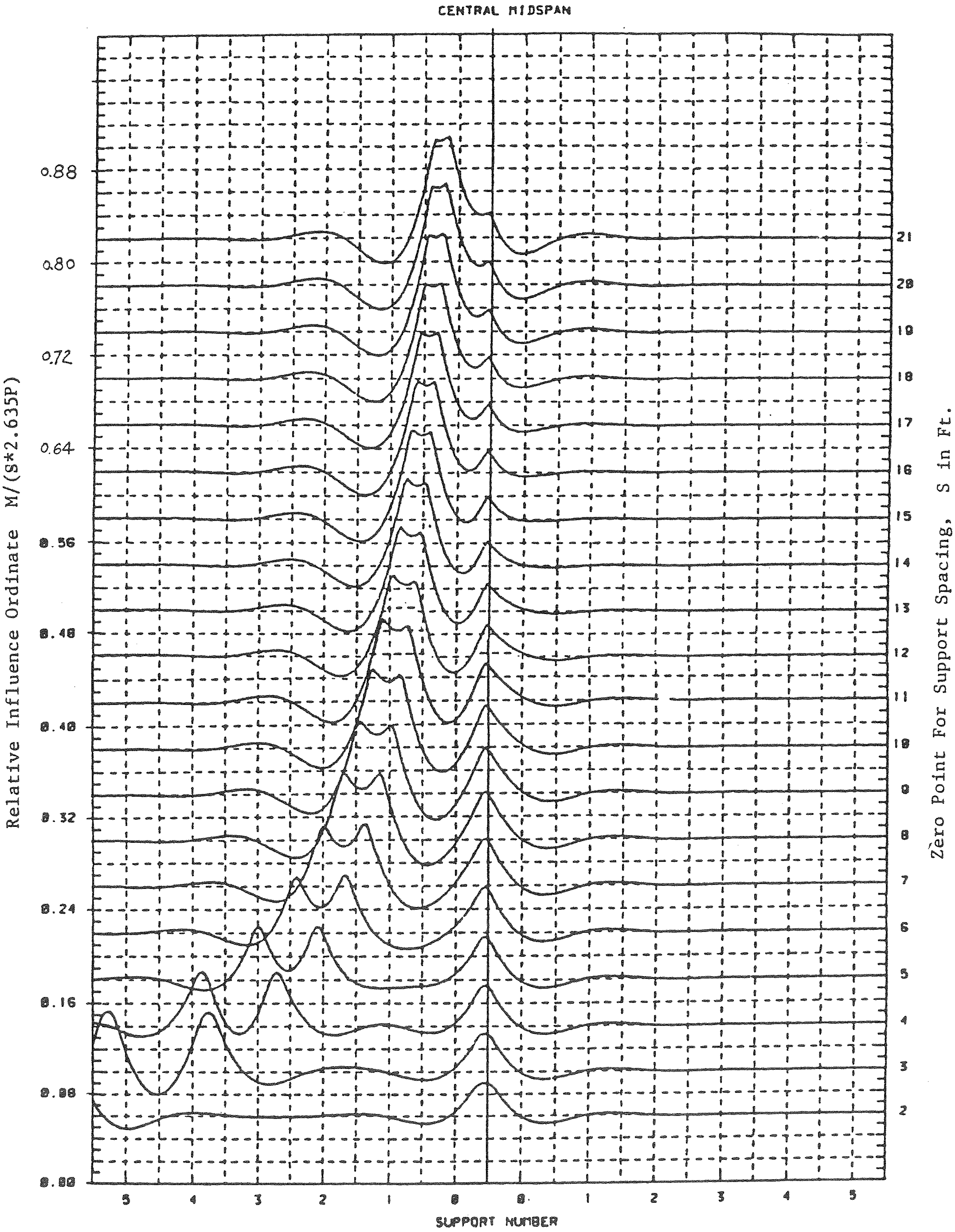


P= 7800 lbs. (nominal)

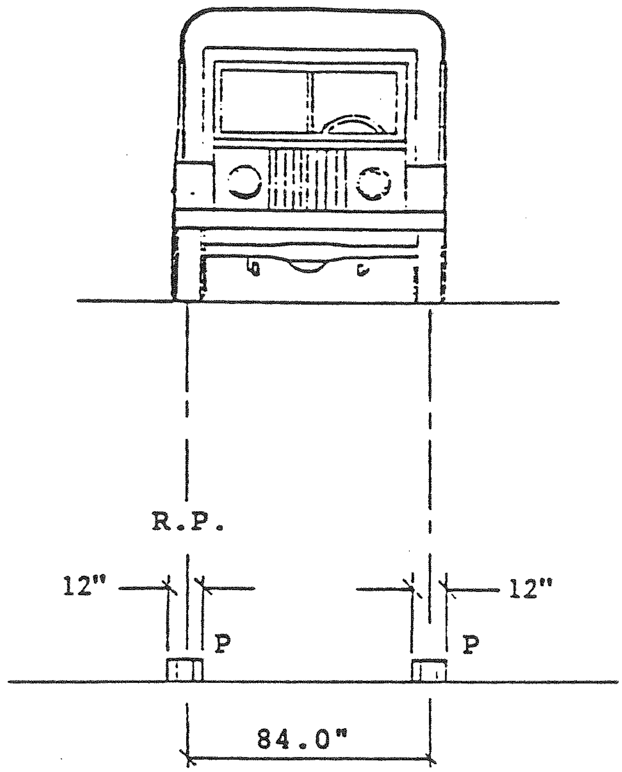
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR M54 5-TON TRUCK (LONG'L)



VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN FOR M54 5-TON TRUCK (LONG'L)

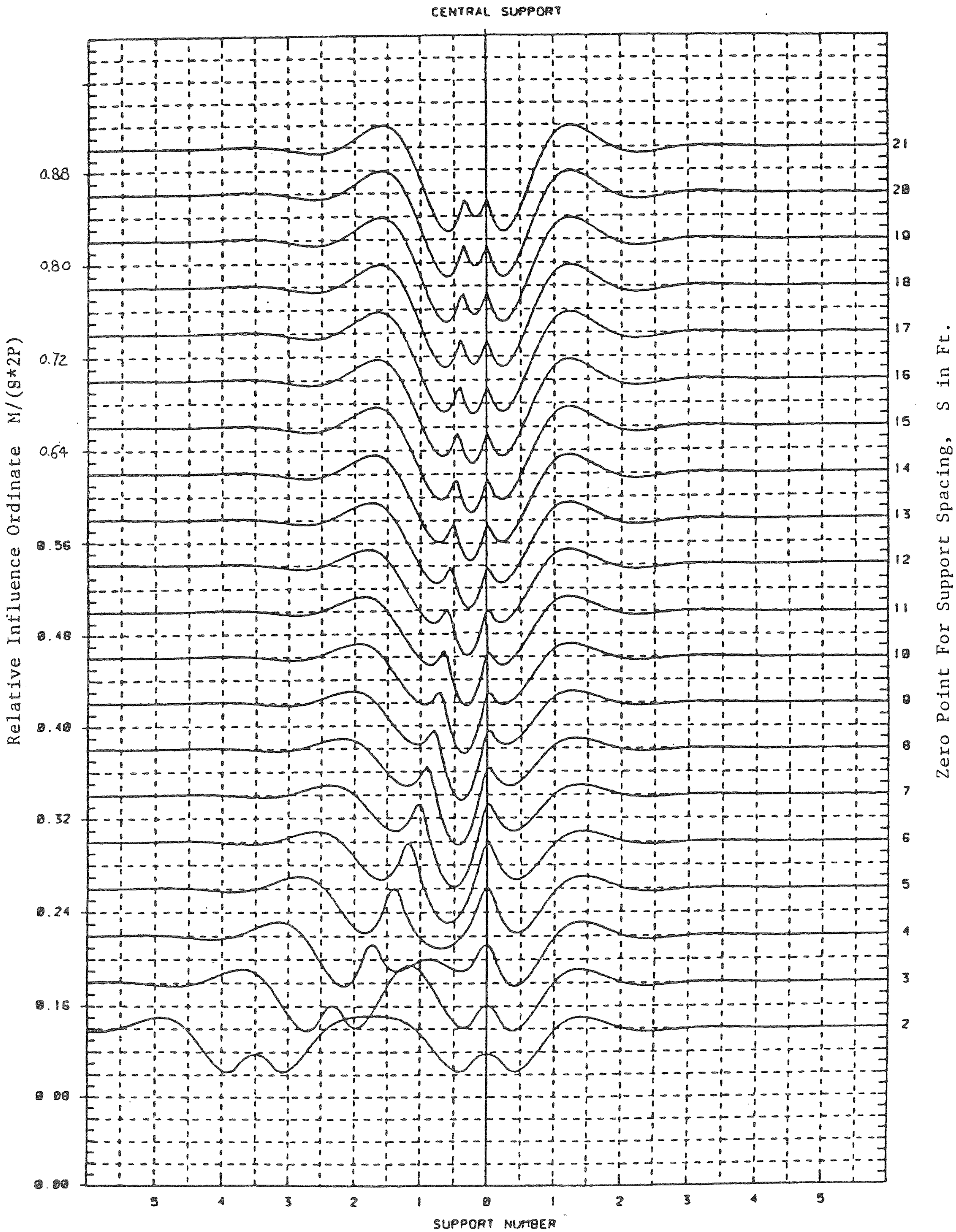


M54 5-TON TRUCK (TRANSVERSE-FRONT)

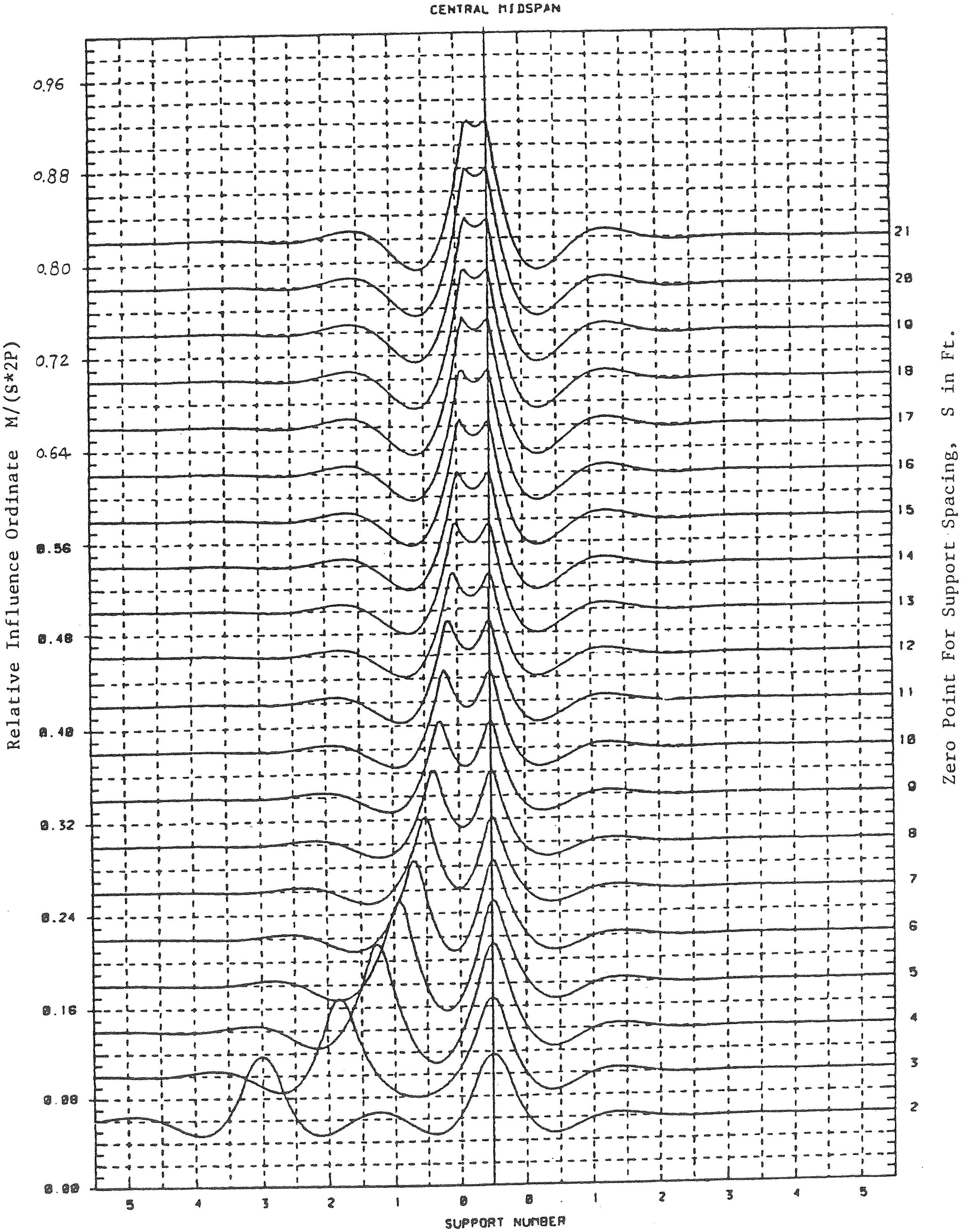


P=4950 lbs. (NOMINAL)

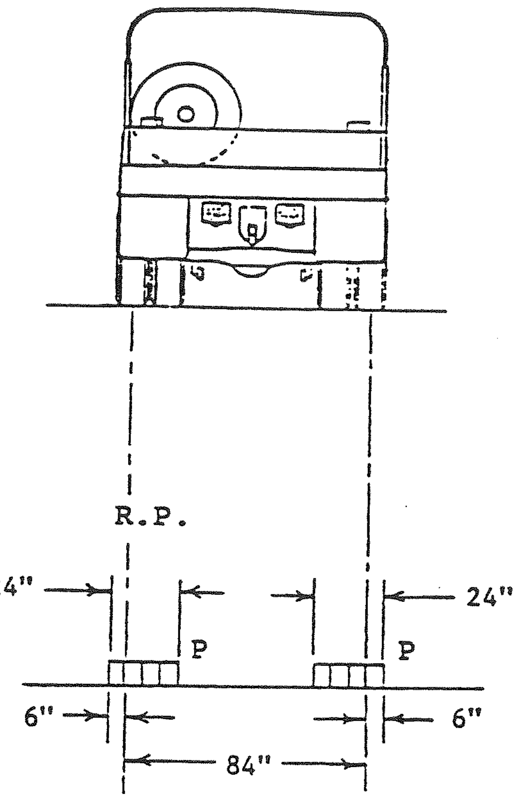
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR M54 5-TON TRUCK (TRANS.) (FRONT WHEEL)



VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN FOR M54 5-TON TRUCK (TRANS.) (FRONT WHEEL)

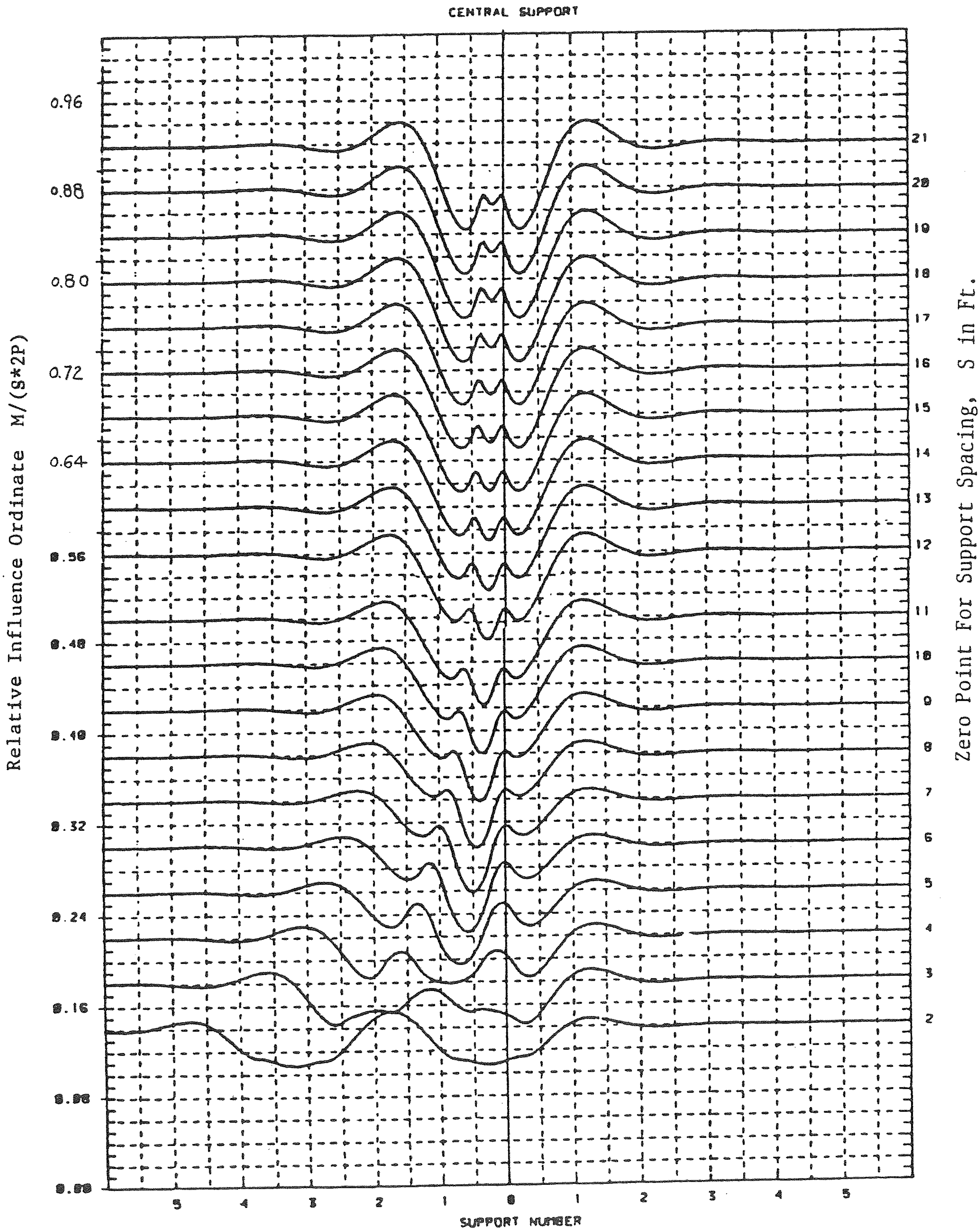


M54 5-TON TRUCK (TRANSVERSE-REAR)

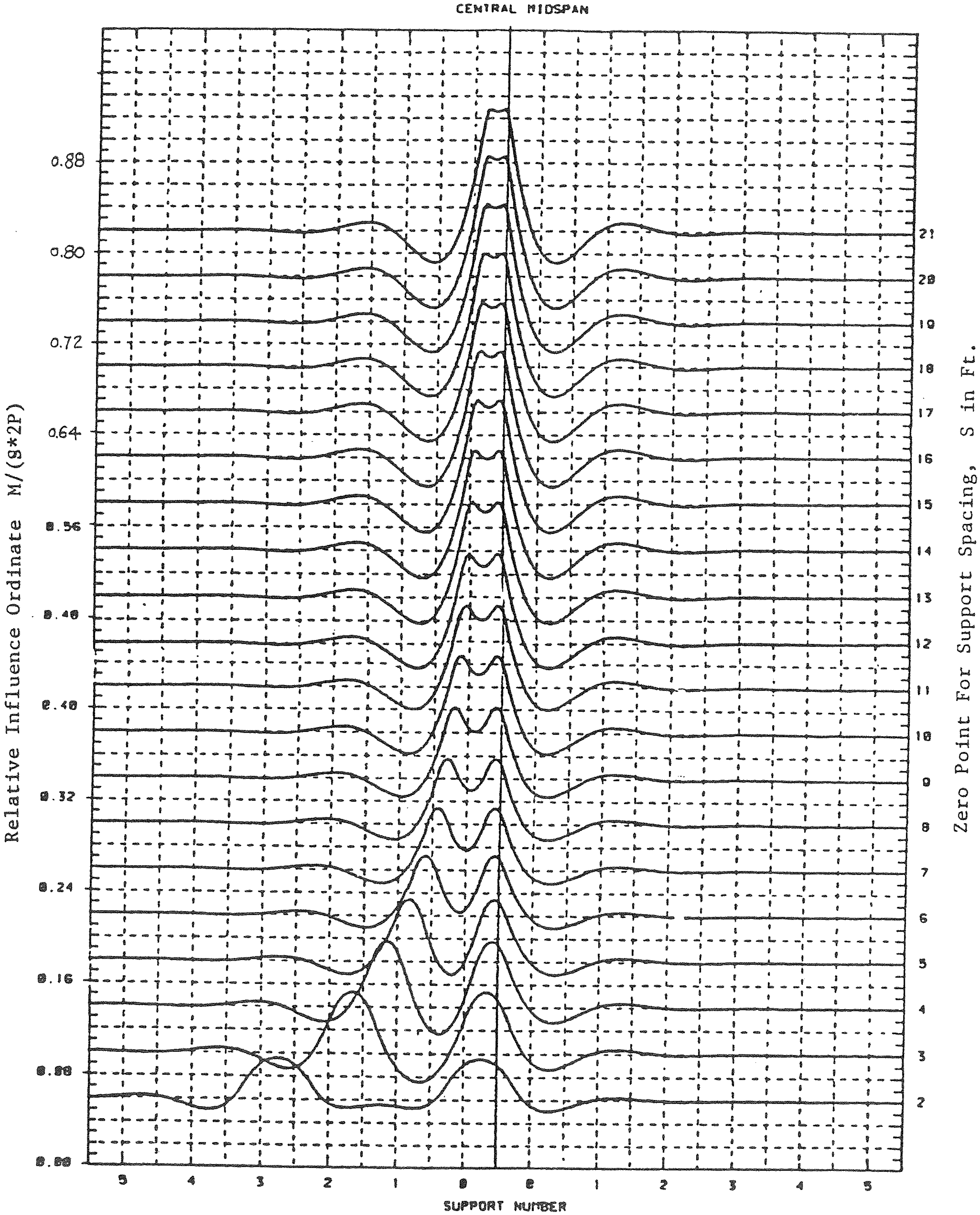


P=7800 lbs. (NOMINAL)

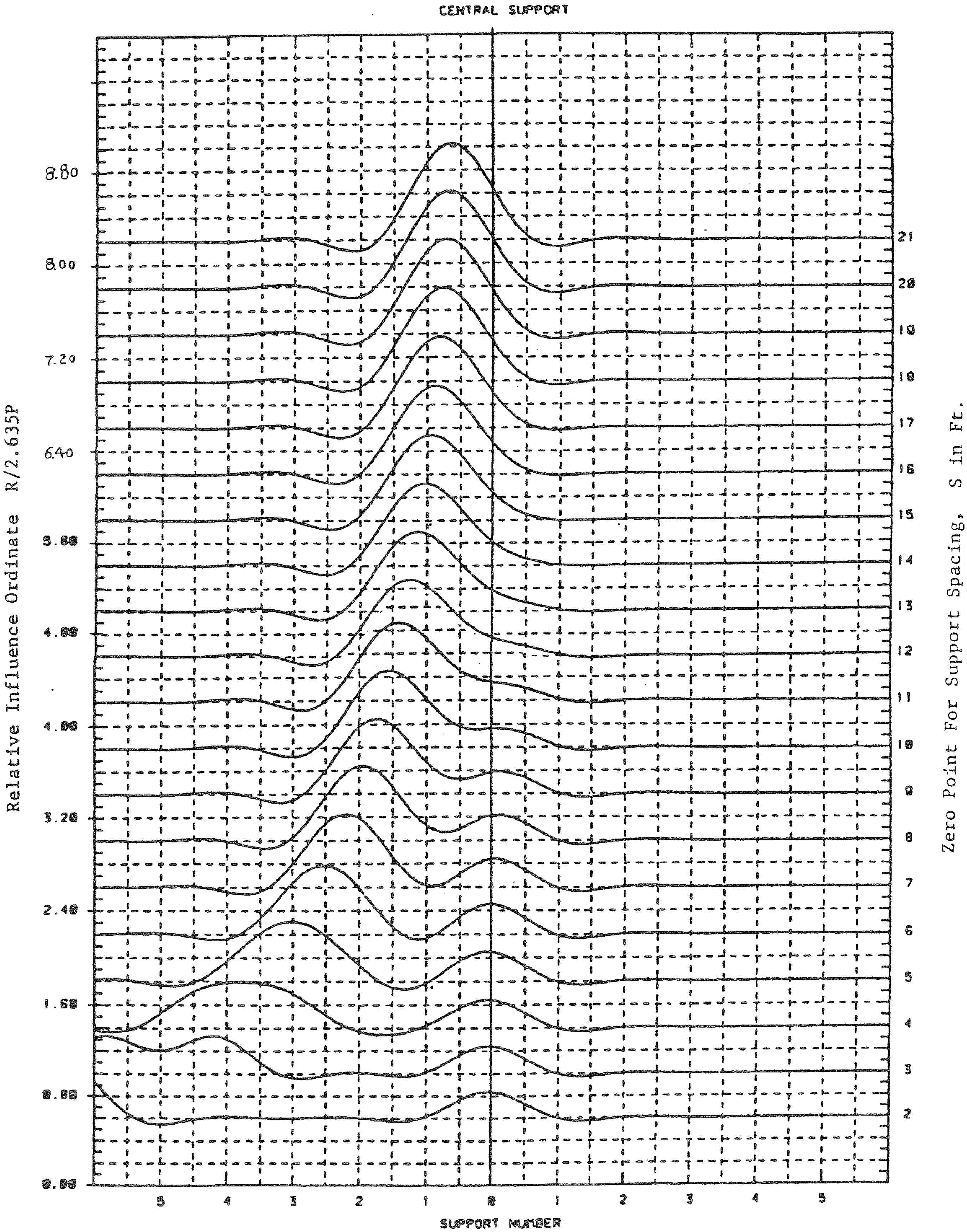
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR M54 5-TON TRUCK (TRANS - REAR WHEELS)



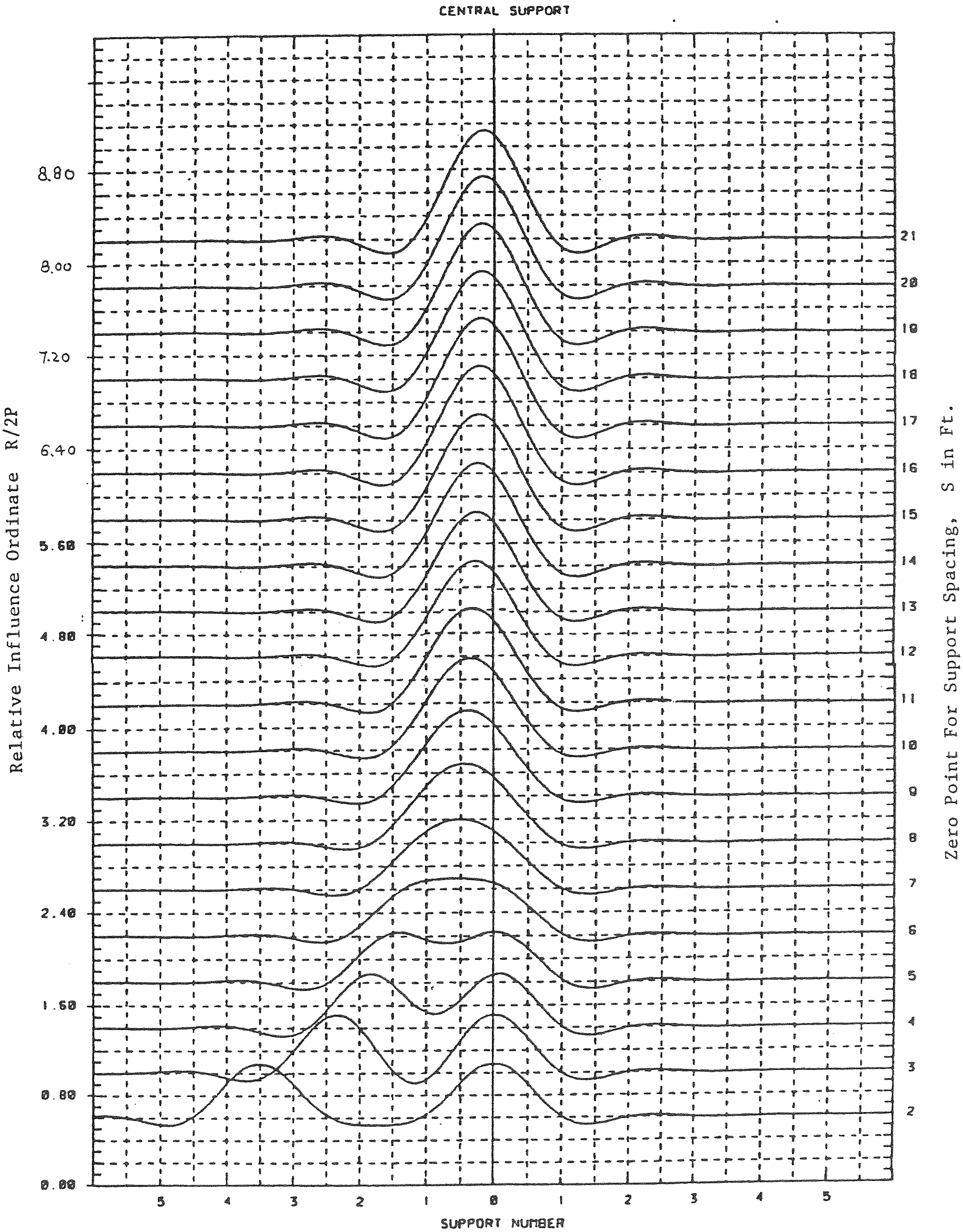
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN
FOR M54 5-TON TRUCK (TRANS - REAR WHEELS)



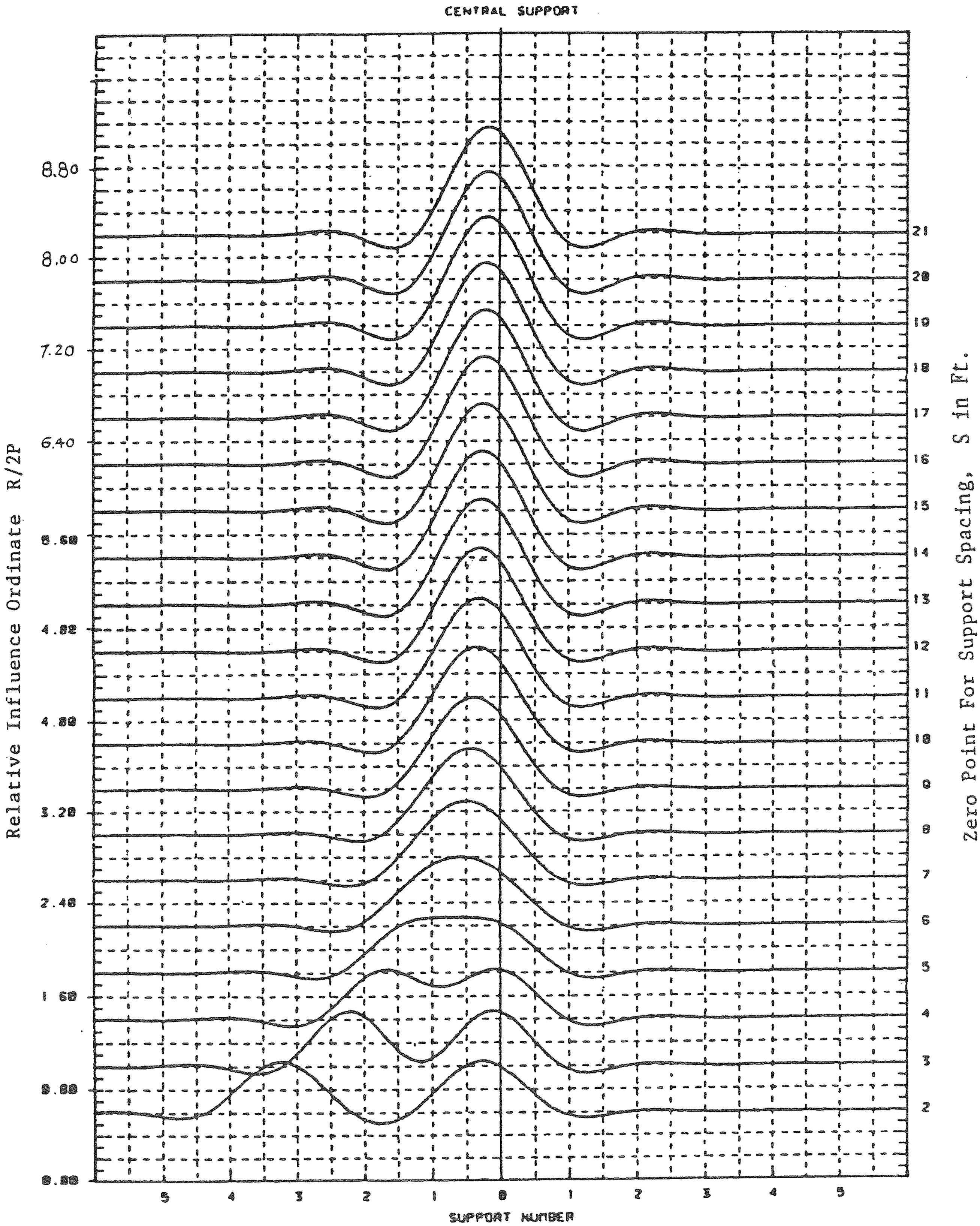
VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR M54 5-TON TRUCK (LONG'L)



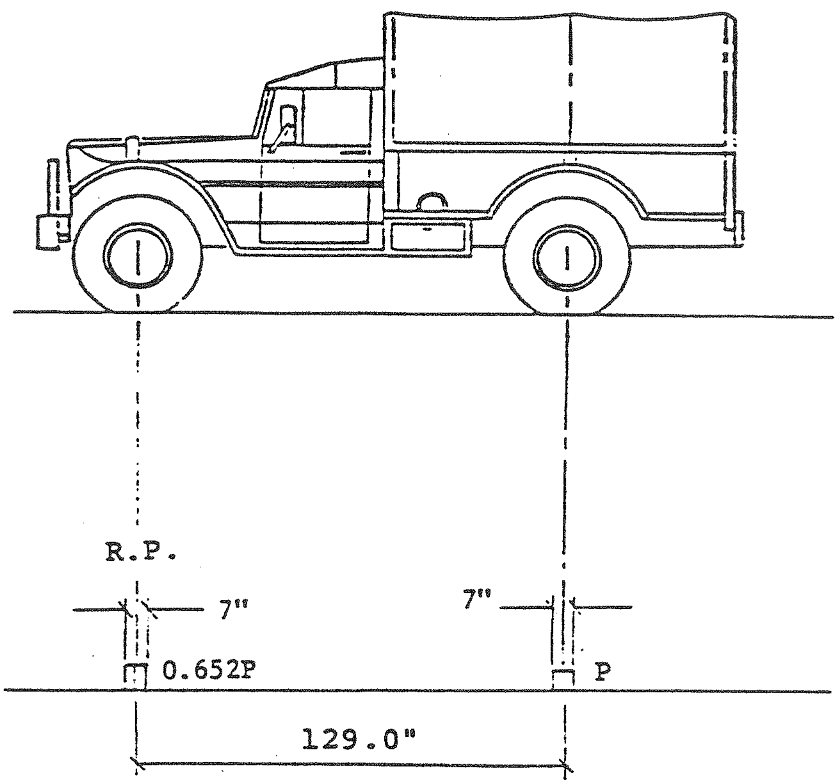
VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT
FOR M54 5-TON TRUCK (TRANS.) (FRONT WHEEL)



VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR M54 5-TON TRUCK (TRANS - REAR WHEELS)

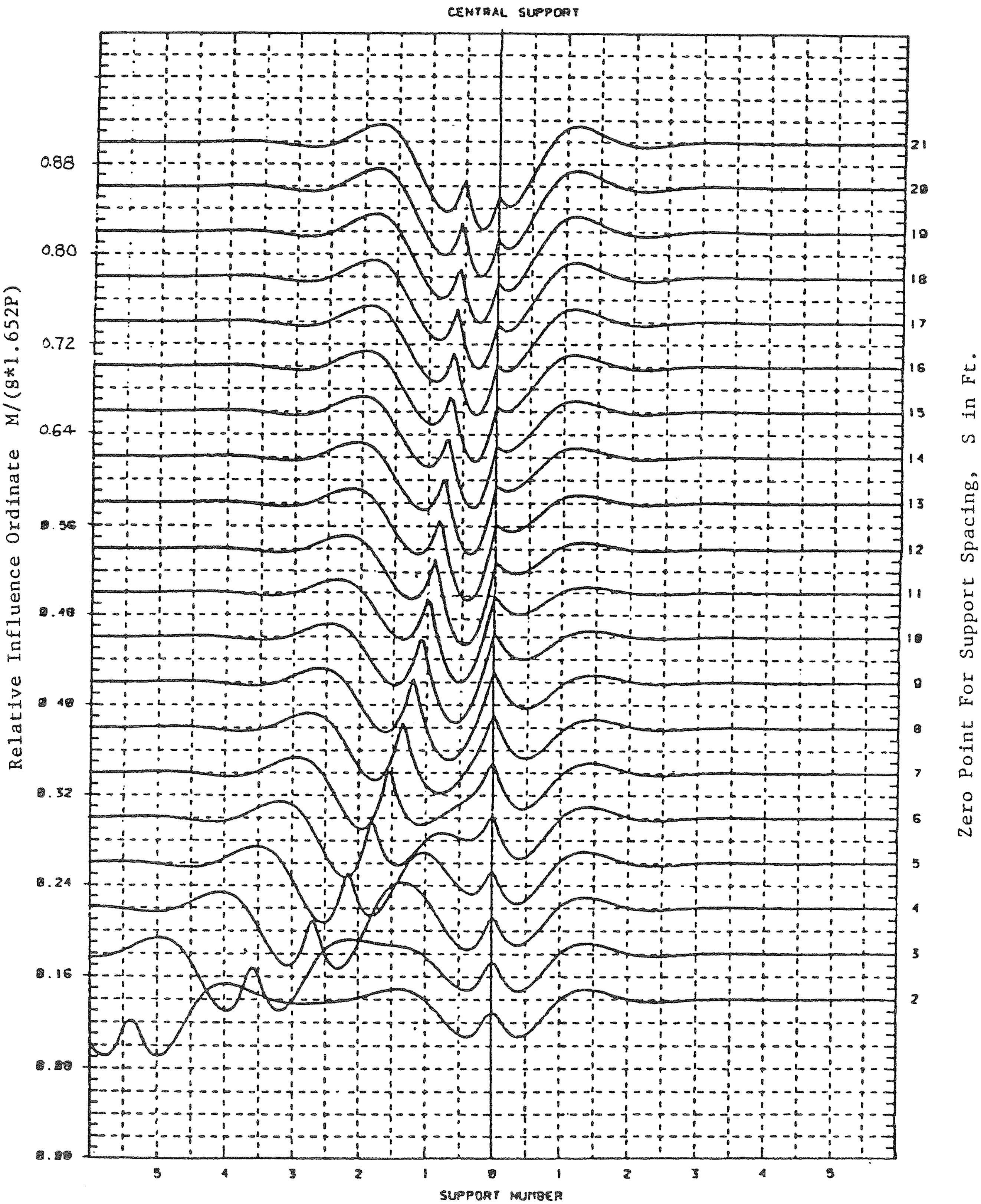


M715 1-1/4 TON TRUCK (LONGITUDINAL)

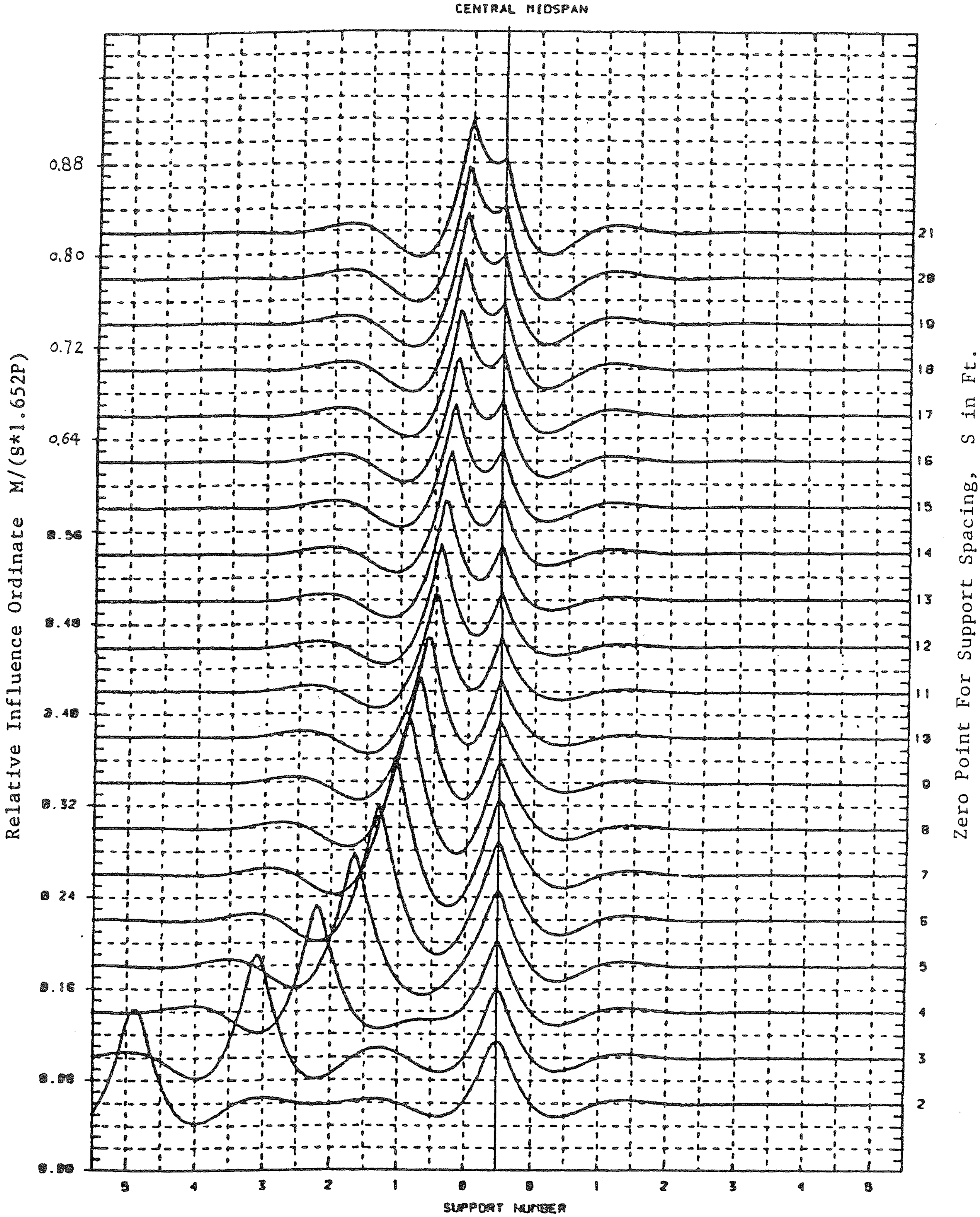


P=2709 lbs. (nominal)

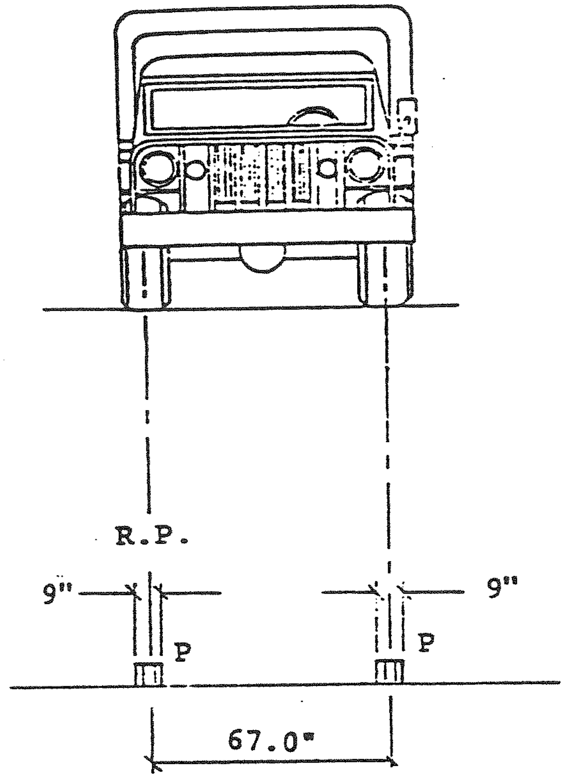
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR M715 1-1/4 TON TRUCK (LONG'L)



VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN
FOR M715 1-1/4 TON TRUCK (LONG'L)

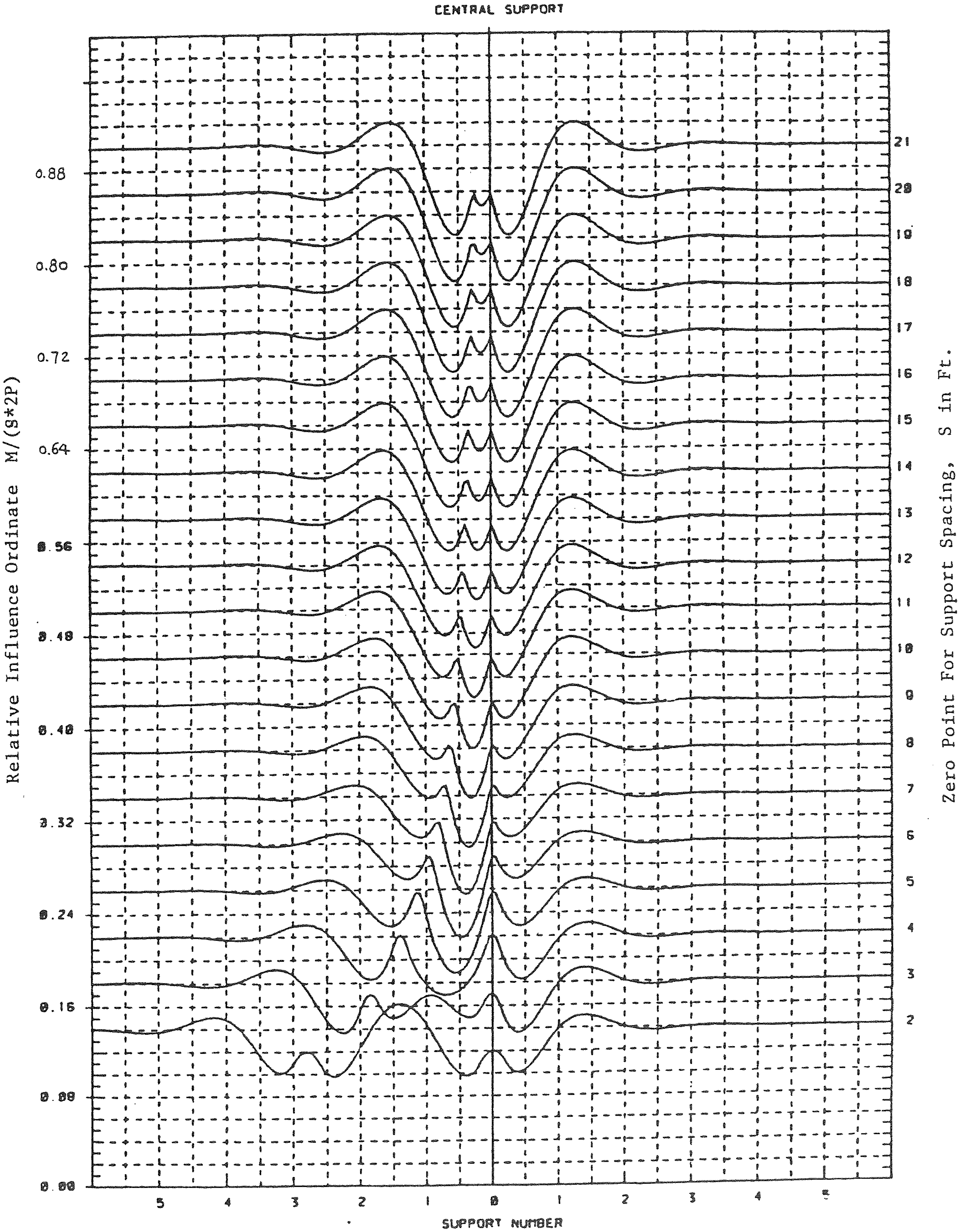


M715 1-1/4 TON TRUCK (TRANSVERSE)

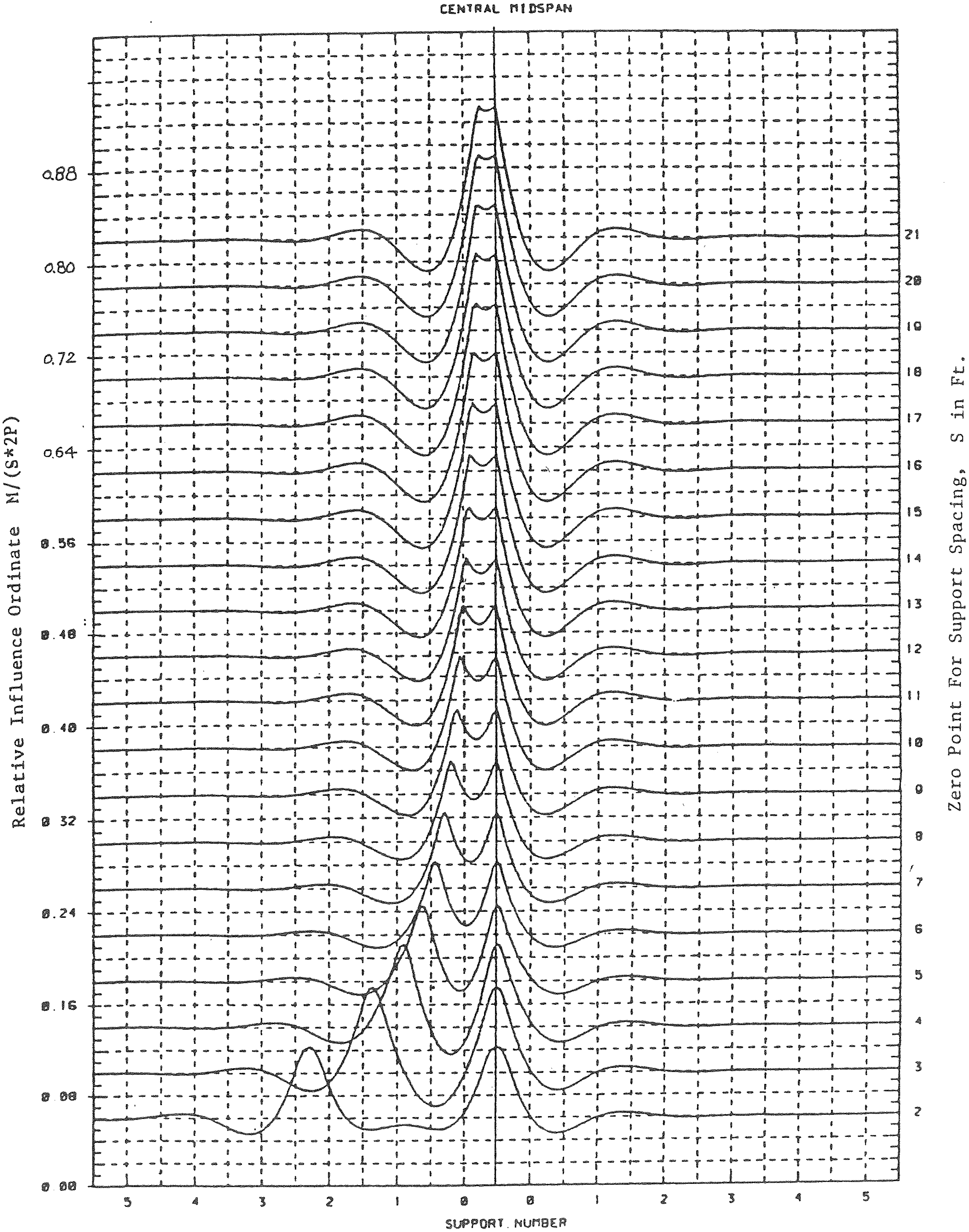


P=2709 lbs (nominal)

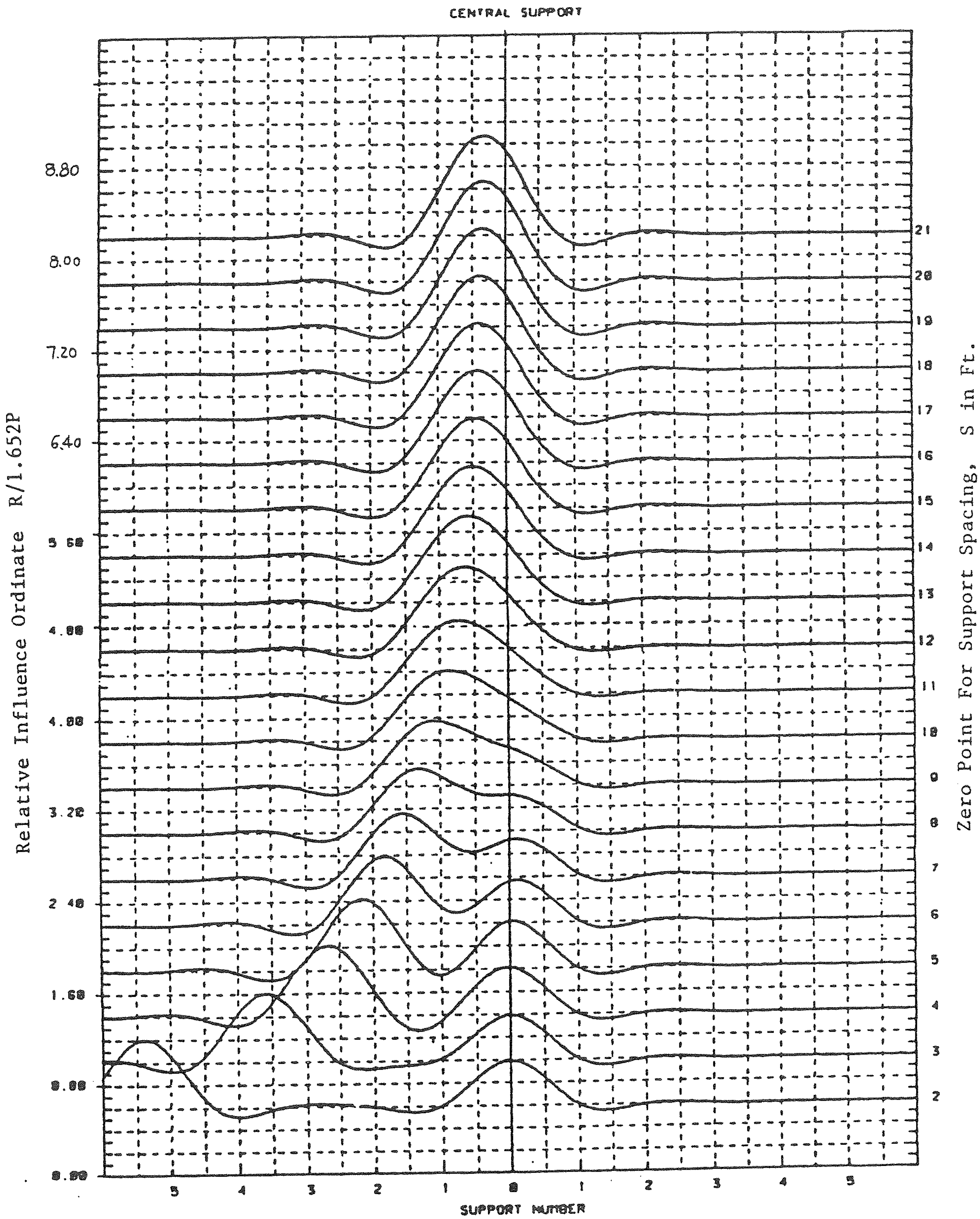
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR M715 1-1/4 TON TRUCK (TRANS.)



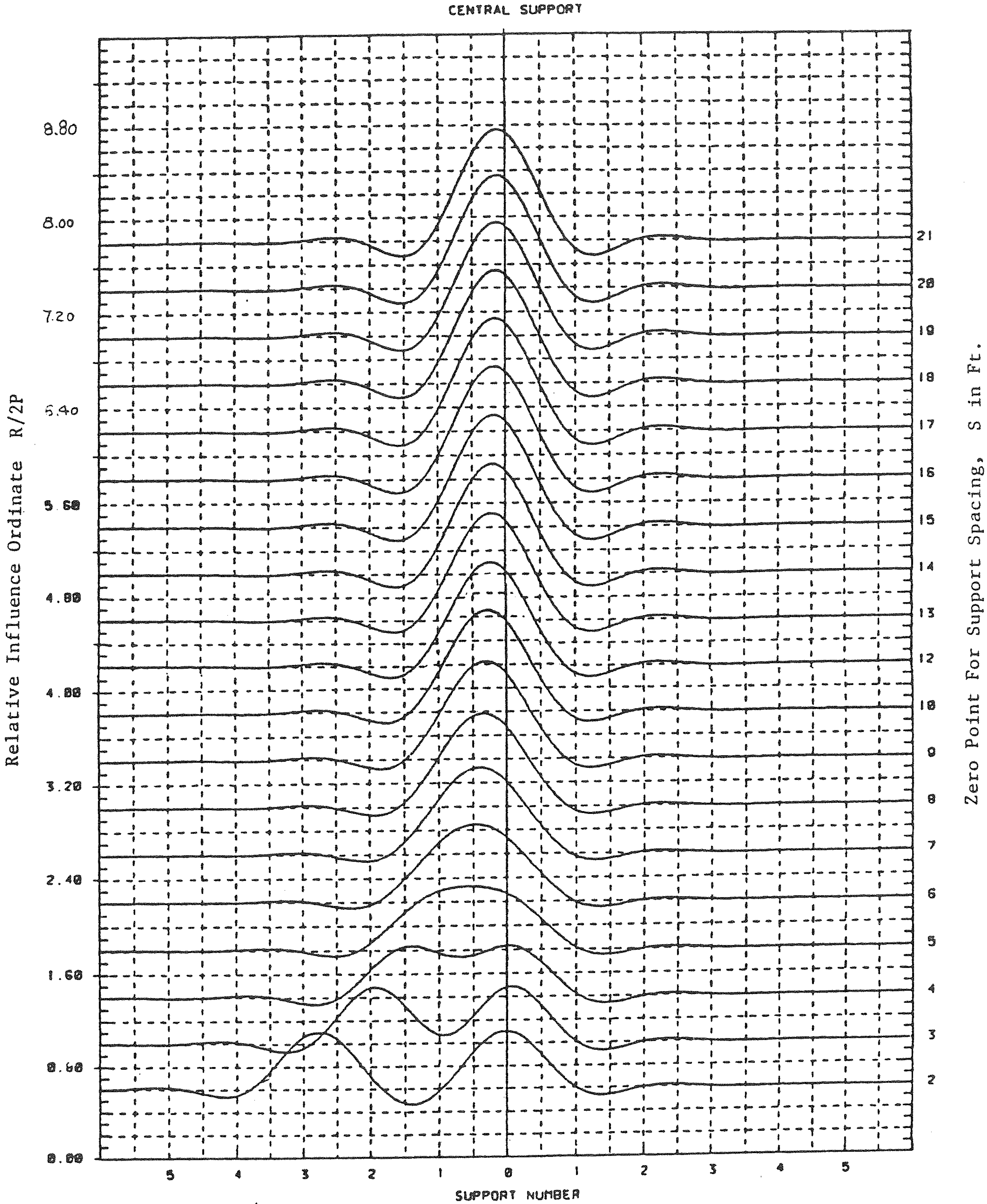
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN FOR M715 1-1/4 TON TRUCK (TRANS.)



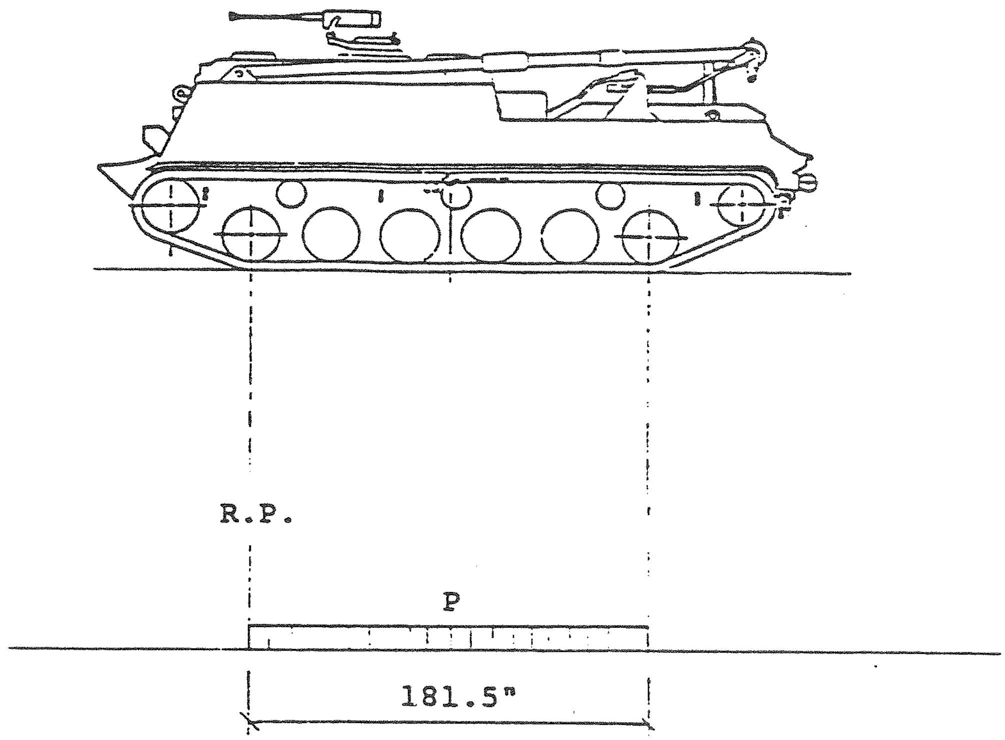
VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR M715 1-1/4 TON TRUCK (LONG'L)



VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR M715 1-1/4 TON TRUCK (TRANS.)

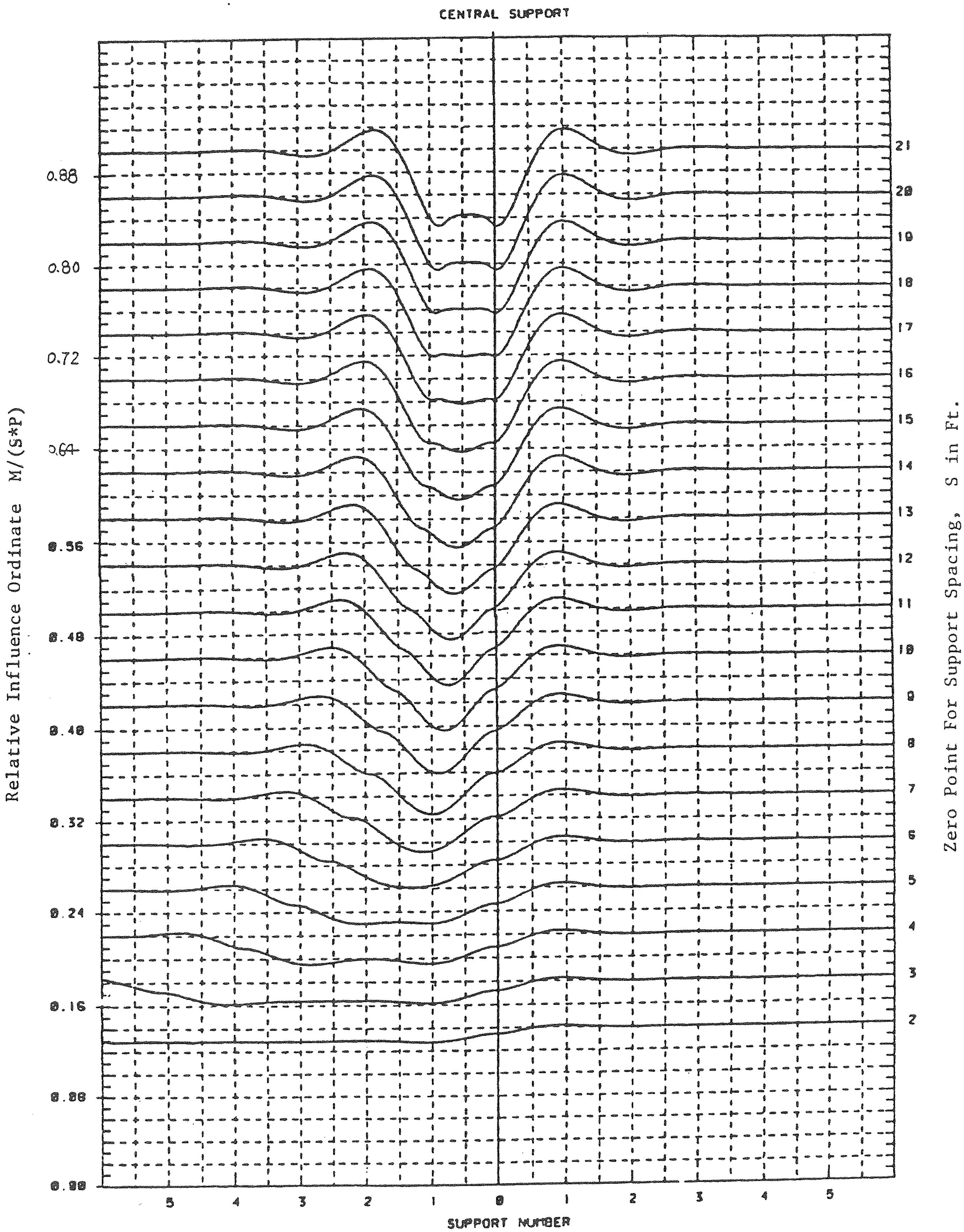


M88 TRACKED VEHICLE (LONGITUDINAL)

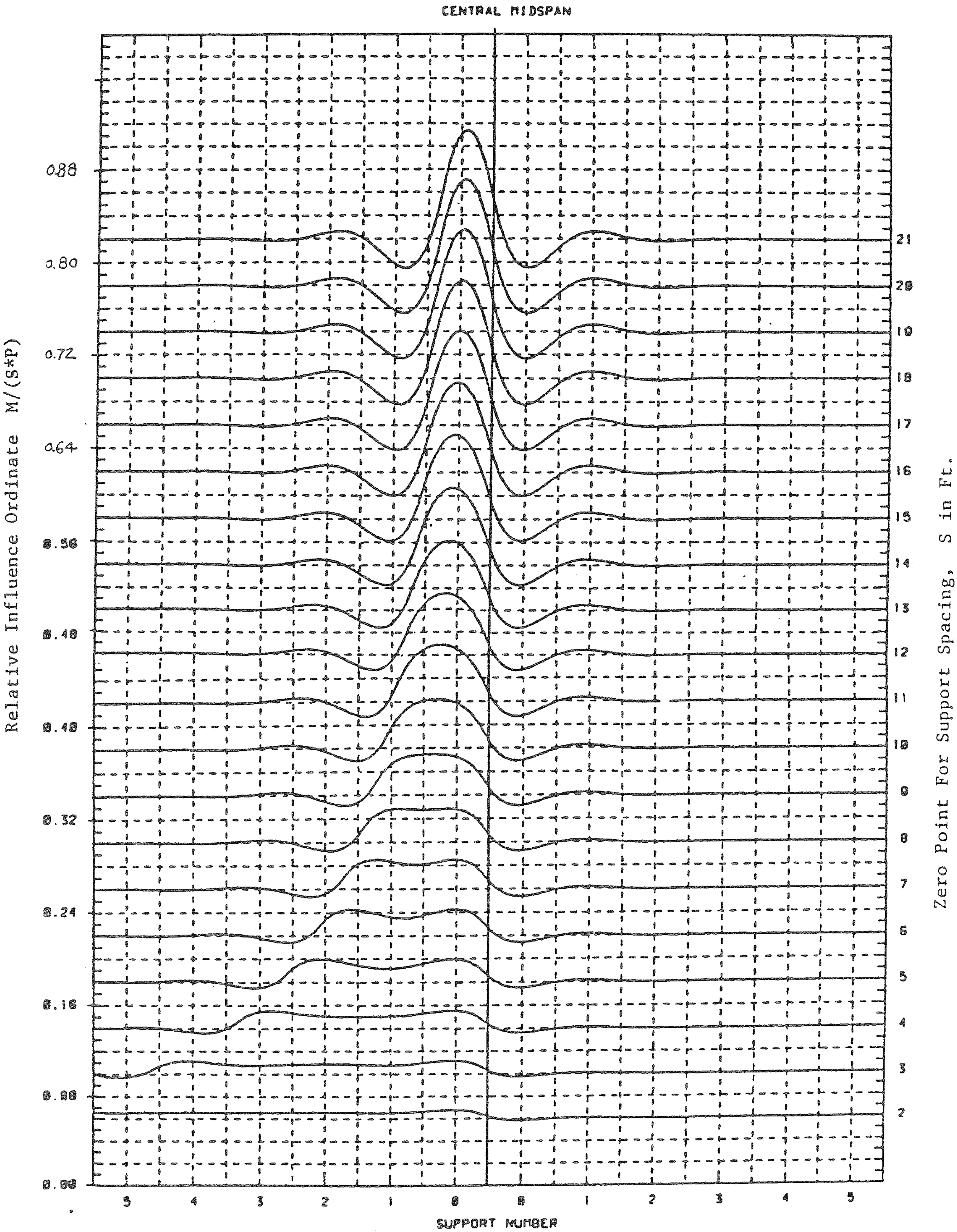


P=56000 lbs. (NOMINAL)

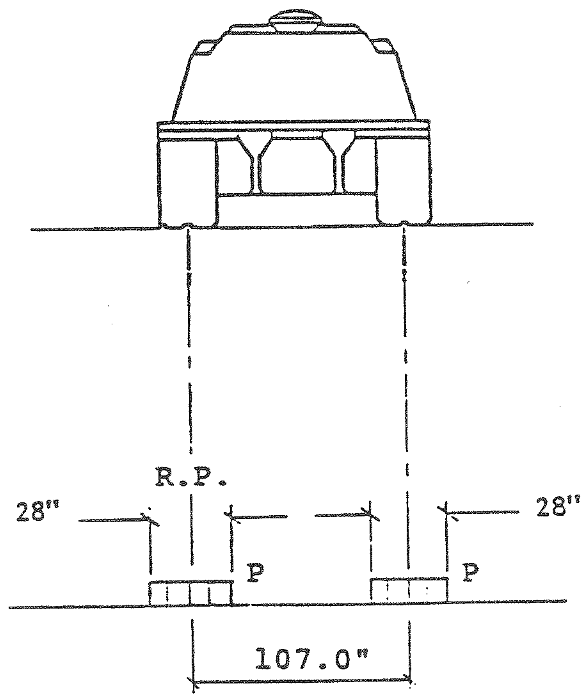
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR M88 TRACKED VEHICLE (LONG'L)



VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN FOR M88 TRACKED VEHICLE (LONG'L)

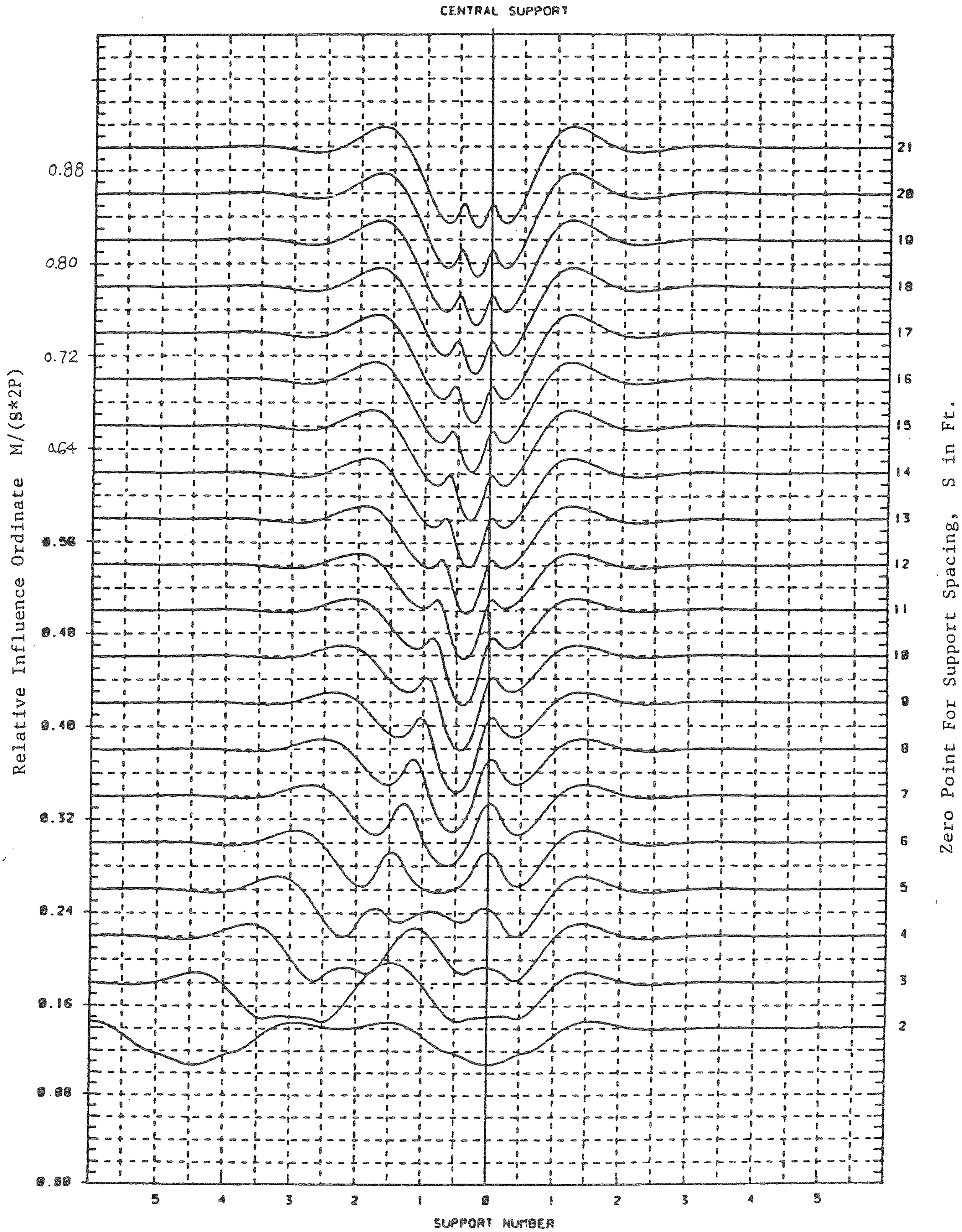


M88 TRACKED VEHICLE (TRANSVERSE)

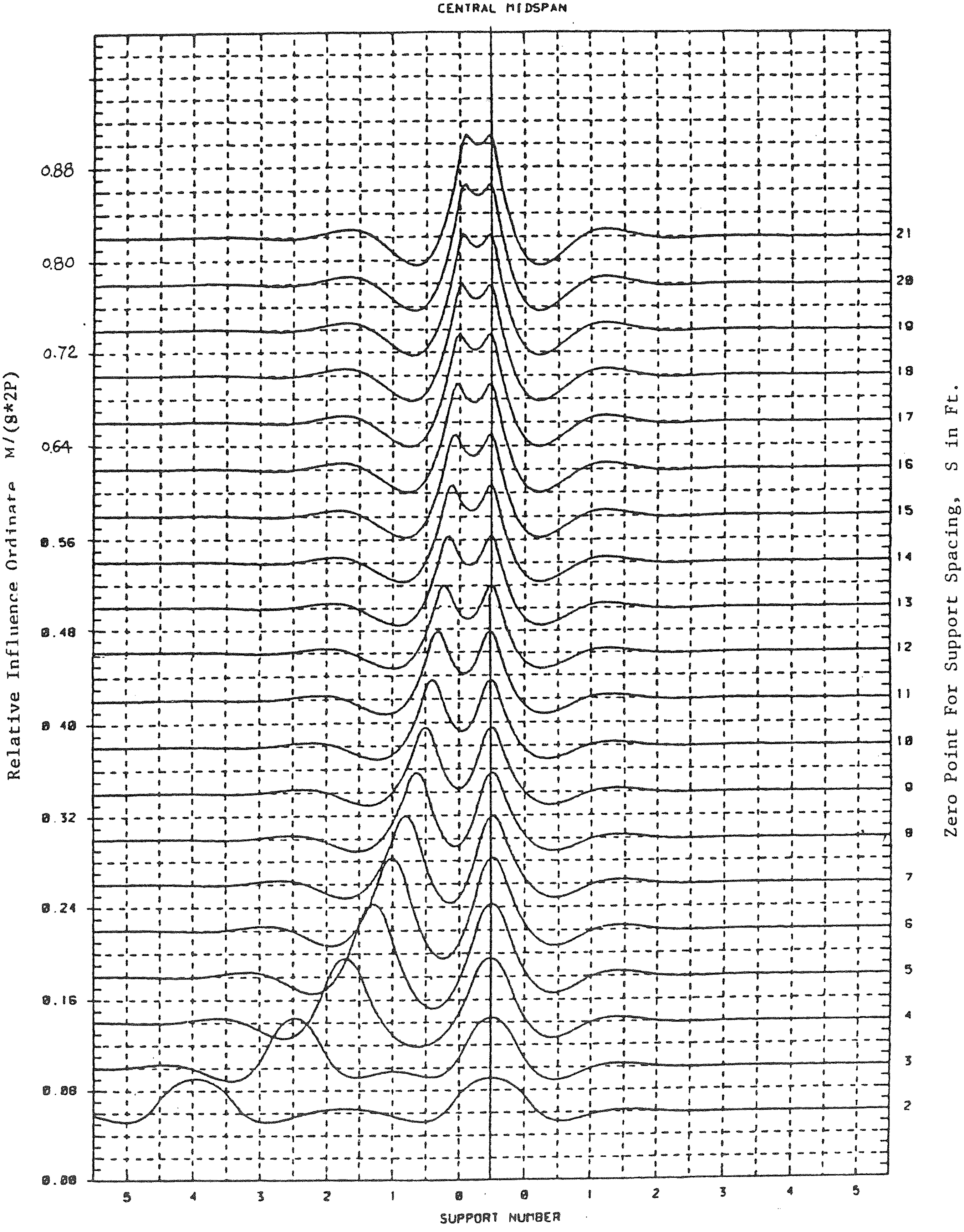


P=56000 lbs. (NOMINAL)

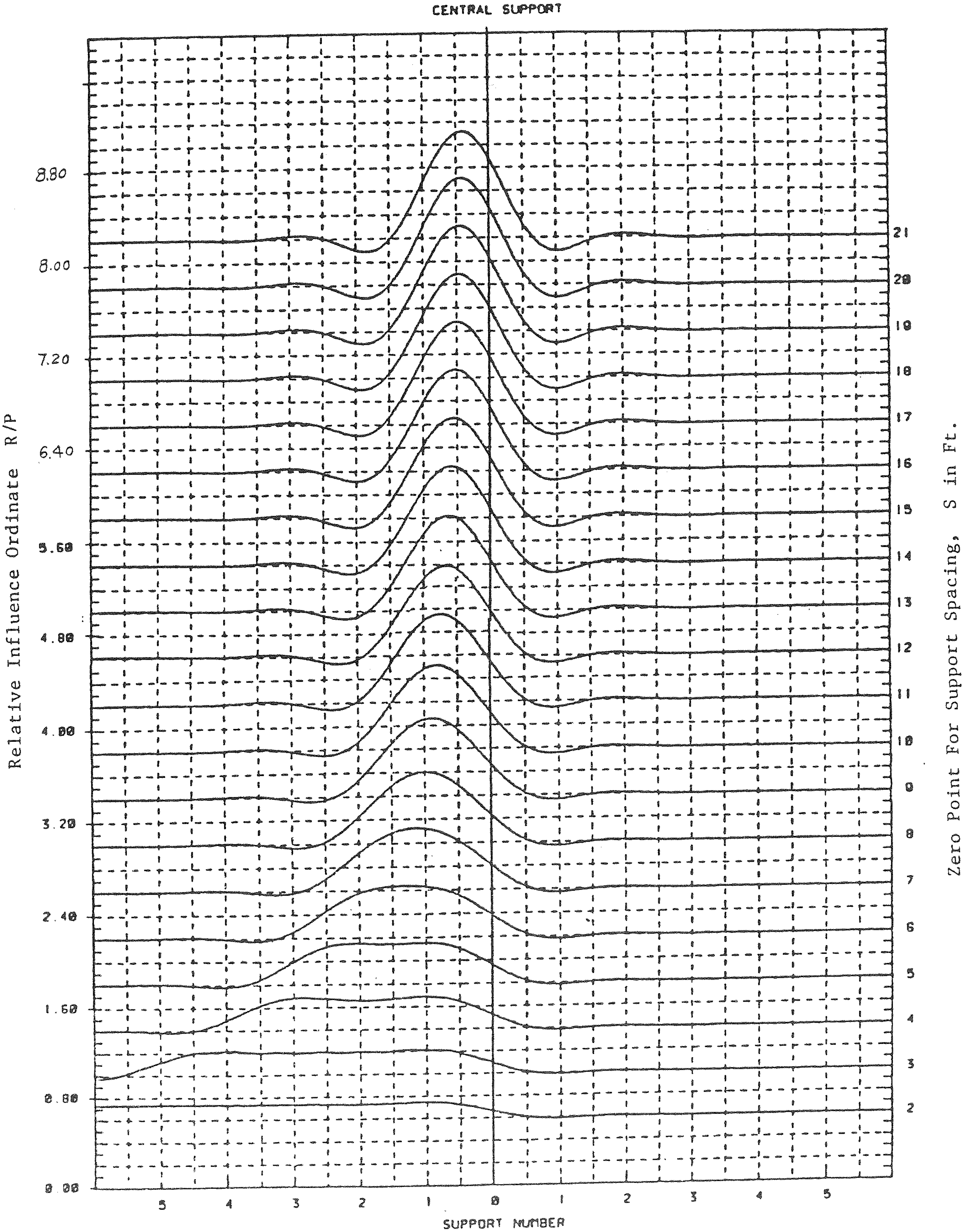
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR M88 TRACKED VEHICLE (TRANS.)



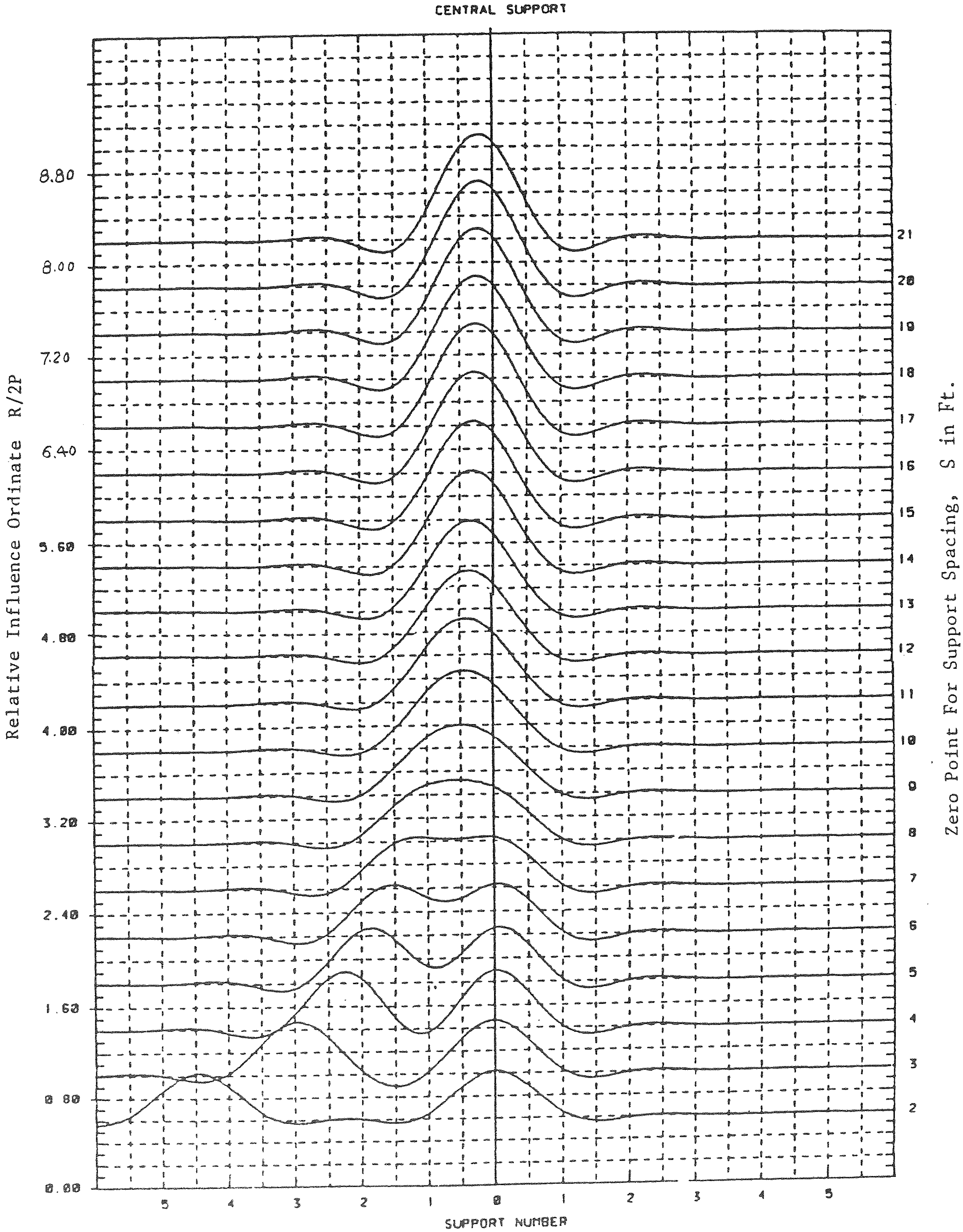
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN FOR M88 TRACKED VEHICLE (TRANS.)



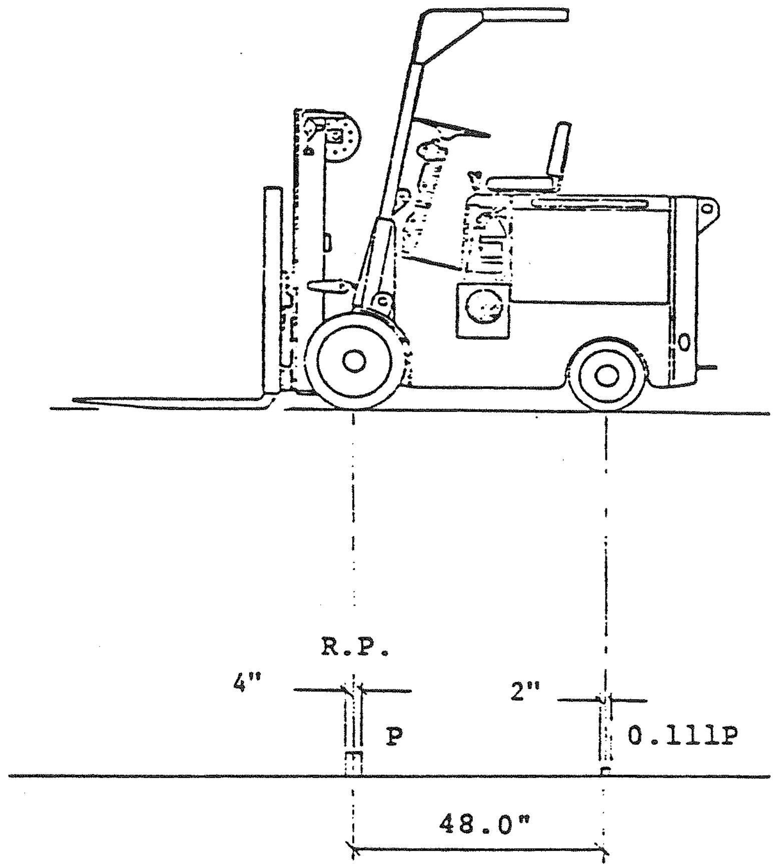
VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR M88 TRACKED VEHICLE (LONG'L)



VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR M88 TRACKED VEHICLE (TRANS.)

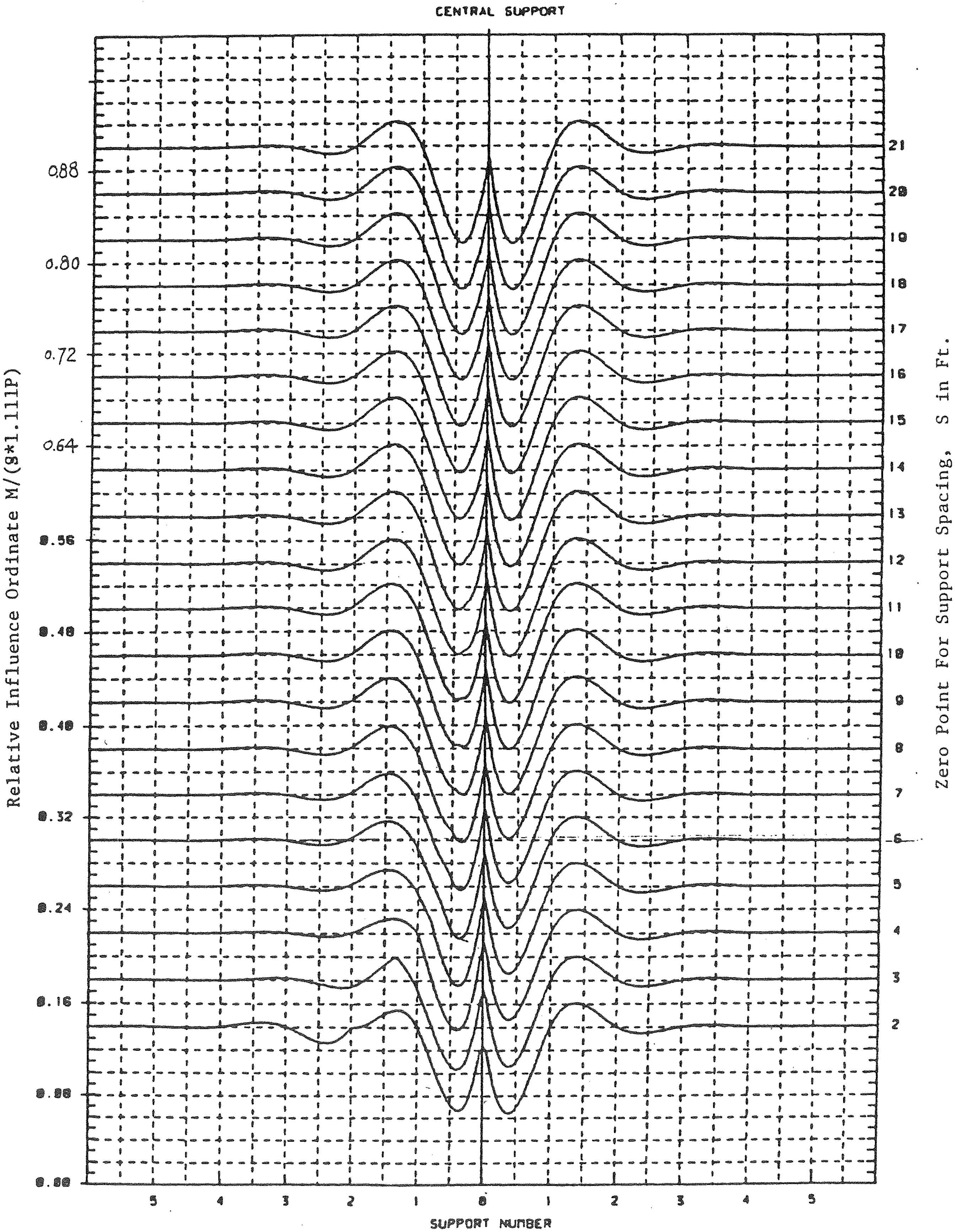


6000 lb. FORK LIFT (LONGITUDINAL)

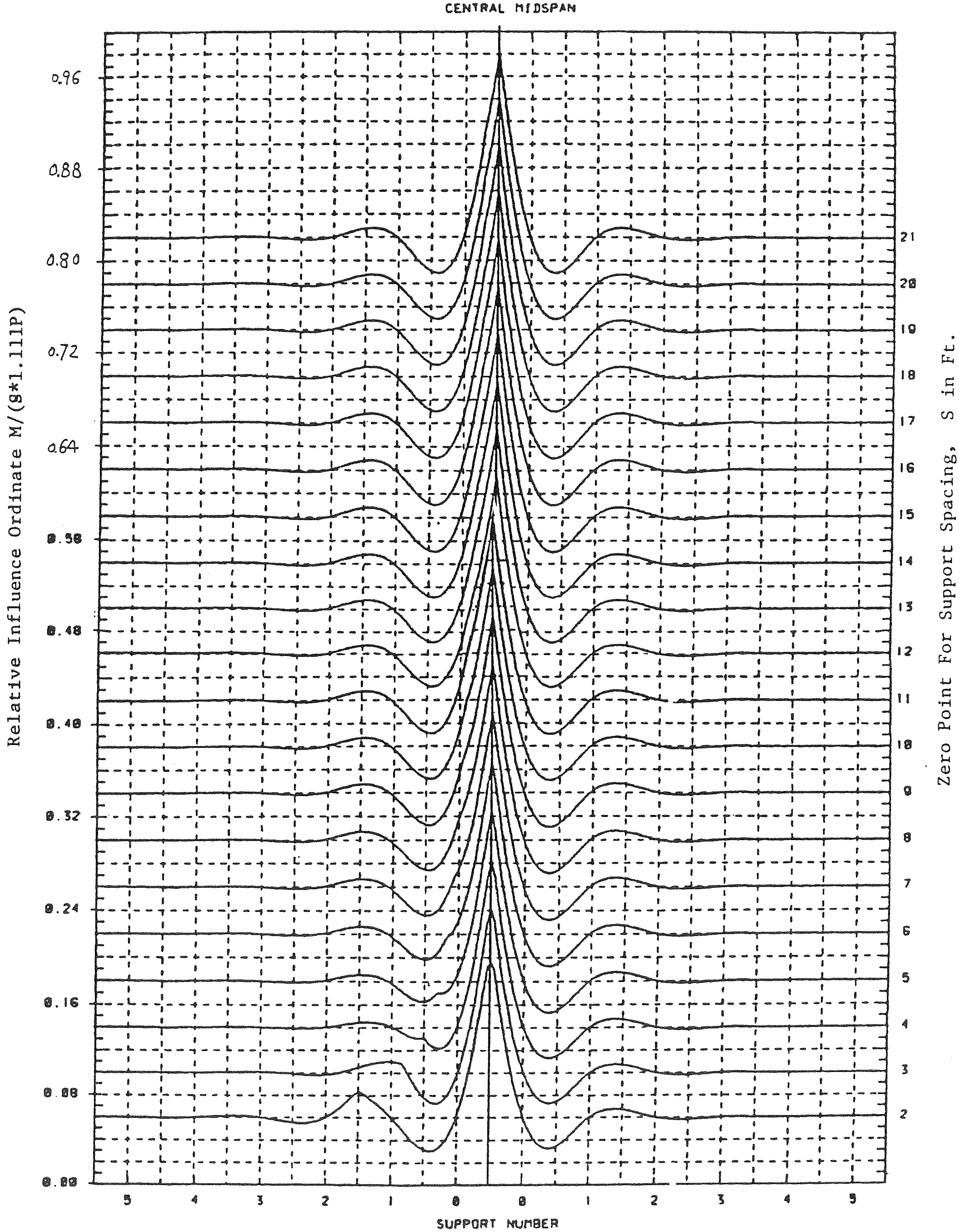


P=7875 lbs. (nominal - fully loaded)

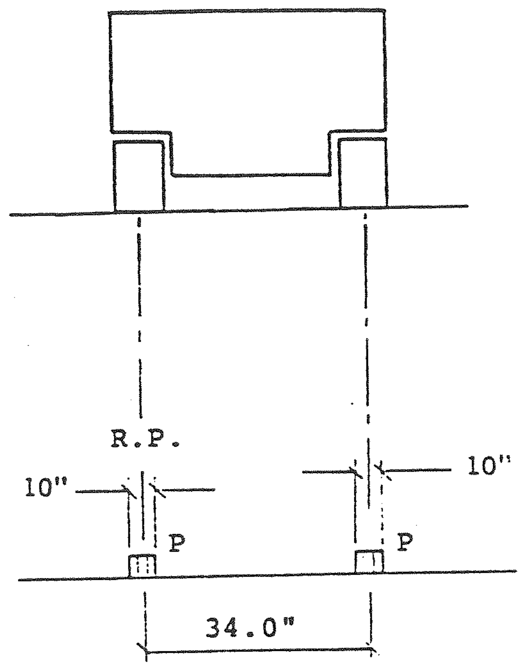
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR 6000 lb FORK LIFT (LONG'L)



VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN FOR 6000 lb FORK LIFT (LONG'L)

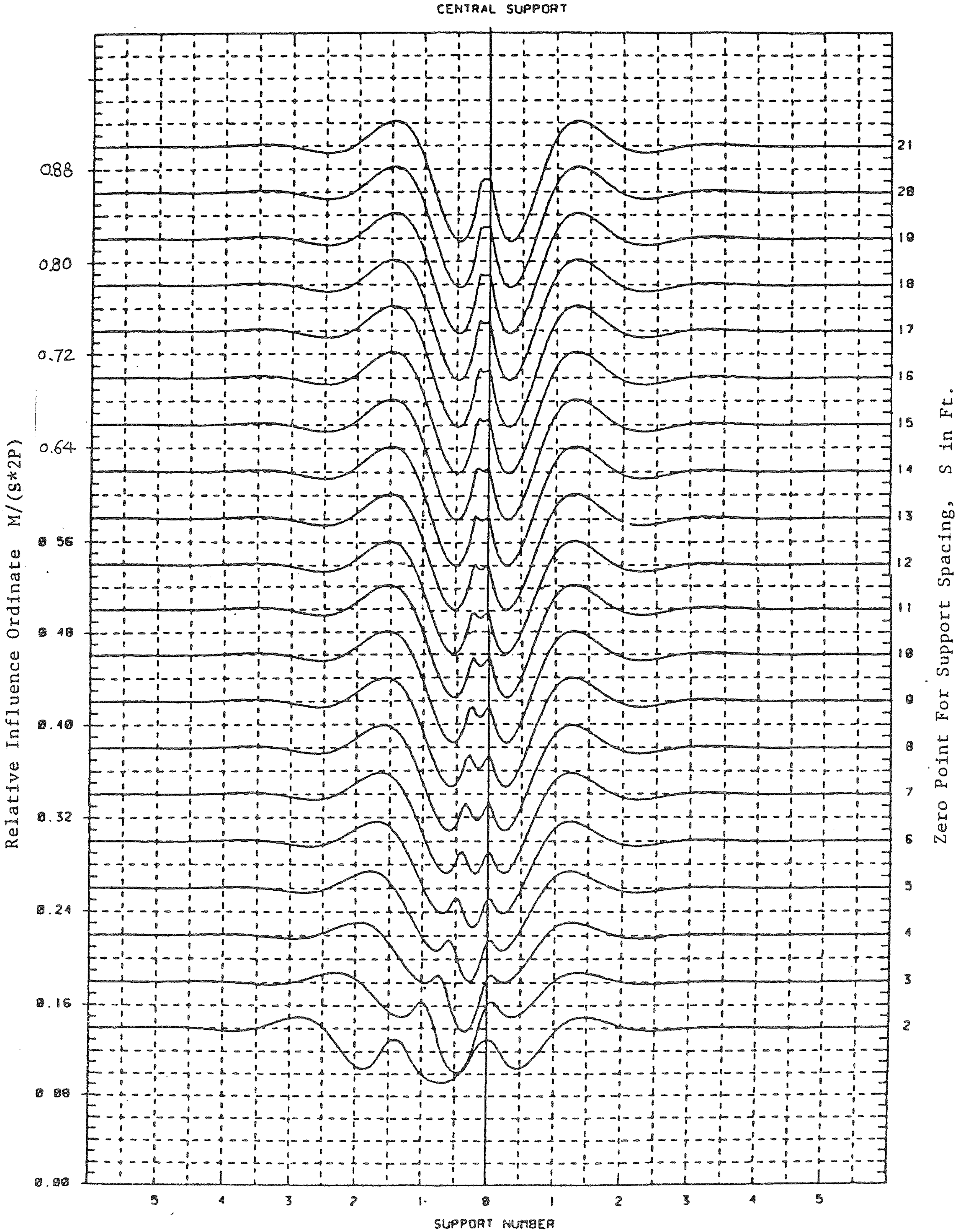


6000 lb. FORK LIFT (TRANSVERSE)

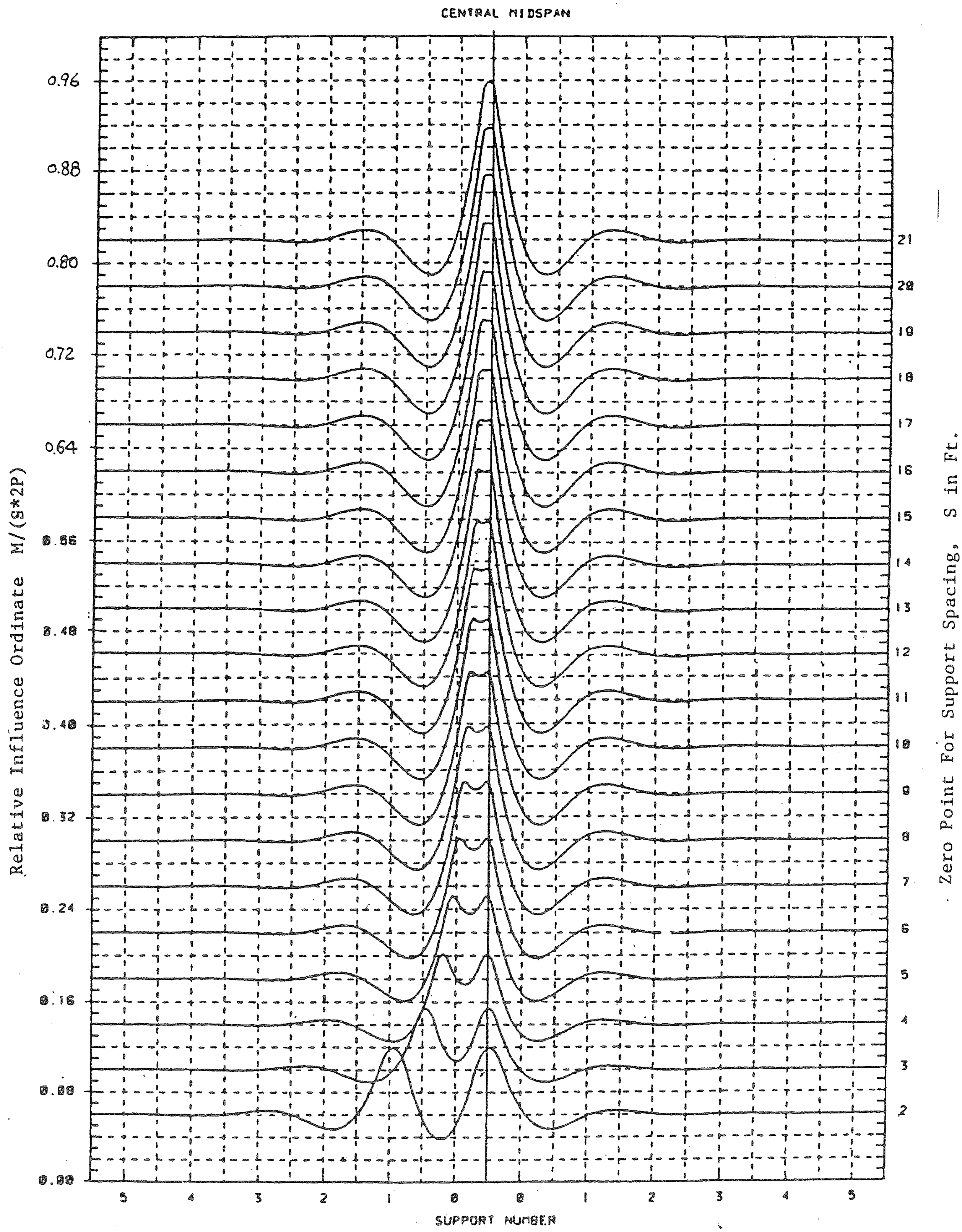


P=7875 lbs. (NOMINAL)

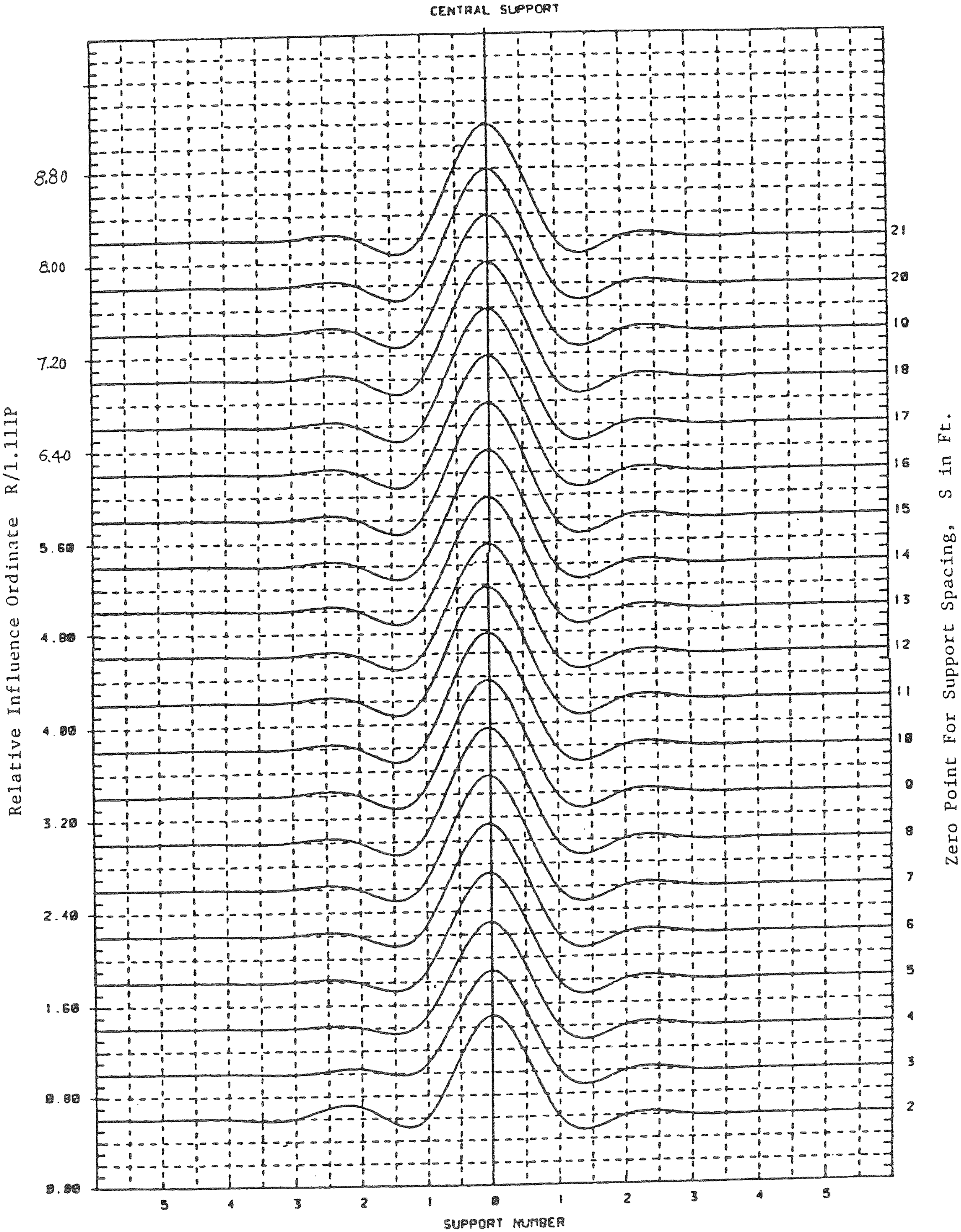
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL SUPPORT FOR 6000 lb FORK LIFT (TRANS.)



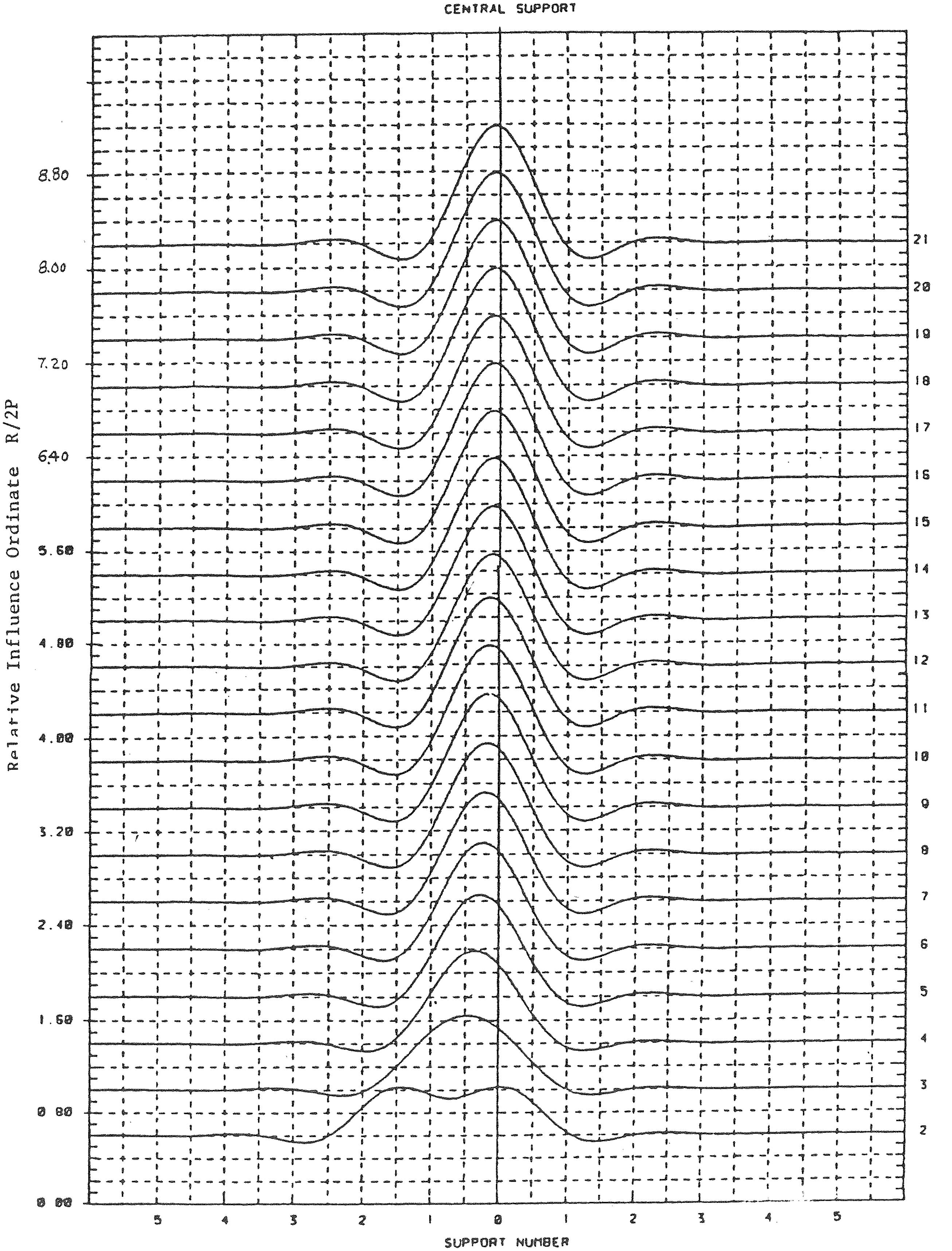
VEHICLE LOAD INFLUENCE LINES FOR MOMENT AT CENTRAL MIDSPAN FOR 6000 lb FORK LIFT (TRANS.)



VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR 6000 lb FORK LIFT (LONG'L)



VEHICLE LOAD INFLUENCE LINES FOR REACTION AT CENTRAL SUPPORT FOR 6000 lb FORK LIFT (TRANS.)



APPENDIX C

SUMMARY OF LOAD AND LOAD DISTRIBUTION METHODS

Summary of Load and Load Distribution Methods

TABLE I Vehicle Weight Conditions

Ship Motion Factors: η_i = Ship Motion Factor in the i direction
η_x = fore and aft factor
η_y = athwartships factor
η_z = vertical factor

Ship Motion Loads: F_i = Ship Motion Load in the i direction
$F_i = \eta_i W_m$

Ship Motion Forces: F_l = Ship Motion Force Longitudinal to Vehicle
F_t = Ship Motion Force Transverse to Vehicle
F_d = Motion Force Downward to Vehicle

Vehicle Oriented Longitudinally:

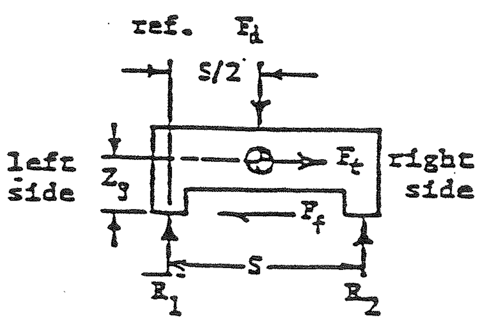
$F_l = F_x$
$F_t = F_y$
$F_d = F_z$

Vehicle Oriented Athwartships:

$F_l = F_y$
$F_t = F_x$
$F_d = F_z$

Wind Loads: $F_w = 0.030\ A_s$ for storm seas
$F_w = 0.015\ A_s$ for moderate seas
A_s = Vehicle Sail Area

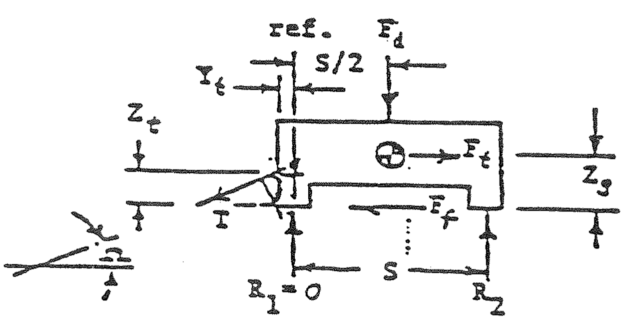
a. Laterial Balance



$R_2 = F_d/2 + F_t \times Z_g/S$

$R_1 = F_d - R_2$

If $R_1 < 0$



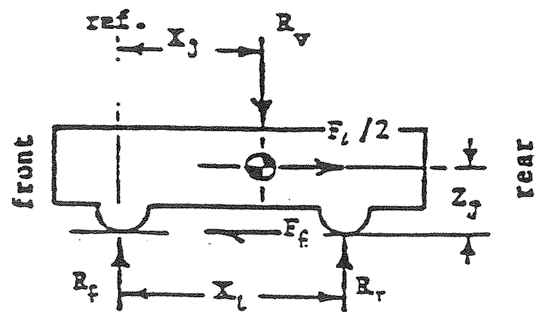
$R_1 = 0$
$T = (F_t \times Z_g - F_d \times S/2)/K$
$R_2 = F_b + T \sin\Omega$
$F_f = F_t - T \cos\Omega$
where, $H = Z_t \times \cos\Omega + Y_t \times \sin\Omega$
$K = H + S \times \sin\Omega$

similar method can be used if $R_2 < 0$.

b. Lengthwise Balance - No tie down force

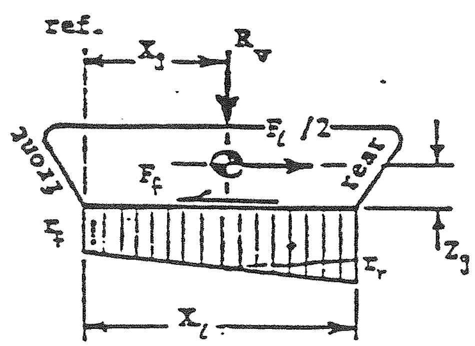
Case 1: Lengthwise motion force toward rear of vehicle

i) Wheeled Vehicle



$$
\begin{aligned}
R_v &= R_1 \text{ (left side)} \\
&= R_2 \text{ (right side)} \\
R_r &= (R_v \times X_g + F_1/2 \times Z_g)/X_1 \\
R_f &= R_v - R_r
\end{aligned}
$$

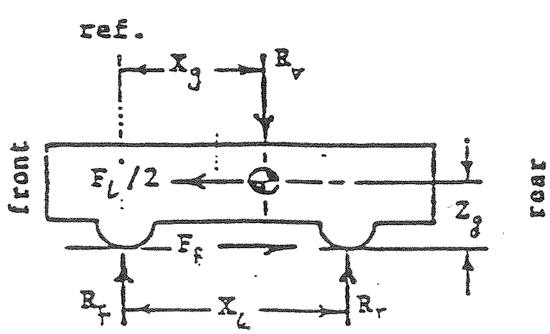
ii) Tracked Vehicle



$$
\begin{aligned}
R_v &= R_1 \text{ (left side)} \\
&= R_2 \text{ (right side)} \\
r_r &= 6/X^2{}_1 \times [R_v \times (X_g - X_1/3) \\
&\quad + F_1/2 \times Z_g] \\
r_f &= 2 \times R_v/X_1 - r_r
\end{aligned}
$$

Case 2; Lengthwise motion force toward front of vehicle

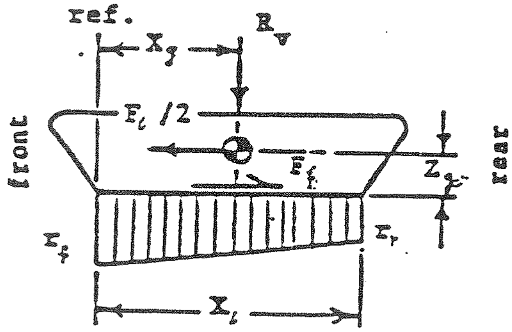
i) Wheeled Vehicle



$$
\begin{aligned}
R_v &= R_1 \text{ (left side)} \\
&= R_2 \text{ (right side)} \\
R_r &= (R_v \times X_g - F_1/2 \times Z_g)/X_1 \\
R_f &= R_v - R_r
\end{aligned}
$$

If $R_r \geq 0$ then R_f and R_r valid, otherwise, tie-down is required.

ii) Tracked Vehicle



$$
\begin{aligned}
R_v &= R_1 \text{ (left side)} \\
&= R_2 \text{ (right side)} \\
r_r &= 6/X^2_1 \times [R_v \times (X_g - X_1/3) \\
&\qquad - F_1/2 \times Z_g] \\
r_f &= 2 \times R_v/X_1 - r_r
\end{aligned}
$$

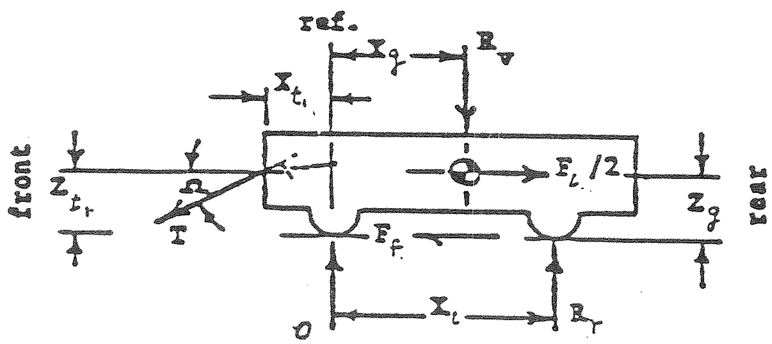
c. Lengthwise Balance - Tie down required

$R_f < 0$, $R_r < 0$, $r_f < 0$, or $r_r < 0$ in 7 a); therefore, perform following lengthwise balance where friction and tie-down forces are accounted for with following assumptiom:

Case	Vehicle Type	Assumption
1	Wheeled	$R_f = 0$
	Tracked	$r_f = 0$
2	Wheeled	$R_r = 0$
	Tracked	$r_r = 0$

Case 1; lengthwise motion force toward rear of vehicle

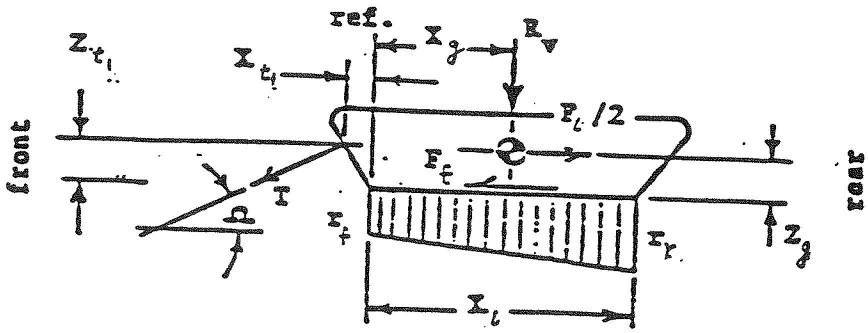
i) Wheeled Vehicle



$$
\begin{aligned}
R_v &= R_1 \text{ (left side} \\
&= R_2 \text{ (right side)} \\
R_f &= 0 \\
T &= (F_1\, Z_g/2 - R_v\, (X_1 - X_g))/K \\
R_r &= R_v + T \sin\Omega \\
F_f &= F_1/2 - T \cos\Omega
\end{aligned}
$$

where,
$$
\begin{aligned}
H &= Z_{t1} \times \cos\Omega + X_{t1} \times \sin\Omega \\
K &= H + X_1 \times \sin\Omega
\end{aligned}
$$

ii) Tracked Vehicle

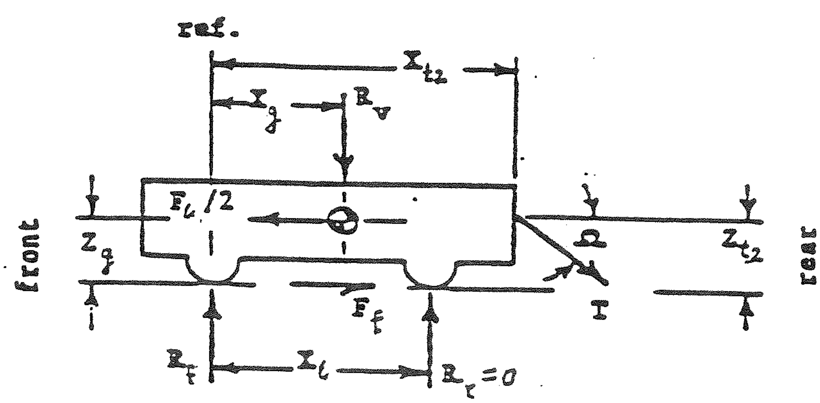


R_v = R_1 (left side)
= R_2 (right side)
r_f = 0
T = $(F_1\ Z_g/2 - R_v\ (2\ X_1/3 - X_g))/K$
K = $H + 2X_1 \times \sin\Omega\ /3$
H = $Z_{t1} \times \cos\Omega + X_{t1} \times \sin\Omega$
r_r = $2\ (R_v + T)/X_1$

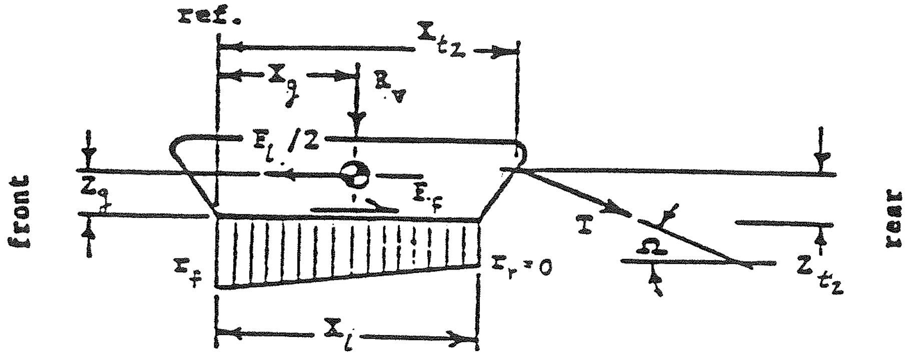
Case 2; Lengthwise motion force toward front of vehicle

i) Wheeled vehicle;

R_v = R_1 (left side)
= R_2 (right side)
R_r = 0
T = $(F_1\ Z_g/2 - R_v\ (X_1 - X_g))/H$
F_f = $F_1/2 - T \cos\Omega$
R_f = $R_v + T \sin\Omega$
H = $Z_{t2} \times \cos\Omega + X_{t2} \times \sin\Omega$



ii) Tracked vehicle;



$$
\begin{aligned}
R_v &= R_1 \text{ (left side)} \\
&= R_2 \text{ (right side)} \\
r_r &= 0 \\
T &= (F_1 Z_g/2 - R_v (X_g - X_1/3))/K \\
r_f &= 2/X_1 \, (R_v + T \sin\Omega) \\
F_f &= F_1/2 - T \cos\Omega \\
H &= Z_{t2} \times \cos\Omega + X_{t2} \times \sin\Omega \\
K &= H - X_1 \sin\Omega/3
\end{aligned}
$$

APPENDIX D

SUMMARY OF PLATING ANALYSIS METHOD AND CRITERIA

SUMMARY OF PLATING ANALYSIS METHOD AND CRITERIA

Analysis Method -

Midspan moment coefficient (β) and Edge moment coefficient (γ)

Use B/b and A/b on Figures 5 and 6 to obtain β or

Maximum plate bending stress, f_p:

$$f_p = \frac{6\beta \; P}{C_o \; t^2} \quad \text{or} \quad \frac{6\gamma P}{C_o \; t^2}$$

for multiple patches

$$f_p = \frac{6}{C_o t^2} \sum_{i=1} (\beta P)_i$$

where C_o is deck function coefficient (Table IV).

Design Criteria -

Storm Sea Parkig Allowable Stress:

$$\sigma_p = F_y$$

Landing and Moderate Sea Parking Allowable Stress

$$\sigma_p = F_b$$

Stress Criteria:

$$f_p \leq \sigma_p$$

APPENDIX E

SUMMARY OF STIFFENER ANALYSIS METHOD AND CRITERIA

Summary of Stiffener Analysis Method and Criteria

Analysis Method

Maximum Midspan Bending Moment and Stress:

M_o: Patch width load distribution factor, ϕ_1:

Use B/b on Figure 12 to obtain ϕ_1.

Plating load distribution factor, ϕ_2:

Calculate the relative rigidity coefficient of plate - stiffener, γ_{ps}

$$ps = \frac{(e_s\ L_s)^4\ t^3}{3.49\ b^3\ \pi^4\ I_s}$$

Use γ_{ps} on Figure 14 to obtain ϕ_2.

Moment due to the live load over rigid supports, M_o:

Evaluate the influence line coefficient, (M_o/PL_s):

Use Figures 9 through 10 to get influence line coefficient for each patch load P_i at critical locations.

Calculate M_o:

$$M_o = \sum_i (M/PL_s)_i\ (PL_s\ \phi_1\ \phi_2)_i = L_s\ \phi_2 \sum_i (\phi_1\ \beta P)_i \qquad \text{where } (M/PL_s) = \beta$$

Moment due to flexibility of the beam suports, M_c:

M_c: Calculate the relative rigidity coefficient of stiffener - beam, γ_{sb}:

$$\gamma_{sb} = \frac{(e_b L_b)^4\ I_s}{0.684 b\ (e_s L_s)^3\ \pi^4 I_b}$$

Moment correction coefficient, (M_c/RL_s):

Use γ_{sb} on Figure 14 to obtain (M_c/RL_s).

Calculate the beam characteristics load, R_o:

$$R_o = \begin{cases} R/B' & \text{, single patch} \\ \dfrac{R}{(b' + B')} & \text{, dual patch} \end{cases}$$

Calculate the beam characteristic load width, B_o:

$$B_o = \begin{cases} B'/2 & \text{, single patch} \\ 1/2\ (b' + B') & \text{, dual patch} \end{cases}$$

Calculate the beam loading coefficient, ϕ_4:

Vehicle aligned with stiffeners and

$L_b \geq 1.5s$

$$\phi_4 = 4/\pi \cos(\pi s/2L_b) \sin(\pi B_o/L_b)(1 + \cos(\pi s/2L_b))$$

Vehicle perpendicular to stiffeners or vehicle aligned with stiffeners and

$L_b < 1.5\ s$

$$\phi_4 = 4/\pi \sin(\pi B_o/L_b)$$

Calculate M_c:

$$M_c = (M_c/RL_s)\ R_o\ b\ L_s\ \phi_4$$

Moment due to dead weight of plating and stiffener,

M_d: Calculate the dead load of plating and stiffener, w_d:

$$w_d = 1/12000 \times (w_s + b/12\ w_p)$$

Calculate M_d:

$$M_d = \eta_z w_d L_s^2/12$$

Maximum bending moment in stiffener, M_s:

$$M_s = M_o + M_c + M_d$$

Parking on a non-strength deck

$$f_{sb} = M_s/SM_{min}$$

Parking on a strength deck

$$f_{sb} = M_s/SM_{min} + \sigma_{primary}$$

Maximum Shear Force and Stress:

Patch width load distribution factor, ϕ_1:

Use B/b on Figure 12 to obtain ϕ_1.

Shear due to live load, V_o:

Calculate the influence line coefficient, (R/P):

Use Figure 11 to get influence line coefficient for each patch P_i at the critical locations:

Calculate V_o:

$$V_o = \sum_i (R/P)_i \quad (P\phi_1)_i \quad = \sum_i (\gamma \phi_1 P)_i \quad \text{where } (R/P) = \gamma$$

Shear due to dead weight of plating and stiffener, V_d:

$$V_d = \eta_z w_d L_s/2$$

Maximum shear force in stiffener, V_s:

$$V_s = V_o + V_d$$

Maximum shear stress in stiffener, f_{sv}:

$$f_{sv} = V_s/A_s$$

Design Criteria -

Allowable Bending Stress:

$$\sigma_{sb} = F_b$$

Bending Stress Criteria:

$$f_{sb} \leq \sigma_{sb}$$

Allowable Shear Stress:

$$\sigma_{sv} = 0.6\ F_b$$

Shear Stress Criteria:

$$f_{sv} \leq \sigma_{sv}$$

APPENDIX F

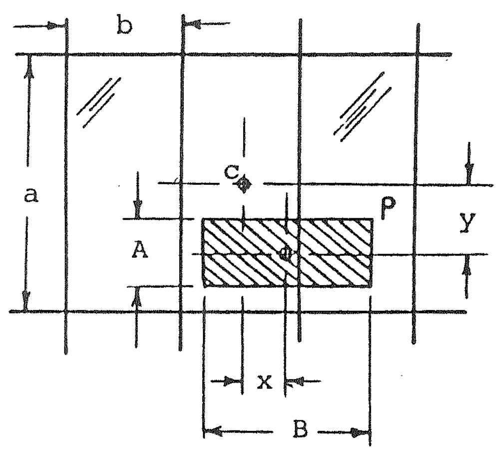
STANDARD WORK SHEETS

WORKSHEET 1; General calculations for midspan response

Title: ______________________________

a = ______, b = ______, A = ______, B = ______, P = ______, x = ______, y = ______

term, n	A_n	B_n	$P_n = \frac{PA_nB_n}{4AB}$ (+ or −)	A_n/b (always +)	B_n/b (always +)	sum factor, f	α_n (always +)	β_n (always +)	$P_n\alpha_n f$	$P_n\beta_n f$
1	2y + A =	2x + B =				+1.0				
2	2y − A =	2x + B =				−1.0				
3	2y + A =	2x − B =				−1.0				
4	2y − A =	2x − B =				+1.0				
								$\sum =$		



P = Total patch load

Range of A: $0 < A \leq a$

Range of B: $0 < B \leq 3b$

Range of x: $0 \leq x \leq 1.5b$

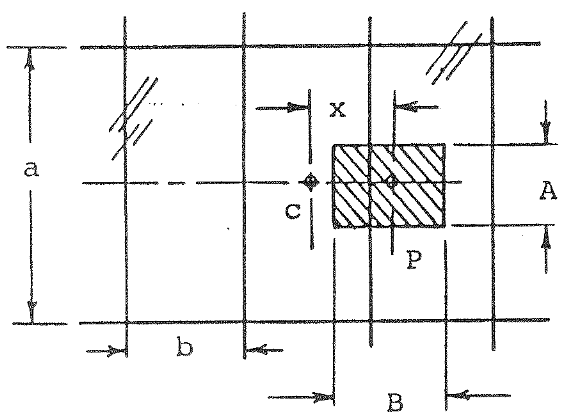
Range of y: $0 \leq y \leq 0.5a$

WORKSHEET 2; General calculations for midspan response when y = 0.0 (longitudinally symmetric)

Title: ______________________________

a = _______, b = _______, A = _______, B = _______, P = _______, x = _______, y = 0.0

term, n	B_n	$P_n = \frac{PB_n}{2B}$ (+ or −)	A/b	B_n/b (always +)	sum factor, f	α_n (always +)	β_n (always +)	$P_n\alpha_n f$	$P_n\beta_n f$
1	2x + B =				+1.0				
2	2x − B =				−1.0				
							$\sum =$		



P = Total patch load

Range of A: $0 < A \leq a$

Range of B: $0 < B \leq 3b$

Range of x: $0 \leq x \leq 1.5b$

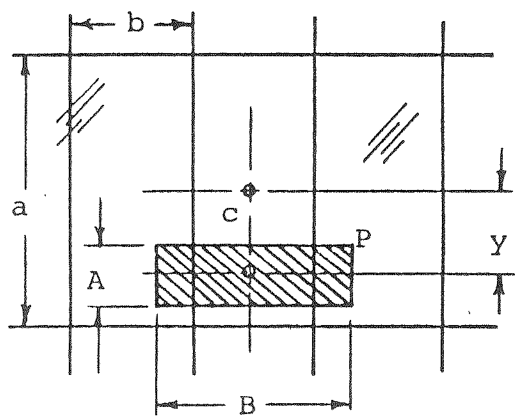
y = 0.0

WORKSHEET 3; General calculations for midspan response when x = 0.0 (transversely symmetric)

Title: ______________________________

a = _______, b = _______, A = _______, B = _______, P = _______, y = _______, x = 0.0

term, n	A_n	$P_n = \frac{PA_n}{2A}$ (+ or −)	A_n/b (always +)	B/b	sum factor, f	α_n (always +)	β_n (always +)	$P_n\alpha_n f$	$P_n\beta_n f$
1	2y + A =				+1.0				
2	2y − A =				−1.0				
							$\sum =$		



P = Total patch load

Range of A: $0 < A \leq a$

Range of B: $0 < B \leq 3b$

Range of y: $0 \leq y \leq 0.5a$

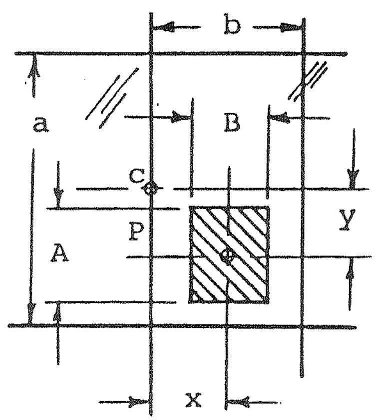
x = 0.0

WORKSHEET 4; General calculations for edge bending moment

Title: ____________________

a = ______, b = ______, A = ______, B = ______, P = ______, x = ______, y = ______

term, n	A_n	B_n	$P_n = \frac{PA_nB_n}{2AB}$ (+ or −)	A_n/b (always +)	B_n/b (always +)	sum factor f	γ_n (always −)	$P_n\gamma_n f$
1	2y + A =	x + B/2 =				+1.0		
2	2y − A =	x + B/2 =				−1.0		
3	2y + A =	x − B/2 =				−1.0		
4	2y − A =	x + B/2 =				+1.0		
							$\sum=$	



P = Total patch load

Range of A: $0 < A \leq a$

Range of B: $0 < B \leq b$

Range of x: $0 \leq x < b$

Range of y: $0 \leq y \leq a/2$

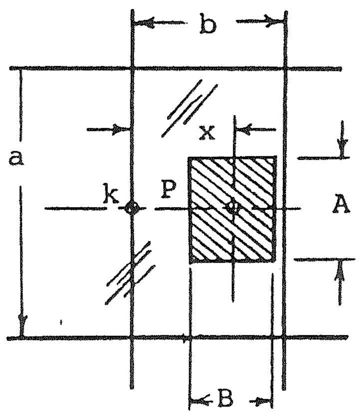
Patch load must be entirely within panel

WORKSHEET 5; General calculations for edge bending moment when y = 0.0 (longitudinally symmetric)

Title: ______________________________

a = ______, b = ______, A = ______, B = ______, P = ______, x = ______, y = 0.0

term, n	B_n	$P_n = \frac{PB_n}{B}$ (+ or −)	A/b	B_n/b (always +)	sum factor f	γ_n (always −)	$P_n \gamma_n f$
1	x + B/2 =				+1.0		
2	x − B/2 =				−1.0		
						$\sum =$	



P = Total patch load

Range of A: $0 < A \leq a$

Range of B: $0 < B \leq b$

Range of x: $0 \leq x < b$

y = 0.0

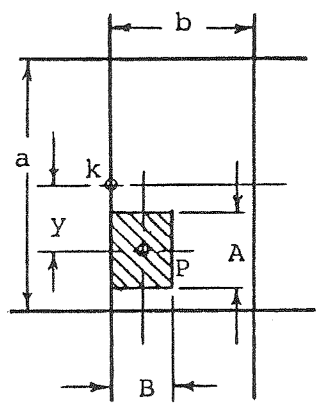
Patch load must be entirely with panel

WORKSHEET 6; General calculations for edge bending moment when $x = B/2$ (patch load aligned on panel edge)

Title: ______________________________

a = _______, b = _______, A = _______, B = _______, P = _______, y = _______, $x = B/2$

term, n	A_n	$P_n = \frac{PA_n}{2A}$ (+ or −)	A_n/b (always +)	B/b	sum factor f	γ_n (always −)	$P_n \gamma_n f$
1	$2y + A =$				+1.0		
2	$2y - A =$				-1.0		
						$\sum=$	



P = Total patch load

Range of A: 0 A a

Range of B: 0 B b

Range of y: 0 y a/2

$x = B/2$

Patch load must be entirely within panel

WORKSHEET 7: Structural Properties

STRUCTURAL PROPERTIES			SHIP OR PROJECT		
Deck Structure					
PLATE	t				
	material				
	w_p				
STIFFENER	size				
	material				
	w_s				
	A				
	A_s				
	b_e				
	Is				
	SM_{min}				
BEAM	size				
	material				
	wb				
	A				
	A_s				
	b_e				
	I_b				
	SM_{min}				
GEOMETRY	b				
	Ls				
	L_b				
PARAMETERS	γ_{ps}				
	γ_{sb}				
	w_p				
	c_s				
	c_b				
PRIMARY					
REMARKS					

WORKSHEET 8: Ship Motion Factors

SHIP MOTION FACTORS AND LOADS | SHIP OR PROJECT

	SPOT	1	2	3	4	5	6	7	8	9	10
LOCATION	X										
	Y										
	Z										
STORM SEA CONDITION	xs										
	ys										
	zs										
	W										
	F_{xs}										
	F_{ys}										
	F_{zs}										
	F_W										
MODERATE SEA CONDITION	xm										
	ym										
	zm										
	W										
	F_{xm}										
	F_{ym}										
	F_{zm}										
	F_W										
REMARKS											

SHIP MOTION EQUATIONS:

η_{xs} =

η_{ys} =

η_{zs} =

η_{xm} =

η_{ym} =

η_{zm} =

WORKSHEET 9: Force Balance Worksheet for ____________________

SPOT SEACONDITION

R_m =	X_g =	Z_t =	Z_{t1} =	Z_{t2} =
F_d =	Z_g =	T_y =	X_{t_1} =	Z_{t2} =
S =	X_1 =	=		

			CASE 1	CASE 2	CASE 3	CASE 4
Vehicle Alignment			Longitudinal	Longitudinal	Transverse	Transverse
Lengthwise Force			Toward Rear	Toward Front	Toward Rear	Toward Front
LATERAL BALANCE	NO TIE DOWN FORCE	F_1				
		F_t				
		R_2				
		R_1				
		Comment				
	TIE DOWN FORCE REQUIRED	H				
		R_1				
		R_2				
		R_f				
		T				

		Vehicle Side	Left	Right	Left	Right	Left	Right	Left	Right
LENGTHWISE BALANCE	NO TIE DOWN FORCE	R_v	R_1 =	R_2 =	R_1 =	R_2 =	R_1 =	R_2 =	R_1 =	R_2 =
		R_r or r_r								
		R_f or r_f								
		Comment								
	TIE DOWN FORCE REQUIRED	H								
		R_f or r_f								
		R_r or r_r								
		T								
		F_f								

WORKSHEET 10: Plate and Stiffener Analysis

PLATE AND STIFFENER STRESS ANALYSIS	SHIP OR PROJECT:
DECK STRUCTURE:	SEA CONDITIONS:
SPOT:	ORIENTATION:

Analysis Parameters:	Remarks
$R =$ $P =$ $A =$ $B =$	
$L_b =$ $e_b =$ $a =$ $b =$	
$L_s =$ $e_s =$ $A/b =$ $B/b =$	

Plate Analysis:	Remarks
$t =$ $F_y =$ $F_b =$	
$\beta =$ $f_p =$ $\sigma_p =$	
$t_{reqd} =$	

Stiffener Analysis:	Remarks
$I_s =$ $A_s =$; $SM_{min} =$; $f_b =$	
$I_b =$ $\sigma_{primary} =$; $w_d =$	
$\phi_1 =$ $\phi_2 =$	
$\left(\frac{M_o}{SP}\right)_{support} =$ $\left(\frac{M_o}{SP}\right)_{mid\text{-}span} =$ $M_o = \sum\left(\frac{M_o}{SP}\right) \phi_1 \phi_2 SP =$	
$R_o =$ $B_o =$	
$O_4 =$ $\gamma_{sb} =$	
$\left(\frac{M_c}{RL_s}\right) =$ $M_c =$	
$M_d =$	
$f_{sb} =$ $\sigma_{sb} =$ $SM_{reqd} =$	
$\left(\frac{R_o}{P}\right) =$	
$V_o = \left(\frac{R_o}{P}\right) P\phi_1$ $V_d =$ $V_s =$	
$f_{sv} =$ $\sigma_{sv} =$ $As_{reqd} =$	

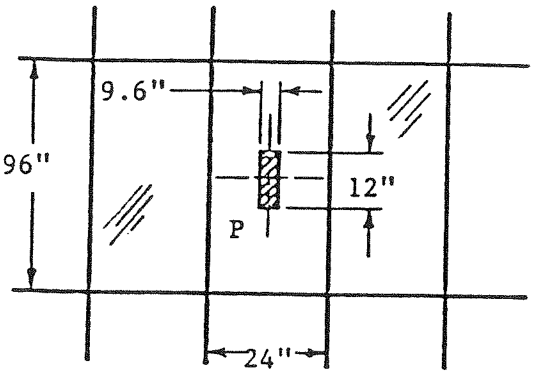
APPENDIX G

EXAMPLES

EXAMPLES

Examples 1 through 7 are plating analysis problems (from Reference e) to illustrate solution of plating response using plots and worksheets. Deck function coefficients are not included in these examples. Example 8 is a design problem to illustrate the procedures.

Example 1; Find A) midspan transverse bending moment and B) edge transverse bending moment for the following tire patch load arrangement:

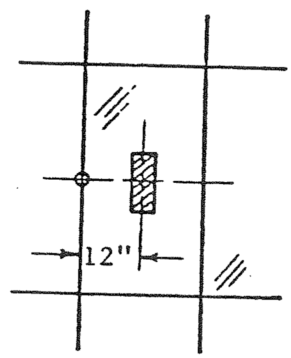


Load, P = 2500 lbs.
a = 96", b = 24"
a/b = 4
A = 12", B = 9.6"
A/b = 0.5
B/b = 0.4
Plate thickness = 0.375"

A) Midspan bending moment and stress; Using Figure 5.5,

$\beta = 0.15124$
$M = \beta P = 0.15124 \times 2500 = 378.1$ in-lb
σ = bending stress = $6M/t^2$
$\sigma = 6 \times 378.1/(0.375^2) = 16520$ psi.

B) Edge bending moment and stress; Using Figure 6.5,



x = 12", y = 0"

WORKSHEET 5. General calculations for edge bending moment when y = 0.0 (longitudinally symmetric)

Title: EXAMPLE 1B

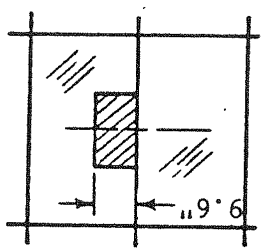
a - 96", b - 24", A - 12", B - 9.6", P - 2500#, x - 12", y - 0.0

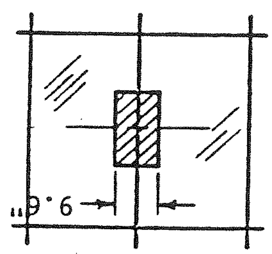
term, n	B_n	$P_n = \frac{PB_n}{B}$ (+ or -)	A/b	B_n/b (always +)	sum factor f	γ_n (always -)	$P_n \gamma_n f$
1	x + B/2 = 16.8	4375#	0.5	0.7	+1.0	-0.09015	-394.4
2	x - B/2 = 7.2	1875#	0.5	0.3	-1.0	-0.08460	+158.6
						$\Sigma =$	-235.8

From worksheet, M = -235.8 in-lb, which is exceeded by the midspan bending moment.

Example 2; Find edge bending moment for load for Example 1) aligned along long edge.

Consider two possible configurations





a = 96', b = 24"
A = 12", P = 2500 lbs
plate th. = 0.375"

A) patch load aligned on stiffener

B) patch load with edge at stiffener

A) For patch load aligned on stiffener solve for half of load and multiply result by two.

B' = 9.6 / 2 = 4.8", A/b = 0.5, B'/b = 0.2, P' = 1250 lbs

Using Figure 6.5, $\gamma' = -0.06886$

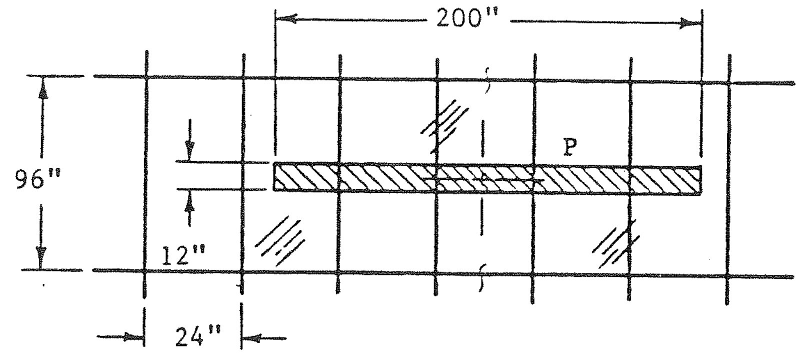
$M = \gamma' P' \times 2 = -0.06886 \times 1250 \times 2 = -172.2$ in-lb

Note that this solution is also obtained with $M = \gamma' P$

B) Patch Load with edge at stiffener; B = 9.6", B/b = 0.4, $\gamma = -0.09255$

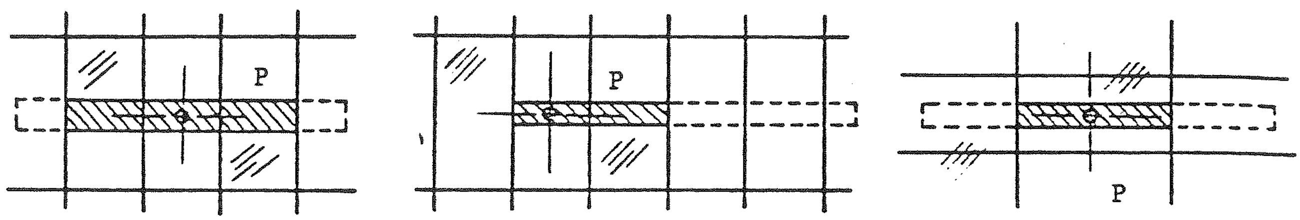
$M = \gamma P = -0.09255 \times 2500 = -231.4$ in-lb

Example 3; Find maximum deflection and midspan bendng moment and approximate maximum edge moment for tank-tread load extending across several panels (uniform load).



P = uniform pressure load of 22000 lbs
a = 96", b = 24"
flexural rigidity,
D = 1.449 x 10^5

Midspan Responses; consider three possible configurations:



A) P = 72/200 x 22000
= 7920 lbs
B = 72", B/b = 3.0
A = 12", A/b = 0.5
a = 96", b = 24"

B) P = 48/200 x 22000
= 5280 lbs
B = 48", B/b = 2.0
A = 12", A/b = 0.5
a = 96", b = 24"

C) P = 96/200 x 22000
= 10560 lbs
B = 12", B/b = 0.5
A = 96", A/b = 4.0
a = 96", b = 24"

Notes: For transversely-aligned loads A) and B), only that portion of load in immediate and adjoining panels may be considered. Remaining portion is assumed to have minimal effect at point of required response.

For longitudinally-aligned load C), only that portion of load in immediate panel may be considered. Remaining portion is assumed to have minimal effect at point of required response.

Config. A); From Figure 5.5, $\beta = 0.0177$

M = midspan bending moment = βP = 0.0177 x 7920
= 140.2 in-lb

Config. B); Apply Worksheet 2

General calculations for midspan response when y = 0.0 (longitudinally symmetric)

Title: EXAMPLE 3B

a = 96", b = 24", A = 12", B = 48", P = 5280#, x = 12", y = 0.0

term, n	B_n	$P_n = \frac{PB_n}{2B}$ (+ or −)	A/b	B_n/b (always +)	sum factor, f	α_n (always +)	β_n (always +)	$P_n \alpha_n f$	$P_n \beta_n f$
1	2x + B = 72	3960	0.5	3.0	+1.0	0.00087	0.0177	3.445	70.1
2	2x − B = −24	−1320	0.5	1.0	−1.0	0.00646	0.0818	8.527	108.0
							Σ =	11.972	178.1

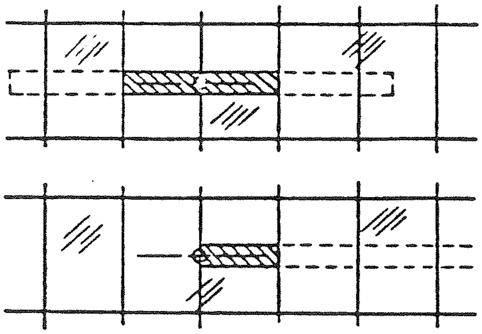
M = Midspan bending moment = βp = 178.1 in-lb (off worksheet)

Config. C); From Figure/5.5 β = 0.02886

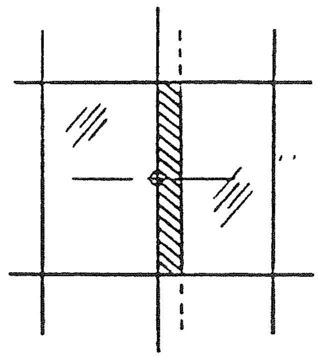
M = midspan bending moment = βP = 0.02886 x 10560
= 304.8 in-lb

Configuration C) governs for midspan response and results in a bending stress, $\sigma = 6M/t^2$
$= 6 \times 304.8/(0.375^2)$
= 13000 psi

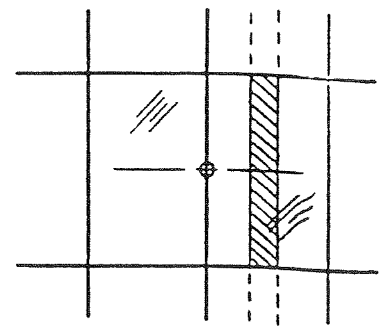
Edge Bending Moment Response; consider three possible configurations:



Configuration A)



Configuration B)



Configuration C)

Config. A); Based on Example 2A), the symmetric configuration will result in twice the magnitude of bending moment than the assymetric configuration. Therefore, solve using the asymmetric arrangement and multiply the result by two.

a = 96", b = 24", A = 12", B = 24", P = 24/200 x 22000
A/b = 0.5, B/b = 1.0 = 2640 lbs

From Figure 6.5, $\gamma = -0.07091$

M = edge bending moment = $2\gamma P$ = -0.07091 x 2640 x 2
= -374.4 in-lb

Config. B); a = 96", b = 24", A = 96", B = 12", P = 96/200 x 22000
A/b = 4.0, B/b = 0.5, = 10560 lbs

From Figure 6.5, $\gamma = -0.01593$

M = edge bending moment = γP = -0.01593 x 10560
= -168.2 in-lb

Config. C); Apply Figure 6.5 and Worksheet 5.

WORKSHEET 5; General calculations for edge bending moment when y = 0.0 (longitudinally symmetric)

Title: EXAMPLE 3C

a = 96", b = 24", A = 96", B = 12", P = 10560#, x = 12", y = 0.0

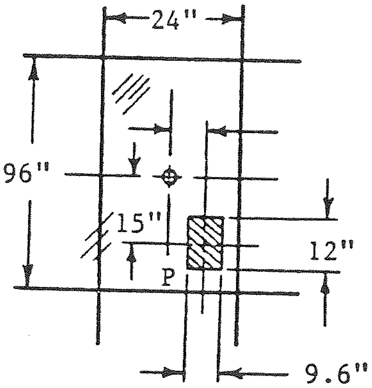
term, n	B_n	$P_n = \frac{PB_n}{B}$ (+ or -)	A/b	B_n/b (always +)	sum factor f	γ_n (always -)	$P_n \gamma_n f$
1	x + B/2 = 18	15840	4.0	0.75	+1.0	-0.01571	-248.8
2	x - B/2 = 6	5280	4.0	0.25	-1.0	-0.01127	+59.5
						Σ=	-189.3

Based on worksheet solution, M = -189.3 in=lb

Configuration A) governs for edge moment and exceeds maximum midspan bending moment and results in a plate bending stress

$= 6M/t^2 = 6 \times (-374.4)/(0.375^2) = 15970$ psi

Example 4; Fully asymmetric patch load; find midspan response and edge bending moment.



P = 2500 lbs
a = 96", b = 24"
A = 12", B = 9.6"
A/b = 0.5, B/b = 0.4
plate thickness - 0.3125"

Midspan respone; Use Fiqure 5.5 and Worksheet 1.

Title: EXAMPLE 4

a = 96", b = 24", A = 12", B = 9.6", P = 2500#, x = 6", y = 15"

term, n	A_n	B_n	$P_n = \frac{PA_nB_n}{4AB}$ (+ or -)	A_n/b (always +)	B_n/b (always +)	sum factor, f	α_n (always +)	β_n (always +)	$P_n\alpha_n f$	$P_n\beta_n f$
1	2y + A = 42"	2x + B = 21.6"	4921.9	1.75	0.9	+1.0	0.00389	0.04465	19.146	219.8
2	2y - A = 18"	2x + B = 21.6"	2109.4	0.75	0.9	-1.0	0.00634	0.07774	-13.374	-164.0
3	2y + A = 42"	2x - B = 2.4"	546.9	1.75	0.1	-1.0	0.00600	0.08990	-3.281	-49.2
4	2y - A = 18"	2x - B = 2.4"	234.4	0.75	0.1	+1.0	0.00991	0.16773	2.323	39.3
								Σ =	4.814	45.9

M = midspan bending moment = βP = 45.9 in-lb (from worksheet)

= midspan bending stress = $6M/t^2 = 6 \times 45.9/(0.3125^2) = 2822$ psi

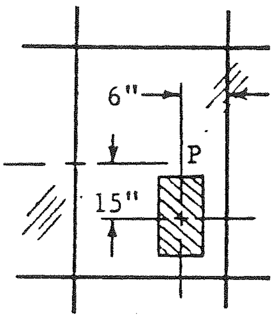
Edge Bending Response; right edge and left edge; use Figure 6.5 and Worksheet 4.

Right edge;

WORKSHEET 4; General calculations for edge bending moment

Title: EXAMPLE 4, RIGHT SIDE

a = 96", b = 24", A = 12", B = 9.6", P = 2500#, x = 6", y = 15"



term, n	A_n	B_n	$P_n = \frac{PA_nB_n}{2AB}$ (+ or -)	A_n/b (always +)	B_n/b (always +)	sum factor f	γ_n (always -)	$P_n\gamma_n f$
1	2y + A = 42"	x + B/2 = 10.8"	4921.9#	1.75	0.45	+1.0	-0.00986	-48.53
2	2y - A = 18"	x + B/2 = 10.8"	2109.4	0.75	0.45	-1.0	-0.01479	+31.20
3	2y + A = 42"	x - B/2 = 1.2"	546.9	1.75	0.05	-1.0	-0.01083	+5.92
4	2y - A = 18"	x + B/2 = 1.2"	234.4	0.75	0.05	+1.0	-0.01631	-3.82
							Σ=	-15.23

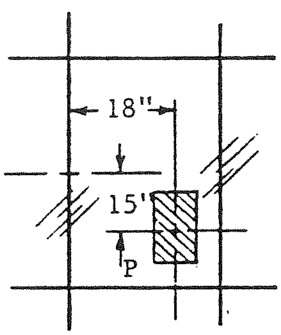
From worksheet, M = γP = -15.23 in-lb

Left edge;

WORKSHEET 4; General calculations for edge bending moment

Title: LEFT SIDE

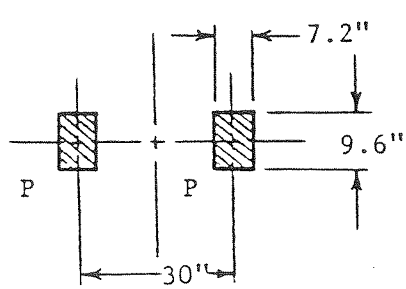
a = 96", b = 24", A = 12", B = 9.6", P = 2500#, x = 18", y = 15"



term, n	A_n	B_n	$P_n = \frac{PA_nB_n}{2AB}$ (+ or -)	A_n/b (always +)	B_n/b (always +)	sum factor f	γ_n (always -)	$P_n\gamma_n f$
1	2y + A = 42	x + B/2 = 22.8	10391	1.75	0.95	+1.0	-0.03154	-327.7
2	2y - A = 18	x + B/2 = 22.8	4453	0.75	0.95	-1.0	-0.06098	+271.5
3	2y + A = 42	x - B/2 = 13.2	6016	1.75	0.55	-1.0	-0.03741	+225.0
4	2y - A = 18	x + B/2 = 13.2	2578	0.75	0.55	+1.0	-0.07510	-193.6
							Σ=	-24.8

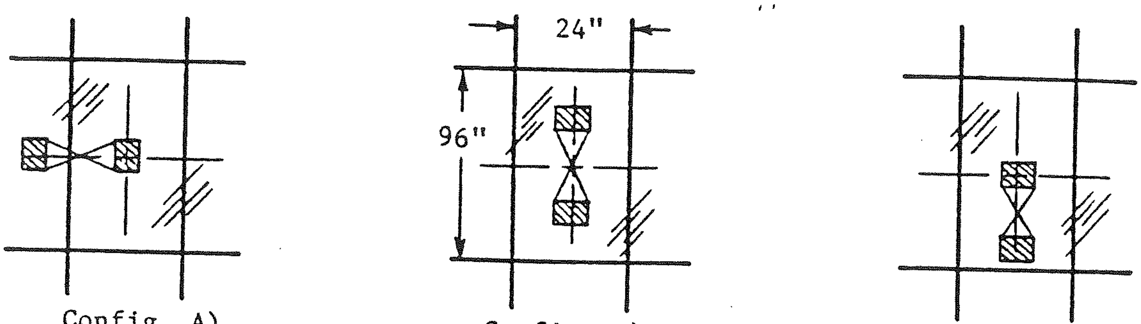
From worksheet, M = γP = -24.8 in-lb

EXAMPLE 5; for the following balanced dual wheel load, find the maximum midspan bending moment:



P = 2000 lbs per tire

It is necessary to solve by trial and error for several possible alignments of the wheel set. The following three alignments are considered for plating thickness = 0.25":



Config. A) Config. B) Config. C)

Config. A) a = 96", b = 24", A = 9.6", B = 7.2", A/b = 0.4 B/b = 0.3

a) for the central patch load, using Figure 5.5
$M_1 = \beta P = 0.18205 \times 2000 = 364.1$ in-lb

b) for the outer patch load, use Figure 5.5 and Worksheet 2;

WORKSHEET 2; General calculations for midspan response when y = 0.0 (longitudinally symmetric)

Title: EX. 5; CONFIG. A; OUTER PATCH

a = 96", b = 24", A = 9.6", B = 7.2", P = 2000#, x = 30", y = 0.0

term, n	B_n	$P_n = \frac{PB_n}{2B}$ (+ or -)	A/b	B_n/b (always +)	sum factor, f	α_n (always +)	β_n (always +)	$P_n \alpha_n f$	$P_n \beta_n f$
1	2x + B = 67.2	9333.3	0.4	2.8	+1.0	–	0.0205	–	191.3
2	2x − B = 52.8	7333.3	0.4	2.2	-1.0	–	0.0291	–	-213.4
							Σ =	–	-22.1

Based on the worksheet, $M_2 = -22.1$ in-lb

Total bending moment, $M = M_1 + M_2 = 364.1 - 22.1 = 342.0$ in-lb

Config. B); Apply symmetry, solve for one patch and multiply solution by factor of two. Use Figure 5.5 and Worksheet 3.

WORKSHEET 3: General calculations for midspan response when x = 0.0 (transversely symmetric)

Title: EXAMPLE 5; CONFIG. B

a = 96", b = 24", A = 7.2", B = 9.6", P = 2000#, y = 15", x = 0.0

term, n	A_n	$P_n = \frac{PA_n}{2A}$ (+ or −)	A_n/b (always +)	B/b	sum factor, f	α_n (always +)	β_n (always +)	$P_n \alpha_n f$	$P_n \beta_n f$
1	2y + A = 37.2	5166.7	1.55	0.4	+1.0	–	0.07749	–	+400.4
2	2y − A = 22.8	3166.7	0.96	0.4	−1.0	–	0.10998	–	−348.3
							Σ =	–	+52.1

Based on worksheet, M = 52.1 x 2 = 104.2 in-lb

Config. C) a = 96", b = 24", A = 7.2", B = 9.6", A/b = 0.3, B/b = 0.4, P = 2000 lbs

Using Figure 5.5 for central patch load, $M_1 = \beta P = 0.17505 \times 2000 = 340.1$ in-lb

For the outer patch load, using Figure 5.5 and Worksheet 3,

WORKSHEET 3: General calculations for midspan response when x = 0.0 (transversely symmetric)

Title: EXAMPLE 5; CONFIG. C; OUTER

a = 96", b = 24", A = 7.2", B = 9.6", P = 2000#, y = 30", x = 0.0

term, n	A_n	$P_n = \frac{PA_n}{2A}$ (+ or −)	A_n/b (always +)	B/b	sum factor, f	α_n (always +)	β_n (always +)	$P_n \alpha_n f$	$P_n \beta_n f$
1	2y + A = 67.2	9333.3	2.8	0.4	+1.0	0.00358	0.04501	33.413	420.1
2	2y − A = 52.8	7333.3	2.2	0.4	−1.0	0.00449	0.05690	−32.927	−417.3
							Σ =	0.486	2.8

Based on the worksheet, $M_2 = 2.8$ in-lb

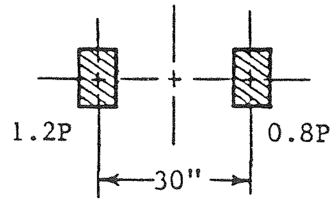
Total midspan moment, $M = M_1 + M_2 = 350.1 + 2.8 = 352.9$ in-lb

Note that the outer patch contributes less than 1 percent to the total moment.

σ = midspan bending stress = $6M/t^2 = 6 \times 352.9/0.25^2 = 33880$ psi

Note that bending stress level is high. Plate thickness increase may be necessary.

Example 6; For dual wheel load of Example 5, apply ship motion loads to produce following imbalanced dual load condition:



P = 2000 lbs

The solution of Example 5 may be utilized. The response will vary proportionately with the load. Solution is found for the same three configurations as in Example 5.

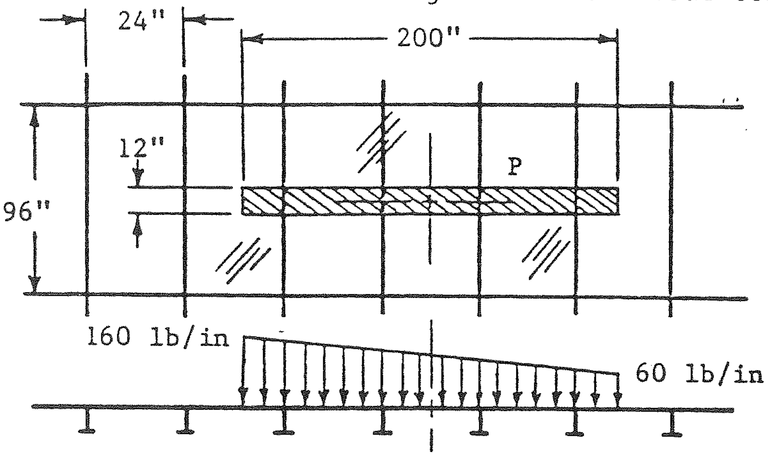
Config. A); $M = 1.2 \times 364.1 + 0.8 \times (-22.1) = 419.2$ in-lb

Config. B); $M = 1.2 \times 52.1 + 0.8 \times 52.1 = 104.2$ in-lb

Config. C); $M = 1.2 \times 350.1 + 0.8 \times 2.8 = 422.4$ in-lb

σ = midspan bending stress = $6 \times 422.4/(0.25^2) = 40550$ psi

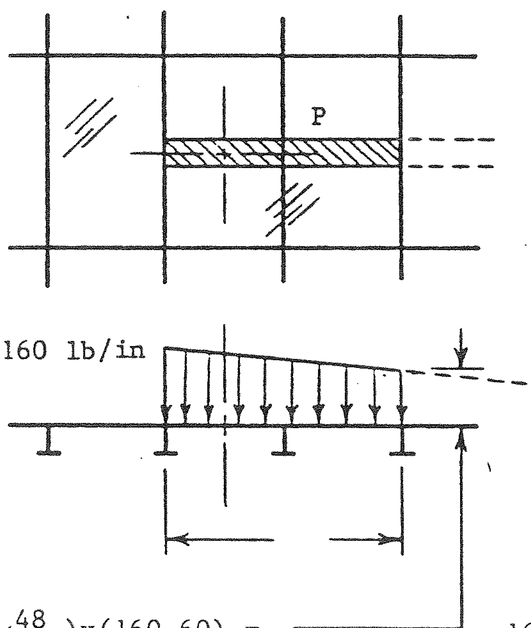
Example 7; Apply sea-motions forces to the tank-tread loading of Example 3) to obtain the following non-uniform load condition:

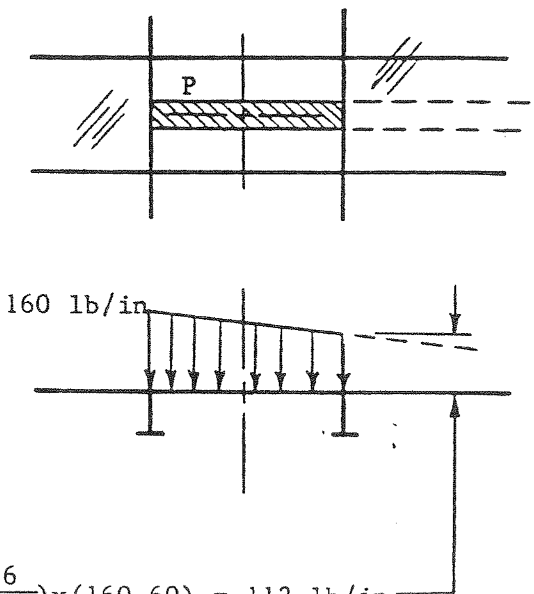


P = non-uniform 22000 lbs

a = 96", b = 24"

Midspan Response; Based on Example 3, consider the following two load configurations:

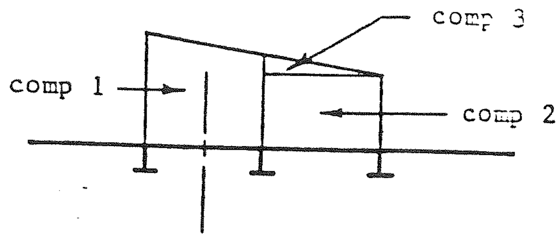




$160-(\frac{48}{200})\times(160-60) =$ 136 lb/in

$160-(\frac{96}{200})\times(160-60) =$ 112 lb/in

Config. A); Load patch must be divided into components, solved separately and summed.



Components 1) and 2) may be solved directly. Component 3) is assymmetric and non-uniform hence must be approximately solved.

Component 1; a = 96", b = 24", A = 12", B = 24", A/b = 0.5, B/b = 1.0
Total load P = 48 x (160 - 136)/4) = 7392 lbs

Applying superposition, symmetric, non-uniform loads give same midspan response as uniform loads of same patch shape and load magnitude. Therefore, solve for this component response directly from Figure 5.5.

$M_1 = \beta P = 0.08189 \times 7392 = 605.3$ in-lb

Component 2; Apply Figure 5.5 and Worksheet 2 with P = 136 x 48 = 6528 lbs

WORKSHEET 2; General calculations for midspan response when y = 0.0 (longitudinally symmetric)

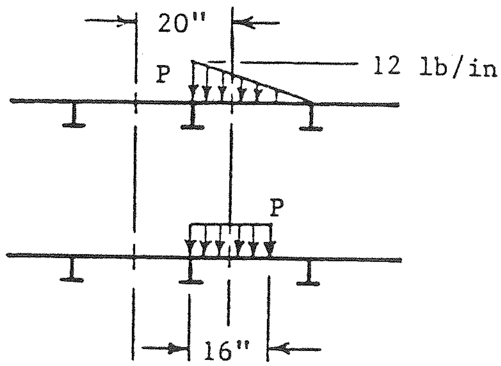
Title: EX. 7; CONFIG. A; COMP. 2

a = 96", b = 24", A = 12", B = 24", P = 6528#, x = 24", y = 0.0

term, n	B_n	$P_n = \frac{PB_n}{2B}$ (+ or −)	A/b	B_n/b (always +)	sum factor, f	α_n (always +)	β_n (always +)	$P_n\alpha_n f$	$P_n\beta_n f$
1	2x + B = 72"	9792#	0.5	3.0	+1.0	0.00087	0.0177	8.519	173.3
2	2x − B = 24"	3264#	0.5	1.0	-1.0	0.00646	0.0818	-21.085	-267.0
							Σ =	-12.566	- 93.7

$M_2 = -93.7$ in-lb

Component 3; Because component 3 patch is non-uniform and assymmetric, it must be approximated by a uniform patch with the same centroid and same load magnitude.



total P = 12 / 2 x 488
= 288 lbs

Approx. uniform load has A = 12", B = 16", P = 288 lbs

Solve approximate load using Figure 5.5 and Worksheet 2.

WORKSHEET 2; General calculations for midspan response when y = 0.0 (longitudinally symmetric)

Title: APPROXIMATED COMP. 3

a = 96", b = 24", A = 12", B = 16", P = 288#, x = 20, y = 0.0

term, n	B_n	$P_n = \frac{PB_n}{2B}$ (+ or −)	A/b	B_n/b (always +)	sum factor, f	α_n (always +)	β_n (always +)	$P_n \alpha_n f$	$P_n \beta_n f$
1	2x + B = 56"	504#	0.5	2.333	+1.0	0.00142	0.0250	0.716	12.6
2	2x − B = 24"	216#	0.5	1.0	-1.0	0.00646	0.08189	-1.395	-17.7
							$\Sigma=$	-0.679	-5.1

M_3 = -5.1 in-lb

Total bending moment, M = $M_1 + M_2 + M_3$ = 605.3 - 93.7 - 5.1 = 506.4 in-lb

Note that approximate component 3 contributes only about 1 percent of the total solution; hence the error due to approximating may be considered negligible.

Config. B); Solve directly from Figure 5.5

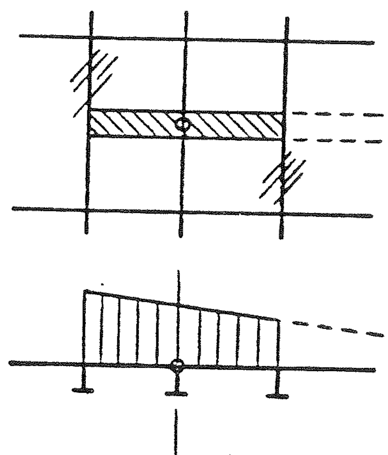
P = 96 x (160 + 112)/2 = 13056 lbs

a = 96", b = 24", A = 96", B = 12", A/b = 4.0, B/b = 0.5

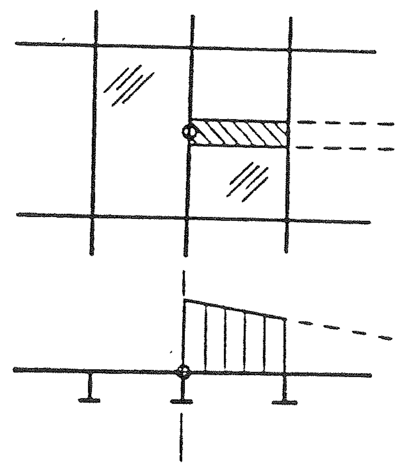
$M = \beta P = 0.02886 \times 13056 = 376.8$ in-lb

It is probably that shifting this load could significantly increase the midspan response. It must be solved by trial and error and will not be done in this example for expediency.

Edge Response; Based on Example 3, the most severe edge moment will result from either of the two following configurations or a transitional configuration between them:

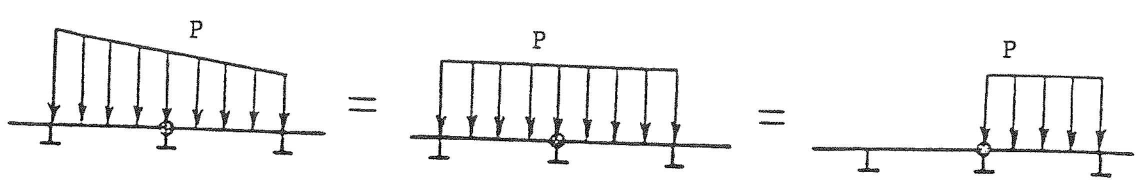


Config. A)



Config. B)

Config. A); Solution is performed using the following uniform load configuration which has the same edge response as the non-uniform load based on superposition:

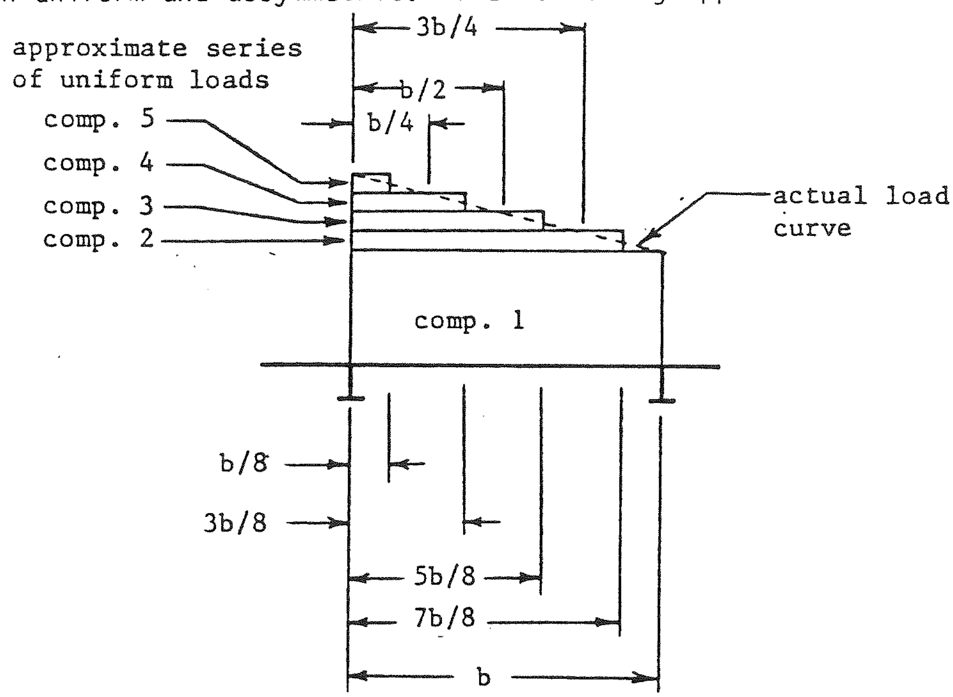


$P = (160 + 136)/2 \times 48 = 7104$ lbs
$a = 96''$, $b = 24''$, $A = 12''$, $B = 24''$, $A/b = 0.5$, $B/b = 1.0$

Solve using Figure 5.5

$M = \gamma P = -0.07091 \times 7104 = -503.7$ in-lb

Config. B); Solution is approximate because this load patch is both non-uniform and assymmetric. The following approximation is used:



Solution is based on Figure/Table A17 from Appendix A.

comp.	a	b	A	B	A/b	B/b	P	γ	P
1	96"	24"	12"	24"	0.5	1.000	3552	-0.07091	-251.9
2	96"	24"	12"	21"	0.5	0.875	63	-0.07632	-4.8
3	96"	24"	12"	15"	0.5	0.625	45	-0.09299	-4.2
4	96"	24"	12"	9"	0.5	0.375	27	-0.09056	-2.4
5	96"	24"	12"	3"	0.5	0.125	9	-0.04858	-0.4
								sum =	-263.7

From work table M = -263.7 in-lb

Configuration A) clearly provides a higher bending moment; however, a transitional load configuration between these two may cause a slightly higher bending moment. This will not be investigated further within this example for expediency.

Example 8. DESIGN EXAMPLE

Determine wheel loads for M54 5-ton truck with following ship motion load factors:

η_x = 0.358
η_y = 0.863
η_z = 1.780
W_m = Nominal vehicle weight = 41100 lbs

1. Ship Motion Forces

Case	Alignment	F_d, Vertical Load	F_l, Lengthwise Load	F_t Lateral Load
1	Longitudinal	$\eta_z \times W_m = 73160$	$\eta_x \times W_m = 14710$	$\eta_y \times W_m = 35470$
2	Longitudinal	73160	-14710	35470
3	Transverse	73160	$\eta_y \times W_m = 35470$	$\eta_x \times W_m = 14710$
4	Transverse	73160	-35470	14710

2. Nominal Vehicle Load Balance

R_m = 41100/2 = 20550 lbs
R_f' = 20550 x (1 - 135.9/179) = 4950 lbs
R_r' = 20550 - 4950 = 15600 lbs

Case 1

Lateral Balance

R_2 = 73160/2 + 35470 x 40.5/72 = 56532 lbs
R_1 = 73160 - 56532 = 16628 lbs
R_1 > 0 ; therefore, solution valid, tie-down not req'd

Lengthwise Balance

a. Left side of truck

R_v = R_1 16628
R_r = (16628 x 135.9 + 14710/2 x 40.5)/179 = 14288 lbs
R_f = 16628 - 14288 = 2340
R_f > 0

b. Right side of truck

R_v = R_2 = 56532
R_r = (56532 x 135.9 + 14710/2 x 40.5) / 179 = 43900 lbs
R_f = (56532 - 43900 = 12632 lbs
r_f > 0; therefore, solution valid

Case 2

Lateral Balance

R_2 = 73160/2 + 35470 x 40.5/72 = 56532 lbs
R_1 = 73160 - 56532 = 16628 lbs
R_1 > 0; solution valid.

Lengthwise Balance

a. Left side of truck

R_v = R_1 = 16628 lbs
R_r = (16628 x 135.9 - 14710/2 x 40.5)/179 = 10960 lbs
R_1 = 16628 - 10960 = 5668 lbs
R_f > 0; solution valid

b. Right side of truck

R_v = 56532 lbs
R_r = (56532 x 135.9 - 14710/2 x 40.5)/179 = 41256 lbs
R_f = 56532 - 41256 = 15276
R_f > 0; solution valid

Case 3

Lateral Balance

R_2 = 73160/2 + 14710 x 40.5/72 = 44854 lbs
R_1 = 73160 - 44854 = 28306 lbs
R_1 > 0; solution valid

Lengthwise Balance

a. Left side of truck

R_v = R_1 = 28306 lbs
R_r = (28306 x 135.9 + 35470/2 x 40.5)/179 = 25503 lbs
R_f = 28306 - 25503 = 2803 lbs
R_f > 0; solution valid

b. Right side of truck

R_v = R_2 = 44854
R_r = (44854 x 135.9 + 35470/2 x 40.5)/179 = 38067 lbs
R_f = 44854 - 38067 = 6787 lbs
R_f > 0; solution valid

Case 4

Lateral Balance

R_2 = 73160/2 + 14710 x 40.5/72 = 44854 lbs
R_1 = 73160 - 44854 = 28306 lbs
R_1 > 0; solution valid

Lengthwise Balance

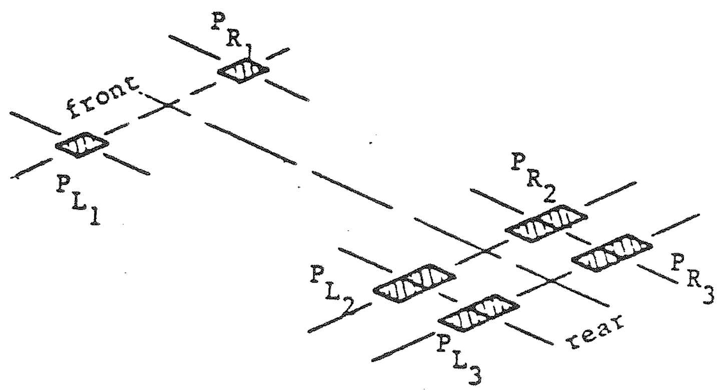
a. Left side of truck

$R_v = R_1 = 28306$ lbs
$R_r = (28306 \times 135.9 - 35470/2 \times 40.5)/179 = 17478$ lbs
$Rf = 28306 - 17478 = 10828$ lbs
$R_f > 0$; solution valid

b. Right side of truck

$R_v = R_2 = 44854$ lbs
$R_r = (44854 \times 135.9 - 35470/2 \times 40.5)/179 = 30041$ lbs
$R_f = 44854 - 30041 = 14813$ lbs
$R_f > 0$; solution valid

SUMMARY OF RESULTANT TIRE PATCH LOADS FOR M54 5-TON TRUCK



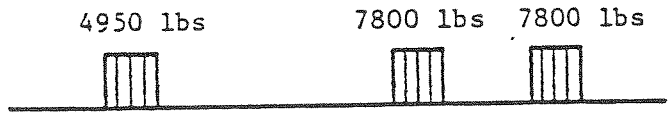
LOAD CASE	VEHICLE ALIGNMENT	Left side Loads (lbs)			Right side loads (lbs)		
		P_{l_1}	P_{l_2}	P_{l_3}	P_{r_1}	P_{r_2}	P_{r_3}
1	Longitudinal	2340	7144	7144	12632	21950	21950
2	Longitudinal	5668	5480	5480	15276	20628	20628
3	Transverse	2803	12752	12752	6787	19034	19034
4	Transverse	10828	8739	8739	14813	15021	15021
Nominal		4950	7800	7800	4950	7800	7800

Responses of Stiffeners:

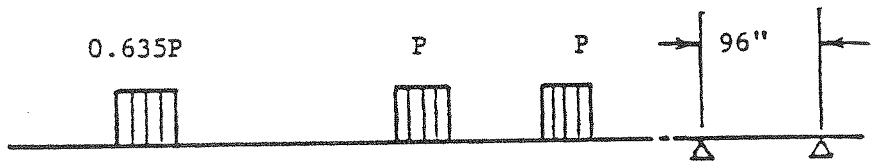
Stiffener bending moment: Find the maximum stiffener bending moment resulting from an M54 5-ton truck aligned longitudinally. Use ship motion vehicle loadings from load case 2.

Using influence lines, $M = \beta \times s \times P$
where,
M = bending moment
β = influence ordinate
s = frame spancing = 96"
P = load
b = stiffener spacing = 24"
B = Patch load width = 24" for rear wheels
= 12" for front wheels
A = Patch load length = 10"

Nominal Loading;



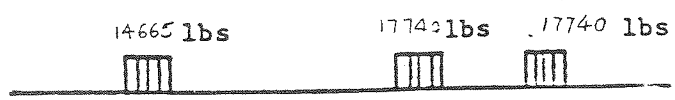
Nominal Load ratio;



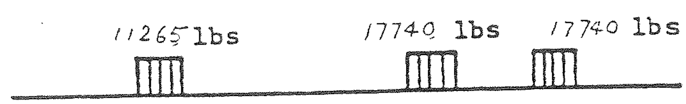
Wheel Load Reduction

Wheel	B/b	ϕ Factor	Wheel Load	Stiffener Load
front	0.5	0.96	15275	14665
rear	1.0	0.86	20628	17740

Required ship motion Loading;

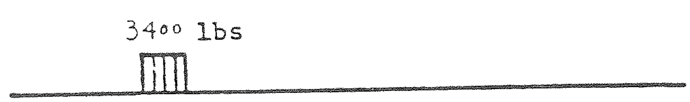


Equivalent ship motion loading;
Load with nominal load ratio;



+

Single patch load;



Moment at Support:

The vehicle load influence ordinate for the load with nominal load ratio is

$$\beta_1 = 0.32 - 0.38 = -0.06$$

which is located 1.9 frame spacings to left of support.

The patch length to frame spacing ratio, B/b is

$$2d/s = 2 \times 5'' / 96'' = 0.104$$

The patch load influence ordinate for the single patch load is

$$\beta_2 = 0.186 - 0.180 = 0.006$$

The total bending moment is $M = \beta_1 \times s \times P_1 + \beta_2 \times s \times P_2$

$$M = 267.3 \text{ in-kips}$$

Moment at Midspan:

The vehicle load influence ordinate for load with nominal load ratio is

$$\beta_1 = 0.06$$

The patch load influence for single patch load is

$$\beta_2 = .003$$

The total bending moment is

$$M = 253.2 \text{ in-kips}$$

Follow the procedures in Appendix E to complete stiffener analysis.

Responses of Plating:

Determination of transverse unit midspan bending moment for panel plating.

For the M54 5-ton truck, using stiffener spacing, b = 24", loading, and load patch sizes for previous examples, find midspan bending moment for panel plating.

Patch	A (in.)	B (in.)	Load (lbs)
right front	10"	12"	15276
right rear	10"	12"	20628

For front patch; (Fig. 5.5)

$a/b = 4$
$A/b = 10/24 = 0.417$
$B/b = 12/24 = 0.50$
$\beta = 0.144$
Unit bending moment = 15276 x 0.144 = 2200 in-lb/in

For rear patch; (Fig. 5.5)

$a/b = 4$
$A/b = 10/24 = 0.417$
$B/b = 24/24 = 1.0$
$\beta = 0.086$

Unit bending moment = 20628 x 0.086 = 1774 in-lb/in

Therefore, critical bending moment is 2200 in-lb/in resulting from front patch. Then decide the deck function coefficient, C_0, from Table IV and follow the procedures in Appendix D to complete plating analysis.

APPENDIX H

SELECTED MEASUREMENT UNITS AND CONVERSION FACTORS

SELECTED SI CONVERSION FACTORS

Category	To Convert From Inch Pount Units	To SI Units	Multiply by
Length:	foot (ft)	meter (m)	0.3048
	inch (in)	meter (m)	2.540×10^{-2}
	inch (in)	mm	25.4
Area:	$foot^2$ (ft^2)	$meter^2$ (m^2)	9.290×10^{-2}
	$inch^2$ (in^2)	mm^2	6.542×10^{3}
Force:	kip	newton (N)	4.448×10^{3}
	pound-force (lbf)	newton (N)	4.448
Mass:	pound (lb)	kilogram (kg)	.454
	ton (long, 2240 lb)	metric ton	1.016
Stress (Force/ Area):	$kip/inch^2$ (ksi)	pascal (Pa)	6.895×10^{6}
	lbf/in^2 (lb/in^2)	pascal (Pa)	6.895×10^{3}

SI makes extensive use of prefixes to form decimal multiples; it officially establishes 16 prefixes. Those 5 prefixes most frequency used are as follows:

mega	M	1,000,000	$= 10^{6}$
kilo	k	1,000	$= 10^{3}$
centi	c	0.01	$= 10^{-2}$
milli	m	0.001	$= 10^{-3}$
micro		0.000001	$= 10^{-6}$

APPENDIX I

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BIBLIOGRAPHY

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