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Factors In Lightweight Guideway Design And Costing

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ABSTRACT

The effect of lightweight guideway design on ride quality is examined. Results for a particular vehicle/guideway system show that ride quality can be maintained while beam weight is reduced to a point where physical constraints or lateral and torsional strength limitations are reached. The effect of guideway pier spacing on vehicle ride quality is also examined. Finally, a procedure for obtaining accurate guideway cost estimates is outlined.

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1.0 INTRODUCTION

This report and companion documents [1, 2] present an approach designed to achieve integrated vehicle/guideway design. Reference 1 is concerned with estimating guideway requirements based on a vehicle concept. These preliminary estimates include the effect of guideway flexibility and roughness. The results are considered preliminary since the guideway models for both flexibility and roughness are simplistic. Reference 2 describes a full scale vehicle/guideway interaction simulation and includes detailed vehicle and guideway models. The guideway segment of the simulation employs sophisticated models for roughness and flexibility. Roughness tolerance parameters, including pier survey error, pier settlement, camber and surface finish are given in terms understandable to potential contractors. Guideway flexibility is modeled by Bernoulli-Euler beam theory and includes full vehicle inertia loading effects. Reference 2 also demonstrates the data processing ability available in the simulation by displaying vehicle cabin acceleration in several ride quality formats. A detailed example is presented which emphasizes the effects of guideway roughness and points out some roughness tolerance trade-offs available to the designer.

This report concentrates on three additional aspects of guideway design. The first concerns the methods and considerations which accompany lightweight guideway design and involves primarily guideway flexibility. The second aspect discusses the effects of pier spacing (span length) on support beam design and vehicle dynamic behavior. The third item details the steps which are required to obtain reliable guideway cost data, or more accurately, the cost of a vehicle/guideway system. No attempt is made to arrive at any cost figures in this report.

The numerical results presented herein are based on the vehicle/guideway model described in Reference 2. This model is designated in this report as the nominal design. For convenience, Table 1 lists all the principal physical properties for the vehicle/guideway design.

TABLE 1

PARAMETERS FOR REFERENCE 2 NOMINAL

VEHICLE/GUIDEWAY SYSTEM

Vehicle Cabin Weight = 50,000 lbs.	Cushion Pad Length = 10 ft.
Weight of Two Cushions = 2,500 lbs.	Secondary Heave Freq. = 1.0 Hz
Vehicle Suspension Base = 30 ft.	Secondary Pitch Freq. = 1.67 Hz
Pier Spacing ℓ = 100 ft.	EI = 1.13×10^{13} lb. in.^{2}
Span Fundamental Freq. f_1 = 3.52 Hz	Vehicle/Guideway Mass Ratio = 0.35

Nominal Maximum Pier Survey Error: D_m = 0.5 in.

Nominal Maximum Camber Tolerance Error: T_m = 20%

Nominal Maximum Pier Settlement: a = 0.75 in.

Nominal Surface Finish Profile Index: P.I. = 1.6 in./mile

2.0 LIGHTWEIGHT GUIDEWAY DESIGN

The potential cost savings to be realized through the use of lightweight guideway structures can be considerable. These savings can be realized through the obvious use of less material and perhaps also through lower costs for transporting materials and for ease of construction. As noted in Reference 2, a fundamental parameter used to describe the flexural rigidity of support beams is the modulus of rigidity EI where:

E = material elastic modulus (usually in lbs/in^2)

and I = second moment of cross-sectional area about neutral axis (in^4).

For the general beam cross-section shown in Figure 1(a),

$$I = \int_A y^2 dA \tag{1}$$

where A is the cross-sectional area and y the coordinate measured from the neutral axis. The parameter I is a measure of how the cross-sectional area is distributed about the neutral axis. The importance of the quantities E, I, and A can be seen by observing the dependence of basic guideway parameters upon their values. For example, the fundamental transverse bending frequency of a guideway span may be written:

$$f_1 = \frac{\beta_1^2}{2\pi\ell^2} \sqrt{\frac{EIg}{\gamma A}} \tag{2}$$

where β_1 = 1st normalized eigenvalue ($\beta_1 = \pi$ for simply supported spans)

ℓ = span length

g = acceleration of gravity

and γ = beam weight per unit volume

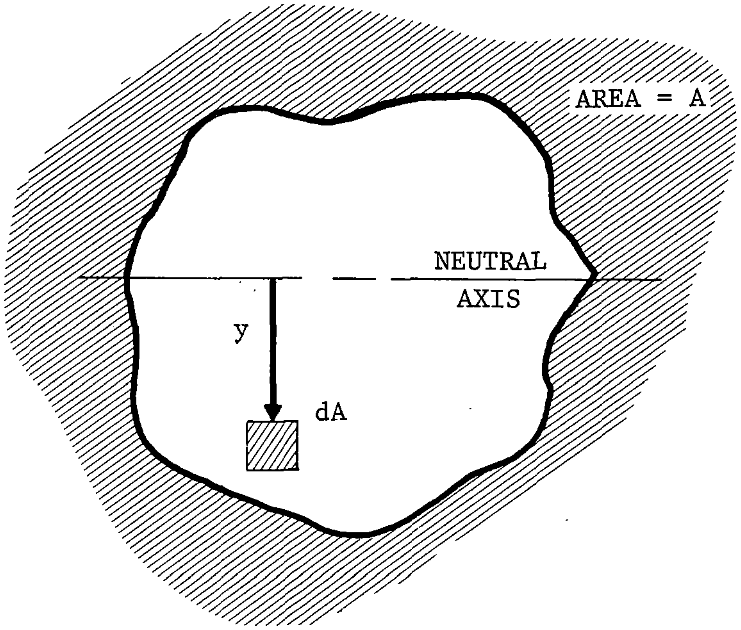

FIGURE 1(a)
GENERAL CROSS-SECTIONAL AREA

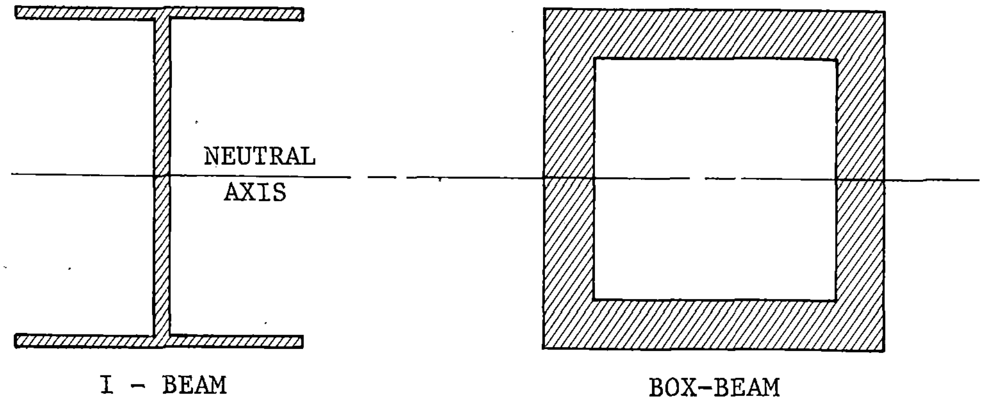

FIGURE 1(b)
BEAM CROSS SECTIONS WITH HIGH VALUES OF r

Likewise, the weight per unit length w of the support beam is given by:

$$w = \gamma A \tag{3}$$

From eq. (3), lighter-weight structures are obtained, for a given material, by reducing the cross-sectional area A. From eq. (1) however, a reduction in the value of A can alter the value of I, unless the material is redistributed. It is also obvious from eq. (2) that changes in A will effect the span fundamental frequency. These observations lead to the definition of an important parameter in the design of flexible guideway structures, the radius of gyration r:

$$r = \sqrt{I/A} \tag{4}$$

The radius of gyration defines the position from the neutral axis at which all the cross-sectional area may be assumed concentrated. Although this parameter is a physical artifice, it serves as a measure of how efficiently, for the purposes of structural rigidity, the cross-sectional area is distributed. It is clear from eqs. (2) and (4) that higher values of r yield higher fundamental frequencies (more rigid beams). For a given cross-sectional area A, eqs. (1) and (4) demonstrate that a higher value of r can be achieved by placing more material as far as possible from the neutral axis. The I-beam and box-beam, shown in Figure 1(b), are typical examples of beam designs used to achieve high values of r. There are, of course, practical drawbacks in pursuing designs which attain arbitrarily high values of r, as will be illustrated in the following example.

2.1 Example: Rectangular Cross-Section

Assume that a two-span semi-continuous support structure consists of side-by-side (twin) concrete beams of rectangular cross-sectional area as shown in Figure 2. The assumption of concrete immediately establishes the following approximate parameter values:

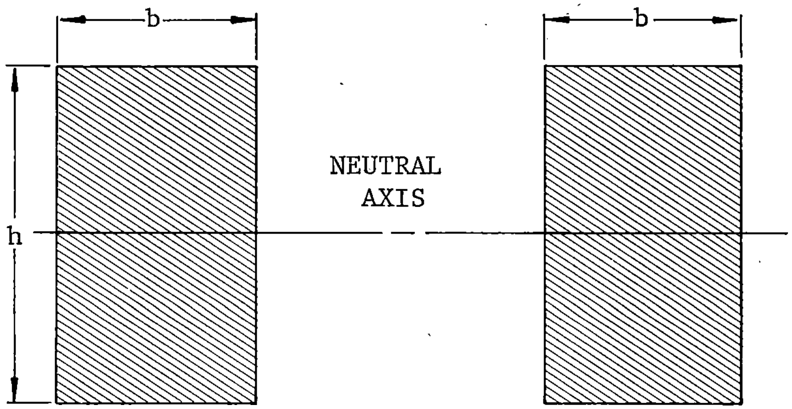

FIGURE 2
TWIN RECTANGULAR SUPPORT BEAM STRUCTURE

$$E = 5 \times 10^6 \text{ lbs/in}^2 \tag{5}$$

and $$\gamma = 0.087 \text{ lbs/in}^3 \tag{6}$$

For the twin rectangular configuration, the total cross-sectional area and the second moment of area are respectively:

$$A = 2\, bh \tag{7}$$

and $$I = 1/6\; bh^3 \tag{8}$$

and where b is the width of one beam and h is the beam depth.

From eqs. (4), (7) and (8),

$$r^2 = h^2/12 \tag{9}$$

Equation (9) is plotted in Figure 3 as a function of beam depth h in inches. From Reference 2, it is recalled that the span length was selected to be $\ell = 100$ ft. and that the nominal flexural rigidity EI was fixed by requiring that the vehicle meet the International Standards Organization (ISO) ride quality criterion at V = 300 mph; this nominal value is:

$$(EI)_{nom} = 1.13 \times 10^{13} \text{ lb.in.}^2 \tag{10}$$

In order to achieve a span fundamental frequency in the neighborhood of 3.5 Hz, eqs. (2), (6) and (10) were used (along with $\ell = 100$ ft.) to obtain the required cross-sectional area:

$$A_{nom} = 4840 \text{ in.}^2 \tag{11}$$

Thus, eqs. (5) and (10) yield:

$$I_{nom} = \frac{1.13 \times 10^{13} \text{lb.in.}^2}{5 \times 10^6 \text{lb./in.}^2} = 2.268 \times 10^6 \text{in.}^4$$

and from eq. (11),

$$r^2_{nom} = \frac{I_{nom}}{A_{nom}} = \frac{2.268 \times 10^6}{4.840 \times 10^3} = 469 \text{ in.}^2 \tag{12}$$

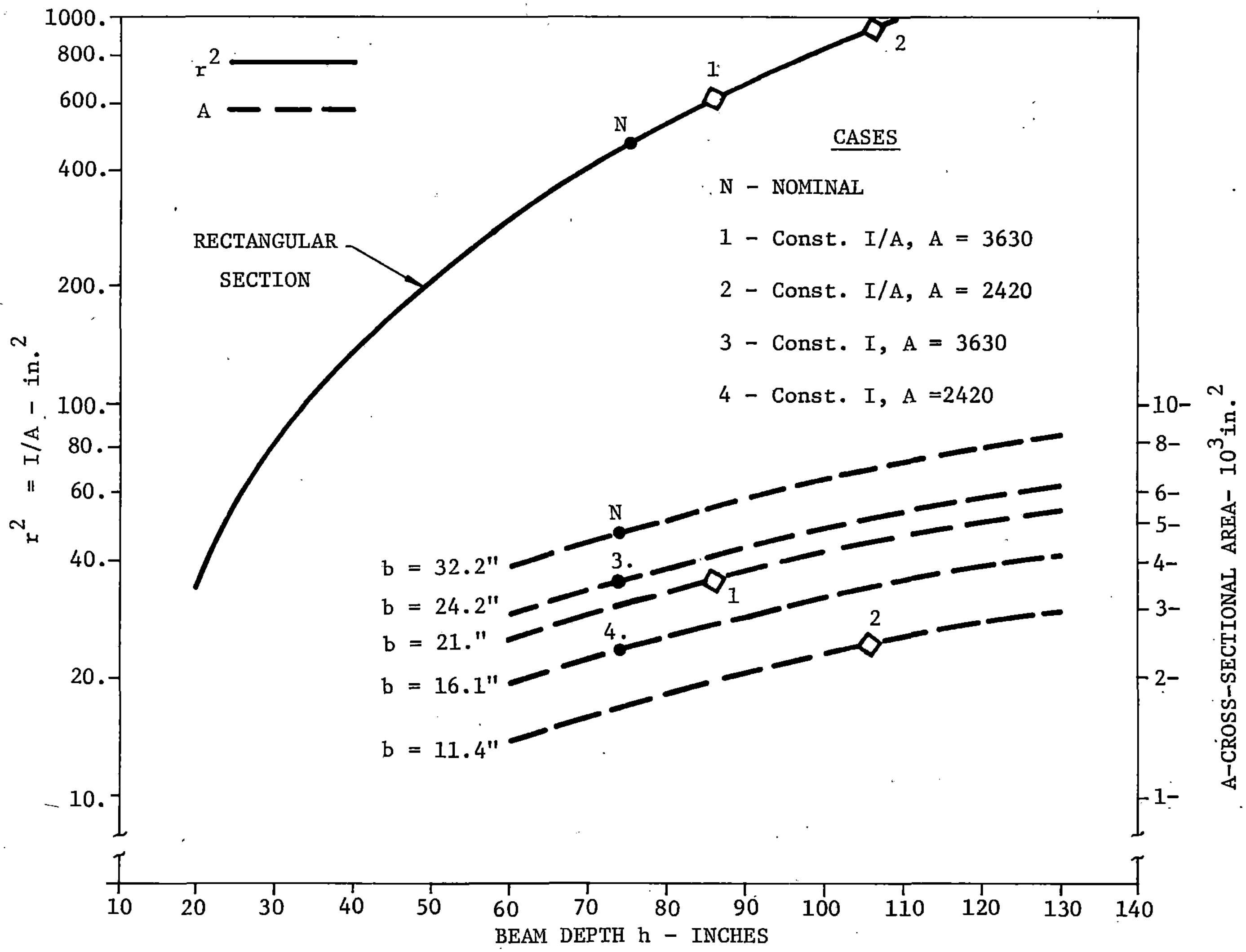

FIGURE 3
CROSS SECTION PROPERTIES FOR TWIN RECTANGULAR BEAM

This nominal value of I/A (r^2) is marked "N" on Figure 3. Note that the corresponding beam depth is 75" (6.25 ft.). Also shown on Figure 3 is the relationship between beam depth h and the cross-sectional area for the configuration of Figure 2. The nominal (Reference 2) design is also indicated on this plot with the letter "N". It is now desired to improve this nominal design by reducing the cross-sectional area, thereby decreasing the weight of the structure.

2.1.1 Preferred Weight Reduction Procedure

As previously noted, the preferred technique for reducing the weight (cross-sectional area) of the guideway support beams is to increase the radius of gyration r. Points 1 and 2 on the I/A curve in Figure 3 show, respectively, the locations of two "improved" designs for which:

$$\frac{A_1}{A_{nom}} = 0.75 \tag{13}$$

and

$$\frac{A_2}{A_{nom}} = 0.50 \tag{14}$$

Holding I constant, it follows that:

$$r_1^2 = \frac{I_{nom}}{A_1} = 625 \text{ in.}^2 \tag{15}$$

and

$$r_2^2 = \frac{I_{nom}}{A_1} = 937 \text{ in.}^2 \tag{16}$$

This approach to reducing beam weight is called the "preferred method" since it leads to a dynamically more rigid configuration, while using less material. For example, in the nominal (see Table 1) case, the ratio of vehicle pass per unit length (m_v)[1] to support beam mass per unit length (m_b) is:

---

[1]The vehicle mass per unit length for the nominal case is calculated from : $m_v = \frac{1}{g}\left(\frac{\text{Vehicle Weight}}{\text{suspension base}}\right) = \frac{1}{g}\left(\frac{52{,}500 \text{ lbs.}}{360 \text{ in.}}\right) = \frac{1}{g}(145.83 \text{ lbs/in.})$

$$M_{nom} = \left(\frac{m_v}{m_b}\right)_{nom} = \frac{m_v}{\left(\frac{\gamma A_{nom}}{g}\right)} = 0.35 \tag{17}$$

Equations (13),(14), and (17) yield:

$$M_1 = 0.47 \tag{18}$$

and

$$M_2 = 0.70 \tag{19}$$

The mass ratio M is used extensively by bridge and overpass designers; an increasing mass ratio is identified with a more efficient design (less structural material for load supported). At the same time, eqs. (2), (12), (15) and (16) yield:

$$(f_1)_1 = 1.15\ (f_1)_{nom} \tag{20}$$

and

$$(f_1)_2 = 1.41\ (f_1)_{nom} \tag{21}$$

Thus, eqs. (18) through (21) show that a lighter-weight design has been achieved while the span has been simultaneously stiffened. Examination of Table 2 indicates that Lightweight Designs 1 and 2, when tested on the full-scale vehicle/guideway simulation TRAVSIM [2], showed only minor changes (4th place) in the cabin acceleration response. Similarly, only minor variations are evident in the Urban Tracked Air Cushion Vehicle (UTACV) ride-quality comparisons shown in Figure 4 while up to half of the original weight has been saved. There are, however, two important limitations to this preferred method:

(1) As noted in the curve relating beam depth to cross-sectional area in Figure 3, Lightweight Designs 1 and 2 require beams of increasing depth and decreasing width: Design 2 ($M_2 = 0.70$) for example, requires two beams approximately 1 ft. wide and 9 ft. deep. This structure might be unacceptable due to lateral and torsional weaknesses and, ultimately, physical construction constraints.

(2) The changes in span fundamental frequency associated with increasing I/A (see eqs. (20) and (21)) must be such that they avoid

	MASS RATIO M	I/A (in^2)	BEAM WIDTH x DEPTH*	CABIN RMS ACCELERATION G's**	COMMENTS
NOMINAL	0.35	469	2.7' x 6.25	0.008	
LIGHTWEIGHT DESIGN #1	0.47	625	1.75' x 7.2'	0.008	
LIGHTWEIGHT DESIGN #2	0.70	937	0.95' x 8.8'	0.008	POSSIBLE LATERAL AND TORSIONAL WEAKNESSES
LIGHTWEIGHT DESIGN #3	0.47	469	2.0' x 6.25'	0.011	
LIGHTWEIGHT DESIGN #4	0.70	469	1.34' x 6.25'	0.016	

* For one support beam in a two-beam configuration.

** V = 150 mph, elastic deflections only.

EFFECT OF GUIDEWAY SUPPORT BEAM
CROSS-SECTION PROPERTIES
ON VEHICLE ACCELERATION

TABLE 2

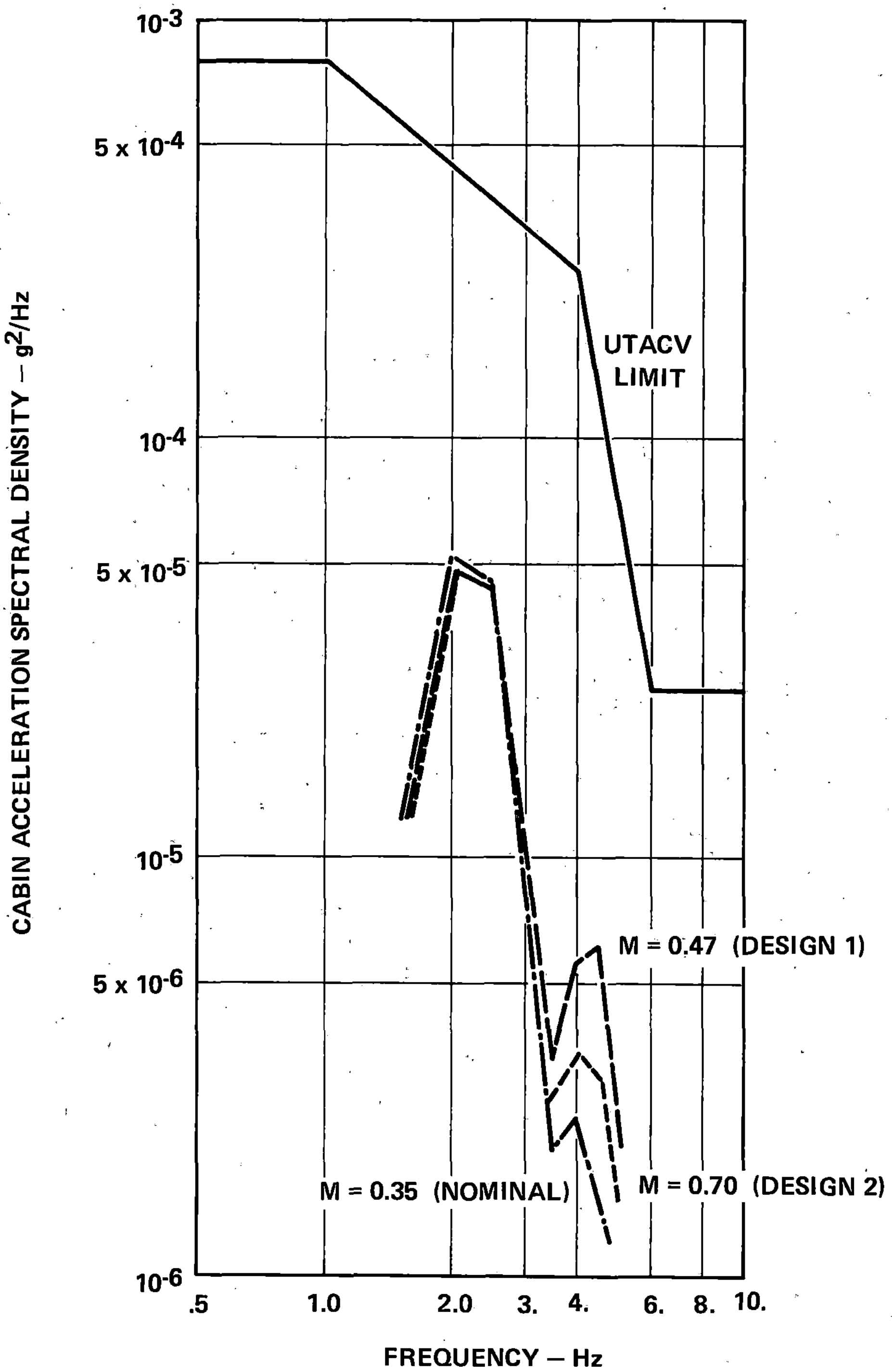

FIGURE 4
ACCELERATION SPECTRAL DENSITY DUE TO FLEXIBILITY ONLY: V = 150 mph

each of the following frequencies:

a) The vehicle primary suspension frequency

b) The vehicle secondary suspension frequencies

c) The frequency $f_{cr} = 0.75\ V/\ell$ for single span beams and $f_{cr} = 0.5\ V/\ell$ for multi-span beams.

Limitations (a) and (b) are obvious. Also, by referring to Figure (11) in Reference 2 it can be seen that span frequencies equal to f_{cr} will yield high beam dynamic amplification factors (DAFs).

The preferred method for designing lightweight structures is therefore based on distributing the cross-sectional area as efficiently as possible, thereby achieving the highest possible values of r ($\sqrt{I/A}$) within the constraints imposed by the above limitations.

2.1.2 Weight Reduction Procedure Based on Frequency Limitation

Under the restriction that the span fundamental frequency remain constant, the only way to achieve a lighter-weight structure for a given material is to reduce I and A in the same proportion; that is, holding r constant. Clearly, from eqs. (2) and (4), reducing I and A to retain r constant implies that frequency remains constant; however, since the span deflection is inversely proportional to I, a reduction of I will lead to larger deflections. This has a degrading effect on ride quality. For example, assume that the span fundamental frequency for the nominal case [2], $f_1 = 3.52$ Hz, must be maintained. Consider design points 3 and 4 where, for comparison with the preferred method,

$$M_3 = 0.47$$

and $$M_4 = 0.70$$

These points are indicated in Figure 3. The corresponding values of I and A required to keep $r^2 = 469$ and $f_1 = 3.52$ Hz, are:

$A_3 = 3630\ \text{in.}^2$, $\quad I_3 = 1.701 \times 10^6\ \text{in.}^4$

$A_4 = 2420\ \text{in.}^2$, $\quad I_4 = 1.134 \times 10^6\ \text{in.}^4$

Clearly,

$$\frac{I_{nom}}{A_{nom}} = \frac{I_3}{A_3} = \frac{I_4}{A_4} = 469 \text{ in.}^2$$

On the A vs. h curve in Figure 3 the overall dimensions for design points 3 and 4 are more favorable than the corresponding points 1 and 2. However, the effect of increased deflection due to decreased flexural ridigity is apparent in Table 2. Note the increase in cabin RMS acceleration levels over the nominal value in Designs 3 and 4. The adverse ride quality trend is also apparent in Figure 5. For the 150 mph vehicle speed, the effect of guideway flexibility alone (no roughness) is almost sufficient to cause the acceleration PSD to surpass the UTACV limit for Design 4 ($M_4 = 0.70$). In fact, the addition of the nominal roughness parameters (see Table 1) would cause the UTACV limit to be exceeded. Figures 6, 7, and 8 represent the trade-off curves developed in Reference 2. Briefly, the curves in Figures 6, 7, and 8 represent equal cabin RMS acceleration curves (in this report, .075 g's) for the nominal, Design 3 and Design 4 mass ratios. For example, Figure 8 depicts the possible trade-off between beam camber tolerance and maximum allowable pier settlement. The vehicle speed in this case is 150 mph and all other tolerances (pier survey error, surface finish, etc.) are at their nominal levels (see Table 1). Figure 8 illustrates that to hold a 0.75" maximum pier settlement, the nominal guideway (M = 0.35) would permit a 245% camber error, while lightweight Design 4 (M = 0.70) would permit only a 195% camber error. A similar pattern is apparent in Figures 6 and 7. Thus, it is clear that reducing beam weight by holding frequency constant has two major effects:

1) It has a degrading effect on ride quality
2) It restricts the trade-off latitude available to the designer

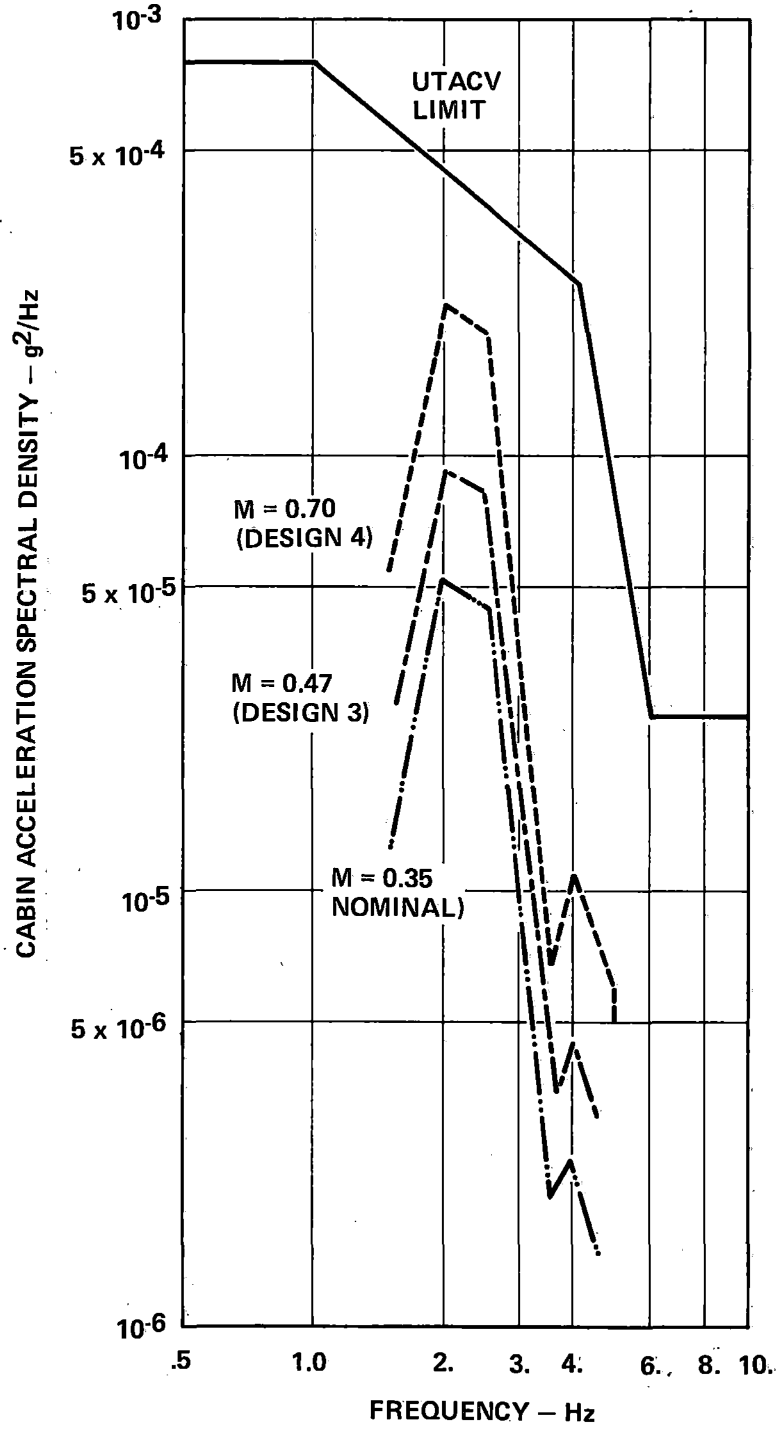

FIGURE 5
ACCELERATION SPECTRAL DENSITY DUE TO FLEXIBILITY ONLY: V = 150 mph

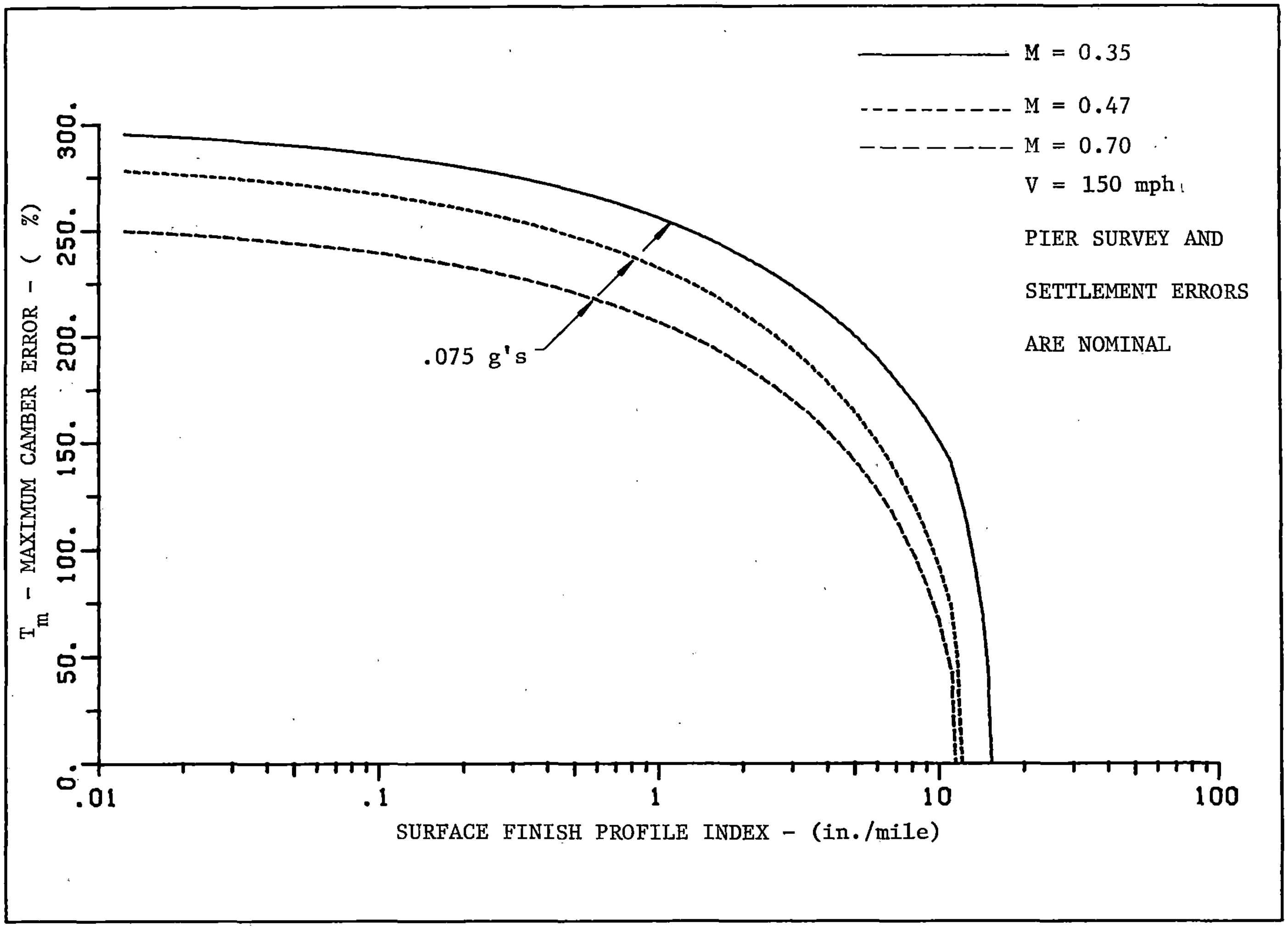

FIGURE 6
EFFECTS OF MASS RATIOS ON CONSTRUCTION TOLERANCE TRADE OFFS

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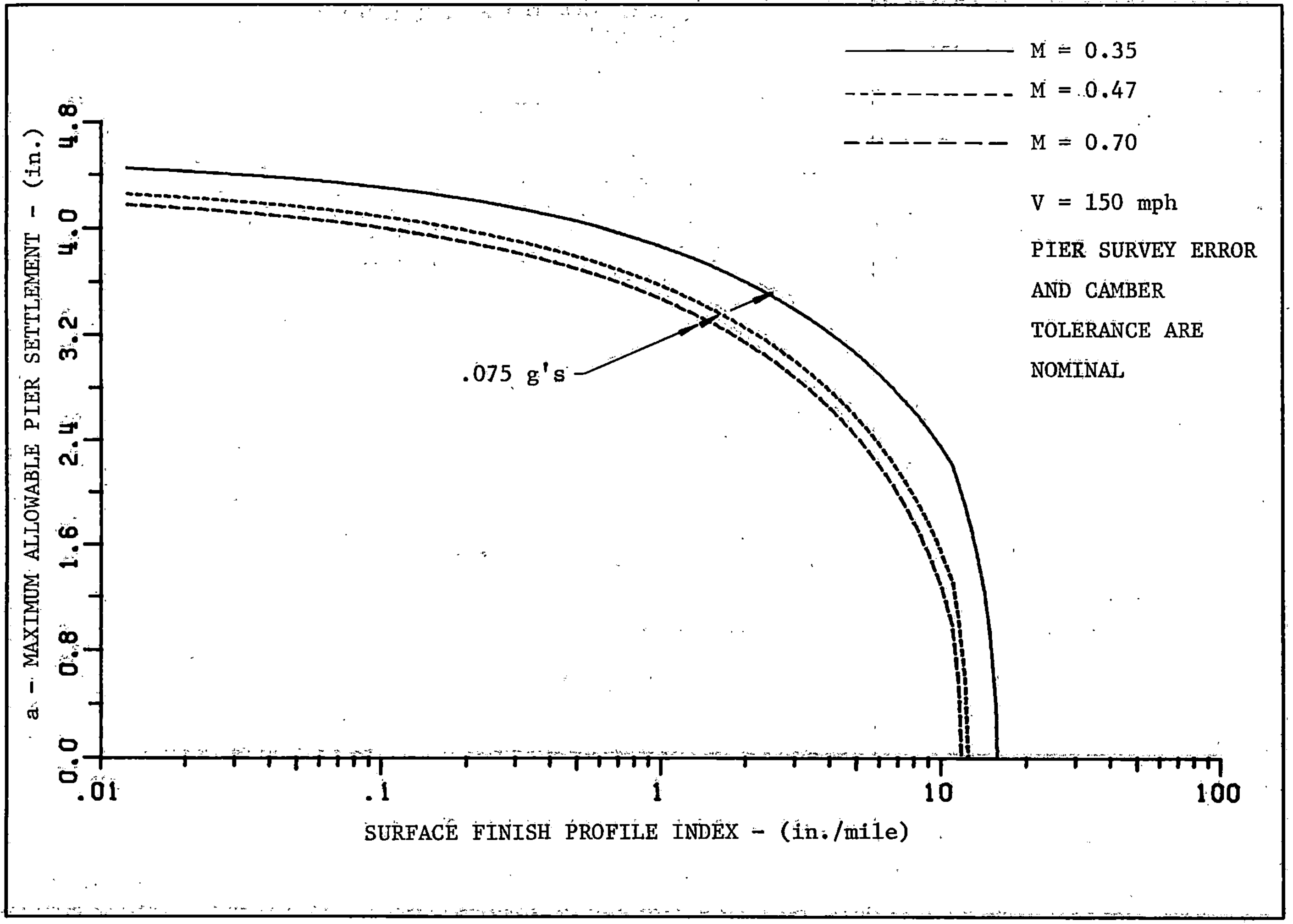

FIGURE 7
EFFECT OF MASS RATIOS ON CONSTRUCTION TOLERANCE TRADE OFFS

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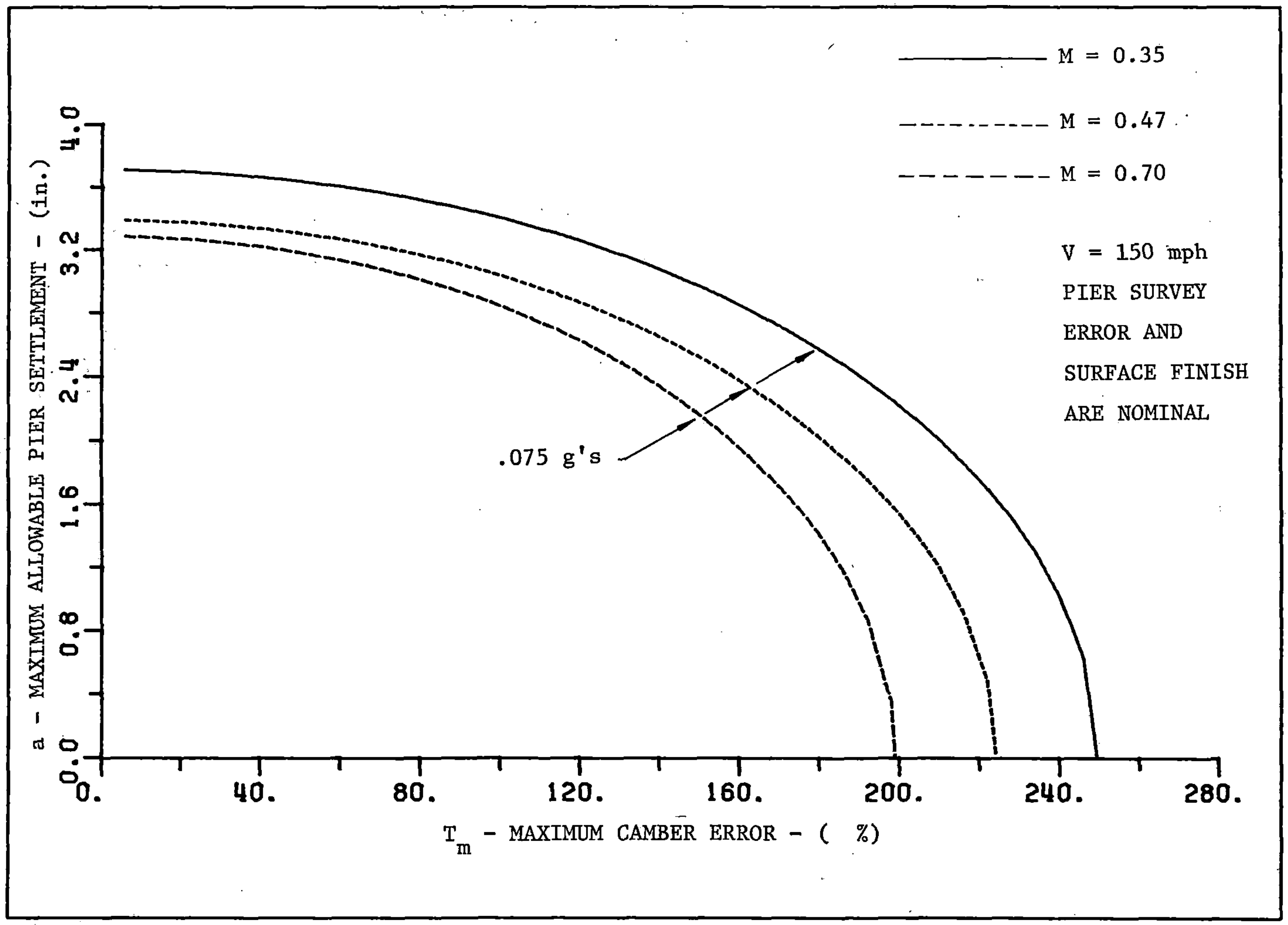

FIGURE 8
EFFECT OF MASS RATIOS ON CONSTRUCTION TOLERANCE TRADE OFFS

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2.2 Strength Considerations

It is worthwhile to note at this point that no mention has yet been made of restrictions on guideway design due to stress. For the range of pier spacings considered in this report (and also considered reasonable in practice), ride quality constraints on deflection usually control the design and the guideways are therefore overdesigned from a stress point of view. For example, a conservative estimate of stress level for Design 4, the worst case from a stress viewpoint, yields σ_{max} = 1000 psi. A 3500 psi strength for concrete can be achieved in practice with a confidence level of 0.99 [3].

3.0 PIER SPACING

It is not without reason that the usual artist's conception of an advanced ground transportation system often depicts the vehicle on very slender support beams, elevated over widely spaced, gently curved support piers. Aesthetics and public acceptance must be considered in promoting new transportation systems. Despite the importance of such subjective considerations, they are in fact beyond the scope of this report. For this study, it is important to understand that pier spacing (i.e. span length) has a significant effect on vehicle and guideway structural performance and cost. It is clear from eq. (2) that span length ℓ has a strong effect on span fundamental frequency, f_1. For a given cross-section, a longer span deflects more and has a lower fundamental frequency. The shorter spans have higher fundamental frequencies and, for the same cross-section, provide a more rigid support. A trade-off therefore develops between guideway support beam material to be saved by using shorter spans and the increased number of support piers that are required. Figure 9 illustrates the dependence of the required number of equally spaced support piers as a function of span length. A 100 ft. pier spacing requires 35 fewer piers per mile than a 60 ft. span. The economic trade-off will strongly depend on the particular system.

There also exist some important vehicle/guideway performance considerations in selecting the appropriate pier spacing. Some important parameters which depend on pier spacing ℓ are:

Transit frequency $f_t = V/\ell$ (22)

and Crossing frequency ratio $V_c = V/\ell f_1 = f_t/f_1$ (23)

where V is the vehicle velocity.

Recall that in Section 2.1, when discussing the limitations on the preferred method, it was noted that f_1 should not be in the neighborhood of the critical frequencies:

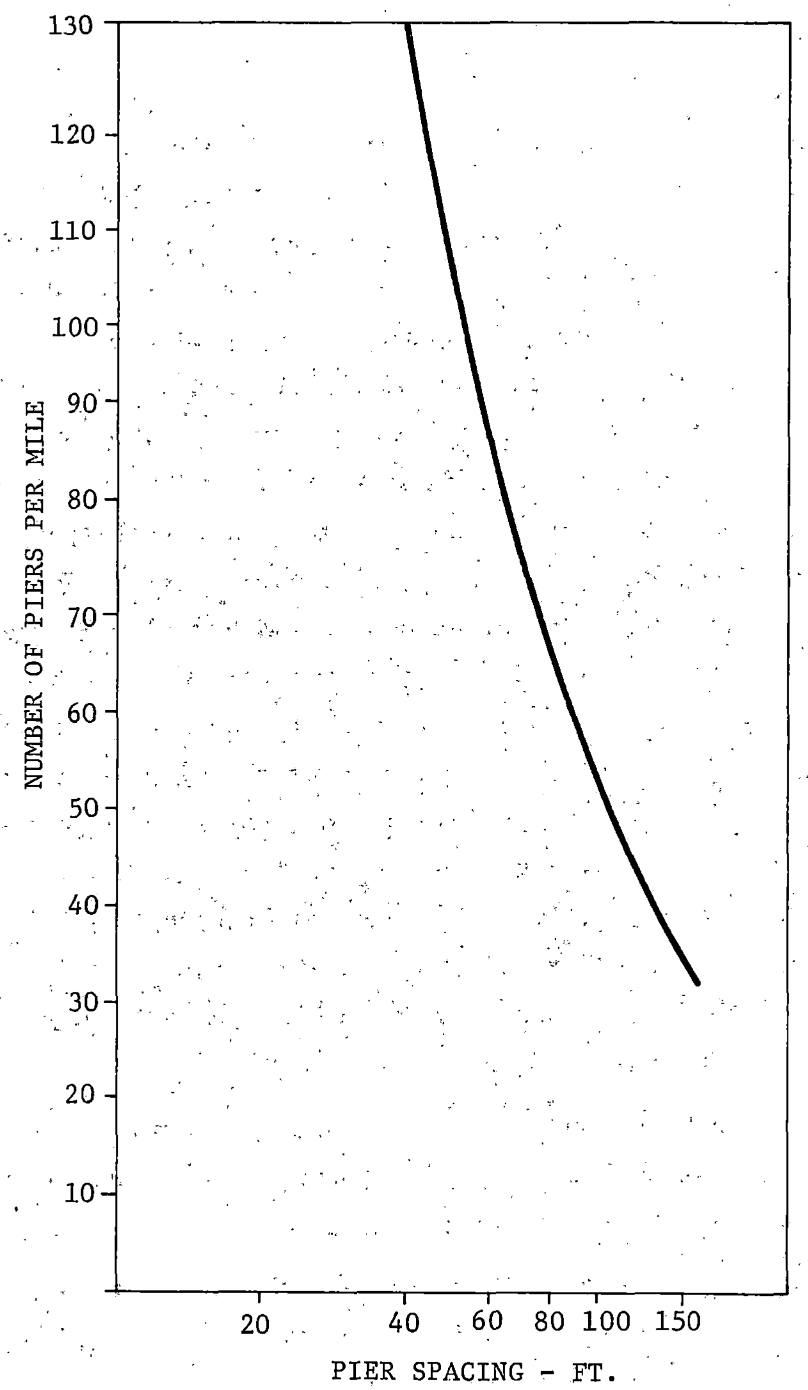

FIGURE 9
NUMBER OF REQUIRED SUPPORT PIERS AS A FUNCTION OF PIER SPACING (ASSUMED EQUALLY SPACED)

$$f_{cr} = 0.75\ V/\ell, \text{ single-span beams} \tag{24}$$

$$f_{cr} = 0.5\ V/\ell, \text{ multi-span beams} \tag{25}$$

Substituting f_{cr} for f_1 in eq. (23) yields $V_c = 1.33$ and $V_c = 2.0$ as crossing frequency ratios to be avoided for single-and multi-span beams, respectively. These conditions represent points of maximum dynamic response for the beams (see Figure 11 in Ref.2). Eqs. (2) and (23) yield:

$$V_c = \frac{VEI\ell}{\beta_1^2} \sqrt{\frac{\gamma A}{gEI}} \tag{26}$$

That is, crossing frequency ratio is directly proportional to span length and a designer must be aware of critical values of V_c when changing the span length.

Table 3 shows one aspect of the effect of varying span length on the performance of a 150 mph vehicle. This table, illustrating the vehicle response to beam flexibility only (no roughness), indicates that there is little effect (4th place) on vehicle acceleration from shortening or lengthening the span by 25 ft. about the nominal 100 ft. The reasons for this are twofold:

(1) There is no appreciable difference in dynamic amplification factors (DAFs) for two-span semi-continuous spans over the range of $0.47 \leq V_c \leq 0.78$.

(2) The larger deflections associated with the 125 ft. spans are offset by the longer wavelength; that is, the deflection slopes remain approximately constant.

An additional case for which ℓ = 145 ft. is also shown in Table 3. This length, combined with a vehicle speed of 150 mph, yields a crossing frequency ratio close to unity (V_c = 0.91). While this is not a critical ratio for two-span semi-continuous beams, a noticeable increase in the DAF does occur at this point (see Figure 11, Ref. 2) and the larger span deflections begin to seriously erode ride quality. In

	SPAN LENGTH	$\frac{\ell_s{}^*}{\ell}$	CROSSING FREQUENCY RATIO, V_c**	CABIN RMS ACCELERATION G's***	COMMENTS
NOMINAL	100 ft.	0.3	0.625	0.008	
SPAN DESIGN 1	75 ft.	0.4	0.47	0.008	
SPAN DESIGN 2	125 ft.	0.24	0.78	0.008	
SPAN DESIGN 3	145 ft.	0.21	0.91	0.017	Span Fundamental Freq., f_1 is equal to vehicle Secondary Pitch Frequency

* ℓ_s = Vehicle Suspension Base = 30 ft.

** For V = 220 ft./sec. (150 mph)

*** Beam Flexibility Only (no roughness)

EFFECT OF SPAN LENGTH
ON VEHICLE ACCELERATION

TABLE 3

addition, the span fundamental frequency in this case equals the vehicle secondary pitch frequency. Notice in Table 3 that the RMS cabin acceleration is double the value of the three other configurations. Figure 10 shows the cabin acceleration PSD for the nominal 100 ft., 75 ft. and 125 ft. span lengths. The plots are labeled according to the ratio:

$$L = \ell_s / \ell$$

where ℓ_s is defined as the suspension base length. With the nominal value of ℓ_s = 30 ft. [2],

$$L_{nom} = 0.3$$

$$L_{75'} = 0.4$$

$$L_{125'} = 0.24$$

Aside from the shifting of the major peak, which corresponds to the value of f_t, no significant changes are observed in Figure 10. The introduction of roughness does however have some interesting and perhaps unexpected effects. It might be argued that the longer span will have longer wavelength surface finish roughness input and thus give rise to increased response. Figure 11 shows, however, that the shorter span (L = 0.4) is the "worst ride"[2] in the group, according to the UTACV standard. The reason for this is that pier settlement and pier survey error were considered independent of length, and assigning the same irregularities over the shorter distances had the most severe effect on degrading ride quality. It is also interesting to observe that the nominal roughness values, combined with flexibility, push the cabin acceleration over the UTACV limit. Finally, as amply demonstrated in Reference 2, the nominal case (L = 0.3), which exceeds the UTACV limit in Figure 11, easily meets the ISO limit shown in Figure 12.

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[2] "Worst Ride" is arbitrarily defined, in the UTACV sense, as the ride with the highest overall rms acceleration level which at the same time violates the UTACV standard.

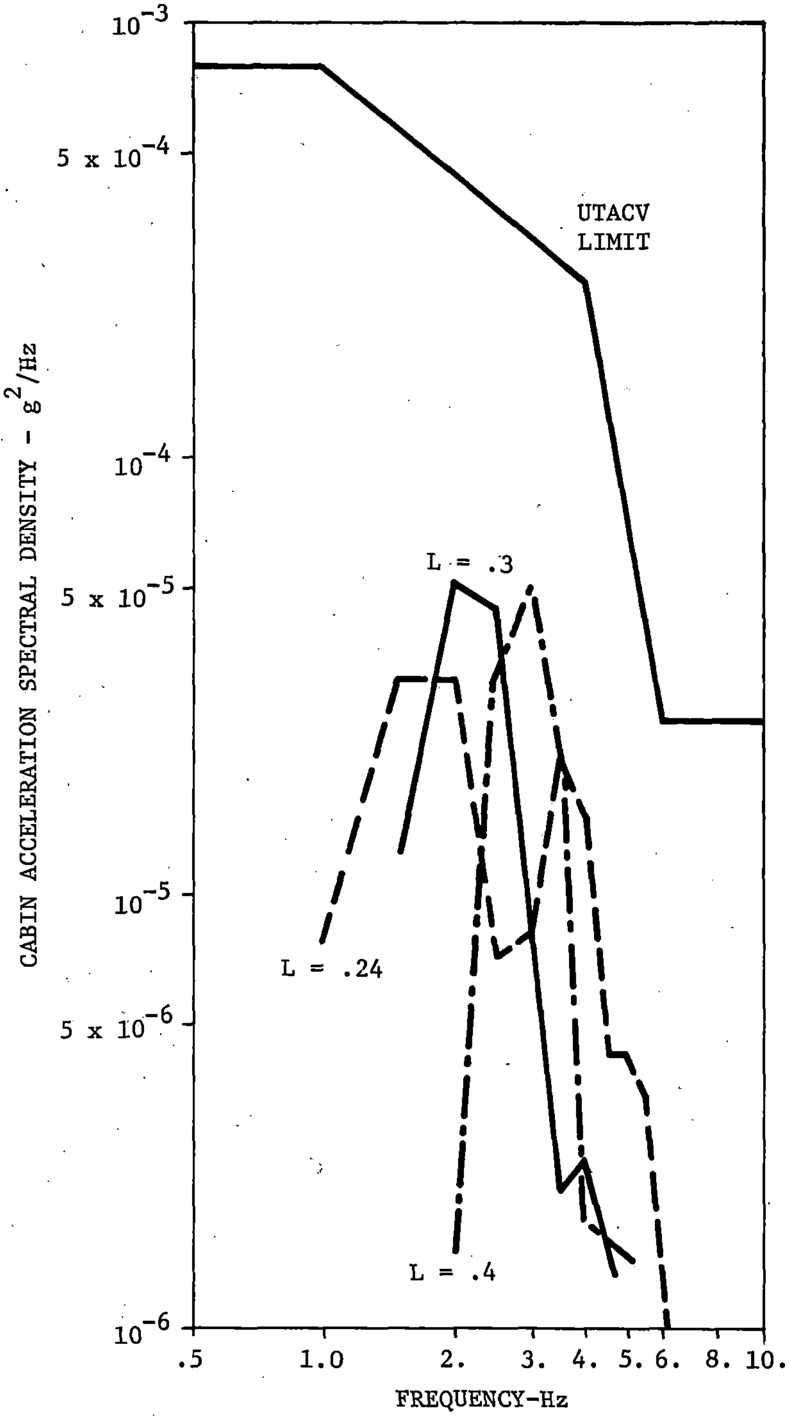

FIGURE 10
ACCELERATION SPECTRAL DENSITY DUE TO FLEXIBILITY ONLY: V = 150 mph

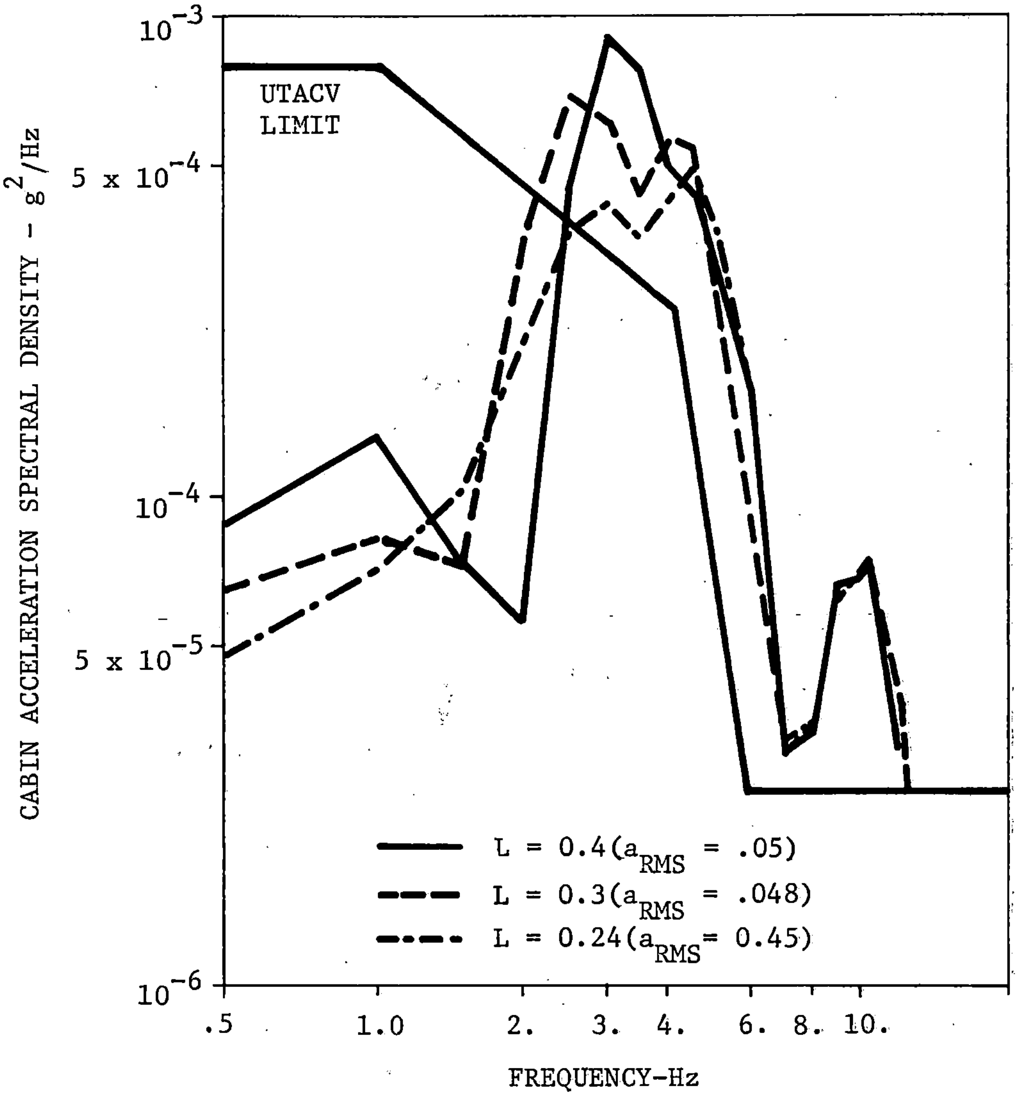

FIGURE 11

ACCELERATION SPECTRAL DENSITY DUE TO FLEXIBILITY AND NOMINAL ROUGHNESS

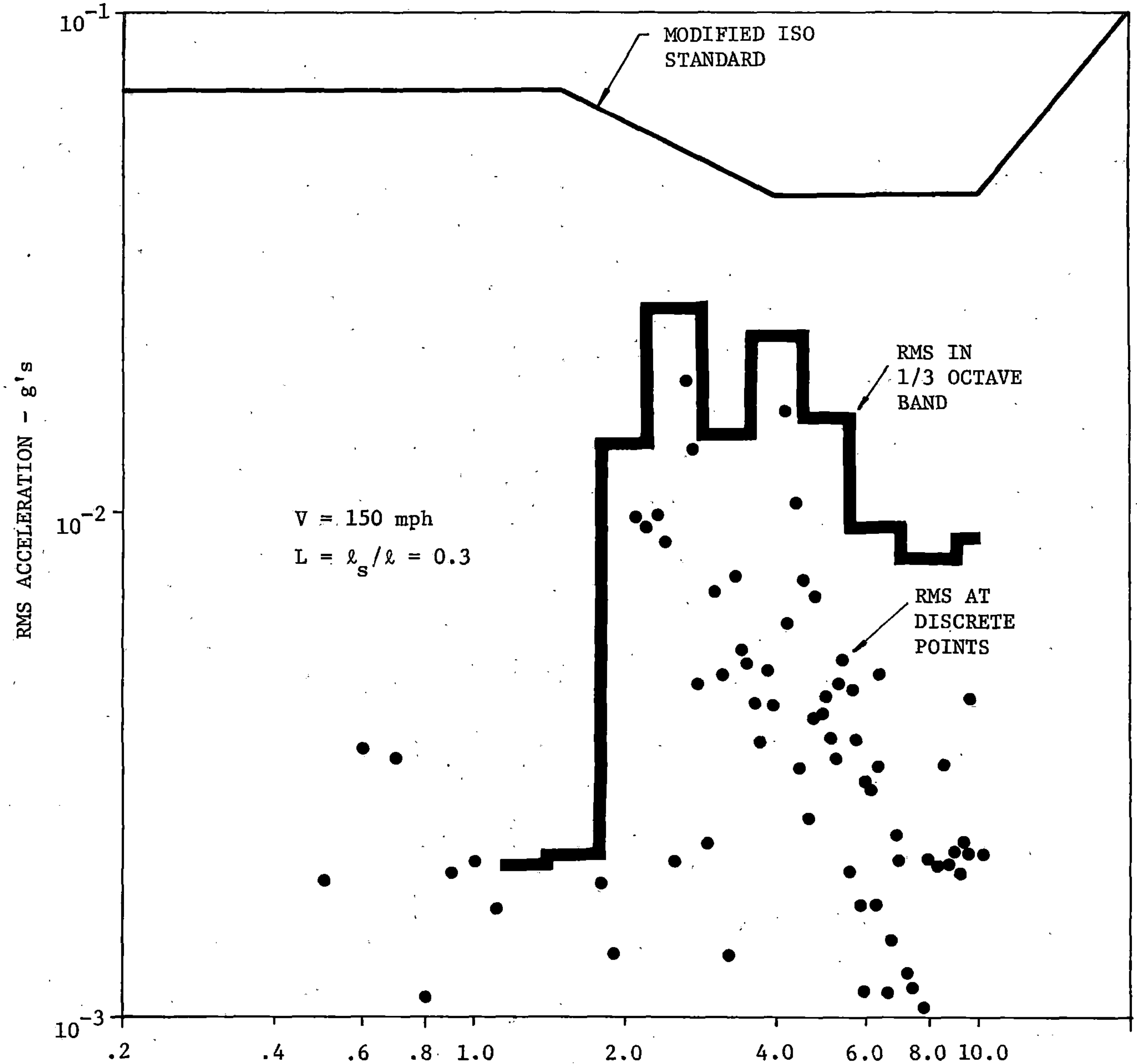

FIGURE 12
CABIN ACCELERATION SPECTRUM DUE TO FLEXIBILITY AND NOMINAL ROUGHNESS

4.0 GUIDEWAY COST ESTIMATION

This report and its companion documents [1, 2] have illustrated various techniques for analyzing the performance of ground transportation vehicles traversing elevated guideway structures. Hopefully, it has also illustrated the limitations as well as the range of options which confront a guideway designer. The most important message to be derived from this report is that guideway design should not be conducted in a vacuum or considered an afterthought to vehicle design. The vehicle/guideway combination is a complex system and innovative guideway design is realizeable only when the detailed design is based on extensive preliminary systems and dynamic analysis work, employing models representative of the vehicle/guideway combination under consideration. The process depicted in Figure 13 illustrates the major elements of a guideway cost estimation procedure. The first step is a systems analysis involving the consideration of one or two (or more) competing vehicle systems. For illustration, two systems labeled A and B are shown. Employing the preliminary analysis (Program QUEST) discussed in References 1 and 2, initial candidate parameters may be selected. Next, using a digital simulation which includes combined guideway flexibility and tolerance irregularities as input [2], the performance responses of vehicles A and B are obtained and analyzed for compliance with ride quality. Typical results of analysis of this type are demonstrated in Reference 2 and Sections 2 and 3 of this report. An iterative procedure is carried out whereby vehicle and guideway designs are refined, to a degree consistent with the state-of-the-art technology, and if acceptable ride quality and other performance indices are met, the guideway requirements can be given to an Architectural and Engineering Firm for detailed design. The parameters should be presented in a format similar to the results shown in Reference 2 and in Section 2 of this paper; that is, sets of equivalent parametric performance curves should be available to the designers, enabling them to choose a least-cost design. During the detailed design phase, testing of the proposed guideway design on the simulation and communication between the systems analysts and the designers, as illustrated by the dashed line in Figure 13, is extremely important. Once the detailed design is complete, the cost and/or price can be determined by submitting the plans to a contractor or estimator. If relevant, comparisons of the vehicle and guideway costs can then be made between system A and system B.

It would seem worthwhile, in light of past experience, that this proposed guideway cost estimating method be employed in some current or future project. A possible scenario for implementing such a study might be to work in conjunction with a current program concerned with conceptual studies of new vehicle/guideway systems. There is no intent for such a guideway cost study to be a parallel effort. Rather, the cost study would augment the conceptual study by using the proposed vehicle's physical and modal related properties and requirements to obtain the optimal guideway parameters required for ride quality compliance.

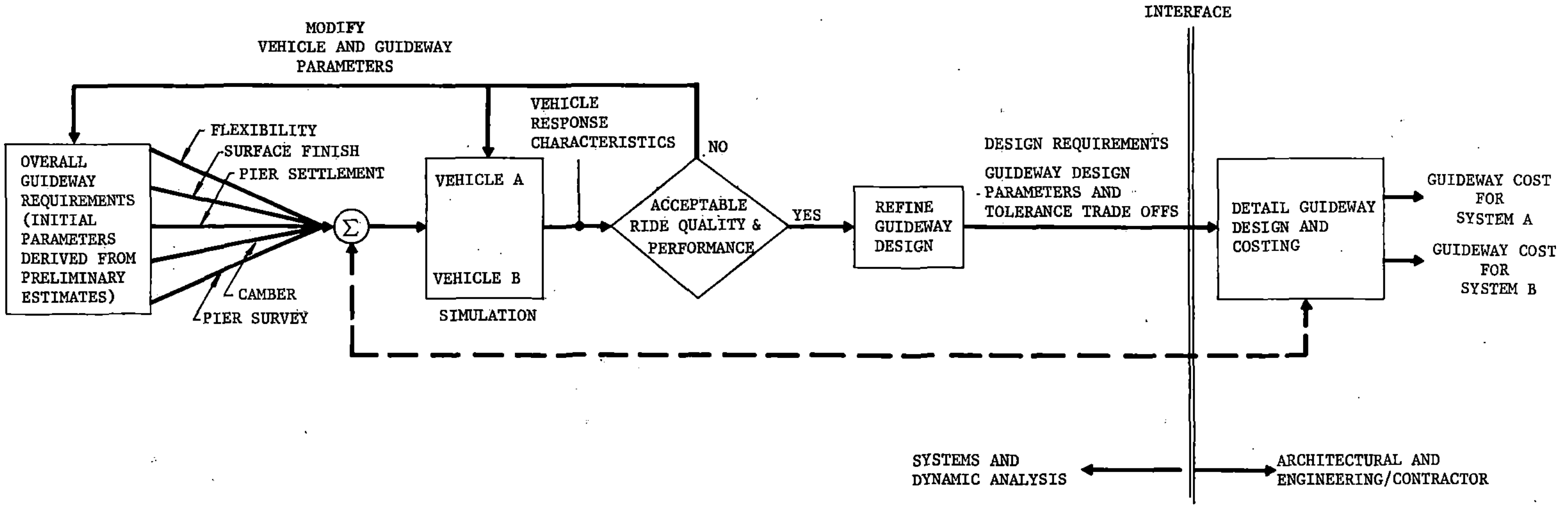

FIGURE 13
A GUIDEWAY COST ESTIMATION PROCEDURE

5.0 CONCLUSIONS

This report has explored some of the potential problems of lightweight guideway design and examined the effects on ride quality. A similar exercise was conducted for the effects of varying pier spacing. A procedure for estimating guideway costs for a particular vehicle/guideway system or for comparing competing systems was outlined.

It has become apparent through this study and companion studies [1, 2] that guideway design for a particular vehicle is an ad hoc process. When all factors such as vehicle length, mass and suspension properties, guideway flexibility properties and accurate roughness models are considered, it becomes obvious that one cannot separate vehicle and guideway design and hope to achieve optimal results.

Generalizations concerning guideway design are associated with analyses which probe only a limited aspect of the problem. Optimal guideway design will be achieved only when extensive dynamic analysis of the vehicle/guideway system is used to generate the guideway flexibility and tolerance parameters, along with the associated trade-off options. Nevertheless, the following conclusions, which are valid <u>for the system considered</u>, can be made.

1. The nominal guideway design weight was reduced by 50%[3] with no adverse effect on ride quality, provided that the nominal value of I (second moment of beam area) was maintained (preferred method). For the twin rectangular beam structure; however, this weight reduction led to a beam which might be unacceptable due to lateral and torsional weaknesses. A weight reduction of 25% for the assumed configuration is probably more realistic. A more efficient design, using I-beams or box-beams, would probably permit weight reductions greater than 25% without attendant lateral or torsional strength defects.

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[3] It should be understood that there was no attempt to refine the nominal design and therefore this large reduction (50%) is neither significant or surprising.

2. When there are restrictions on changing the span fundamental frequencies, weight (area A) reductions are possible only by holding the ratio I/A constant.[4] However, reducing I in proportion to A degrades ride quality since beam deflection is inversely proportional to I. This degradation in ride quality also reduces the tolerance specification trade-offs available to the designer.

3. Assuming that in a span length range of 75 ft.-125 ft., pier settlement and pier survey errors are equivalent, the shorter span (with equivalent cross-sectional area) yields the rougher ride despite the increased stiffness. The pier irregularities at the shorter wavelength dominate vehicle response.

4. As demonstrated in Reference 2, and as shown in this report in Figures 11 and 12, the Urban Tracked Air Cushion Vehicle (UTACV) ride quality standard appears far more stringent than the International Standards Organization (ISO) criterion. The substantial increase in beam stiffness and the tightening of construction tolerances to meet the UTACV criterion, if it is too stringent will require a substantial increase in guideway costs. On the other hand, if the ISO criterion is too lenient, passengers will be subjected to a poor ride.

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[4]This assumes no beam material or length changes.

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