Design Loadings and Structural Considerations for Tracked Air Cushion Research Vehicle

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Simple analyses of vehicle responses to operating conditions are used as the basis for determining the types of loading significant for structural design of the Tracked Air Cushion Research Vehicle (TACRV). A brief development is given for the methods used to determine the magnitude of cushion applied loads along with the internal reactions and load paths associated with the several suspension configurations. Although the choice of materials and conservative design safety factors tend to preclude fatigue problems for the test vehicle, a brief analysis is given to indicate some potential fatigue-critical areas in tracked air cushion vehicle (TACV) prototypes designed to higher stress levels.

Introduction

RESULTS of transportation studies of the Northeast Corridor—a typical, densely populated area—showed the necessity for increased modes of high-speed transportation over medium-distance routes. The tracked air cushion vehicle (TACV) was identified as one of the likely candidates for this use, and the Department of Transportation/Office of High Speed Ground Transportation proceeded with its development. A key point in this development was the decision to extend and confirm the required technology with the construction and test of a fullscale research vehicle. Preliminary design studies resolved the primary characteristics required for the vehicle and its test guideway. The Tracked Air Cushion Research Vehicle (TACRV) has just been completed and will enter its research test phase in the summer of 1972.

Because the TACRV has quite different operating conditions than either wheeled or normal flight vehicles, development of the criteria for its detail design presented many challenging facets. This paper deals with structural design considerations for the particular TACRV, and considers, in a limited way, some of the differences which may arise in the design of vehicles for operating TACV systems.

Vehicle Characteristics and Operating Conditions

Design Configurations

The TACRV, shown in Fig. 1, consists of two major units: body and chassis. The body is supported by the chassis through the suspension system at each end. Also, the four levitation and four guidance cushions are mounted to the chassis through independent suspensions. In actual use, the combined suspensions have two configurations.

Body/Chassis

In this configuration, the body-to-chassis suspension is in use while the levitation cushion suspension is locked out, with rigid links in place of its springs. The body is free to roll about its suspension pivot points; the roll can be biased to provide banking for coordinated turns.

Independent Cushion

The body is linked rigidly to the chassis at four corner points, eliminating vertical and roll freedoms between body and chassis. The levitation cushion suspension provides vertical freedom and a limited roll freedom by differential deflection.

In both suspension configurations, the body is connected to the chassis with lateral restraint links at the roll axis. The lateral dynamic compliance is through the guidance cushion-to-chassis suspension.

Mass Distribution

Initially, the TACRV will be in a lightweight, aeropropulsion configuration for tests to 120 mph. Higher speeds (to 300 mph) will be attained after installation of the LIM propulsion system (LIMPS), with an attendant increase in weight. Table 1 gives the mass properties for the TACRV in the several suspension and propulsion configurations.

Operating Conditions

Because the purpose of the TACRV is to provide a proven technology basis for future TACV development, the vehicle and guideway have been designed to enable exploration of a wide range of operating conditions. The

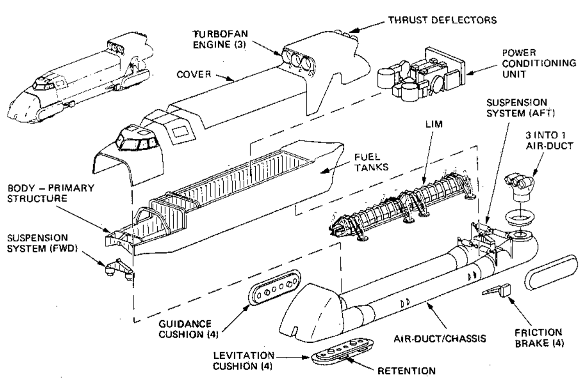

Fig. 1 TACRV exploded view.

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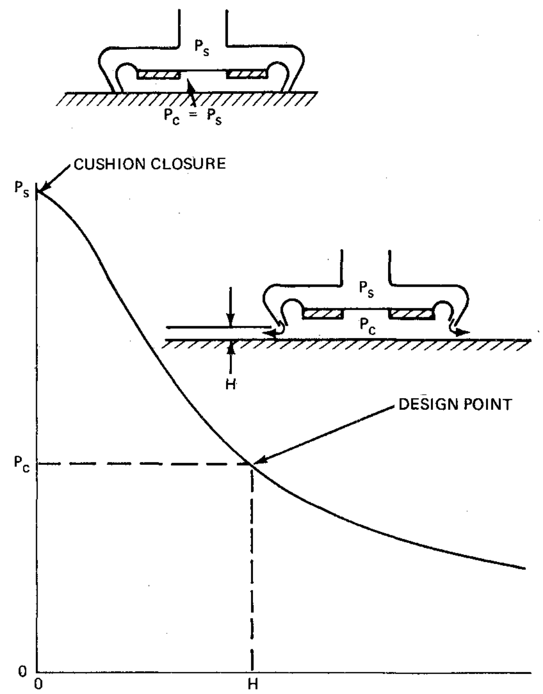

Fig. 2 Cushion lift vs gap.

first segments of guideway will be constructed at-grade, to close tolerances on roughness and waviness. However, the completed guideway (22-mile loop) will include special test segments with substantial waviness and flexible spans. These guideway parameters, in combination with the variations in cushion operating conditions and suspension configurations, provide the conditions for which the TACRV structure must be designed.

Types of Loading Considered

The types of loadings on the TACRV may be readily derived from a review of its operating conditions. The primary vehicle support and guidance loads are applied at the levitation and guidance cushions. The air pressure under the cushions is determined by the system air supply pressure and the cushion gap to the guideway surfaces. When the weight is not supported by the air pressure, the cushion rests on skids. In this situation, the cushions may be subjected to tangential loads as well as normal bearing forces. Although the overall aerodynamic lift and side forces on the vehicle are significant, they are reacted by cushion loads. (Local pressure and loads are significant on the engine inlet and speed brake).

Other major loading inputs come from friction brakes, drag chute, retention, and arrestment. Four friction brake pads, which are pressed against the guideway guidance panels, are a source of both normal and tangential forces. A drag chute, for emergency use in high-speed deceleration, is located in the rear of the engine area. Upstops are provided on the outer edge of each levitation cushion. In case of unusual disturbances, these stops contact the lower surface of the guideway guidance panels to limit excessive motion and provide positive retention of the vehicle in the guideway at all times. Capability for arresting the vehicle at the open end of the track by means other than friction or aerodynamic braking is a safety requirement, until the guideway loop is completed. The arresting is accomplished by the vehicle chassis nose engaging a barrier tape stretched across the guideway.

Hoisting and jacking are the major handling loadings to be considered for vehicle design criteria. (Because the TACRV is one-of-a-kind, it will be given special shipping protection so that such loadings will not be design constraints to the vehicle).

Loads within the suspension system due to the interaction of the various components in reacting external loads are considered internal loads. The load paths and magnitude of these loads depend upon the vehicle mass configuration as well as the suspension mode. The air supply system ducting and various service systems, particularly braking, also contribute substantial internal loads to complete the definition of loading conditions to be considered in establishing the structural design.

Design Loading Conditions

A satisfactory set of design loading conditions for the unique TACRV was developed by first determining the external load levels from the design capability, and then the internal loadings were calculated as dependent functions.

Cushion and Skids

The cushion closure condition is a good example of this rational approach. The basic variation of cushion load (pressure, P_C) with operating gap, H, is shown in Fig. 2. The steady-state design point corresponds to the pressure to sustain the weight of the vehicle. As the gap changes due to guideway unevenness or dynamic motion of the vehicle, the pressure increases and decreases. (The curve is for steady-state cushion characteristics and, strictly

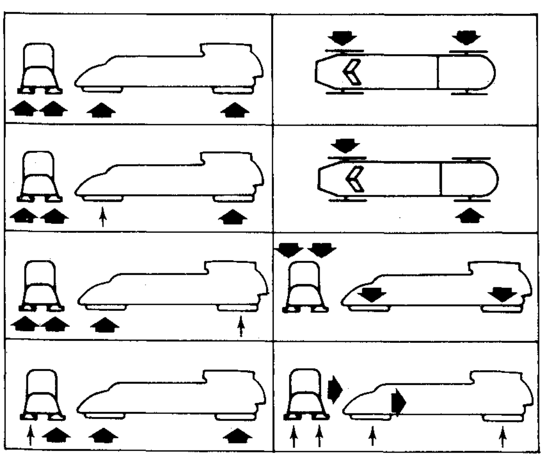

Fig. 3 External loadings.

Table 1 TACRV mass properties

	Aeropropulsion configuration	½ LIMPS[a]	LIMPS[a]
Body weight, lb	21,000	35,000	40,000
pitch inertia, lb-ft^2	2,700,000	2,900,000	3,100,000
roll inertia, lb-ft^2	290,000	340,000	380,000
Chassis weight, lb	13,000	18,000	22,000
pitch inertia, lb-ft^2	3,500,000	3,500,000	3,500,000
roll inertia, lb-ft^2	160,000	160,000	160,000
Cushion weight, lb	3000	3000	3000
Sprung-to-unsprung			
Mass ratio (vertical)			
body/chassis	1.62	2.69	3.07
independent cushion	10.3	15.0	16.7

[a] LIM weight listed under chassis is separately supported vertically and laterally and enters into longitudinal mass inertia only.

speaking, applies only to the extreme points of a particular cycle of motion.) As the gap approaches zero, P_c approaches the system supply pressure, P_s. Some further increase in pressure is possible by compression of the trapped air after cushion closure (zero gap). However, because vehicle tests will be closely monitored, the test conditions will examine the working limit of cushion gap, but will be limited to avoid excessive excursions beyond the cushion closure point. Thus, the design loading to be applied on the air cushions can be rationally set equal to the maximum system supply pressure—700 psf. For a levitation cushion, this corresponds to a load of 28,000 lb; for the guidance cushion, 15,700 lb.

The skids under the levitation cushions bear the weight of the vehicle at rest or during emergency skid stop. In this situation, it is quite possible that diagonally opposite skids would carry one-half the weight (26,000 lb/skid). Also, although the skids are recessed under the cushion, they may contact the guideway after cushion closure. Therefore, the skid design load was raised to the cushion closure load, 28,000 and 15,700 lb for levitation and guidance, respectively. The primary difference between skid loading and cushion closure is the presence of tangential loading. A friction coefficient of 0.4 is used, based on sample material tests, to determine the tangential load in terms of the skid design load.

Brakes

The friction brakes are actuated hydraulically. The normal load against the guideway is determined by the mechanical design and hydraulic system pressure. For the nominal system pressure (3000 psi), the brake pad normal force is 6800 lb. At low speeds, the friction coefficient may approach 1.0. This value is used to define the tangential force.

The aerodynamic speed brake is designed for an average normal pressure equal to the dynamic head at 300 mph with a 60-mph headwind (288 psf at the test site elevation of 5000 ft). The load (8600 lb) is applied in a triangular distribution, 0 at the base and 2X average at the tip.

The parachute loading, opening at maximum speed, is 11,000 lb.

Arrestment

The arresting device to be used at the open end of the guideway provides a nearly constant deceleration force of 100,000 lb applied to the chassis nose.

Retention

There appears to be little possibility of developing sufficient forces on the vehicle to cause it to lift or roll out of the guideway. However, to cover unforeseen combinations of dynamic motions and external forces, a retention-force capability of 8000 lb at each levitation cushion has been assumed for a design negative 1g condition for the lightweight vehicle design.

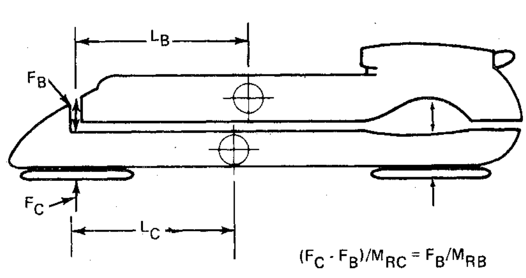

Fig. 4 Asymmetrical loading with bottomed suspension.

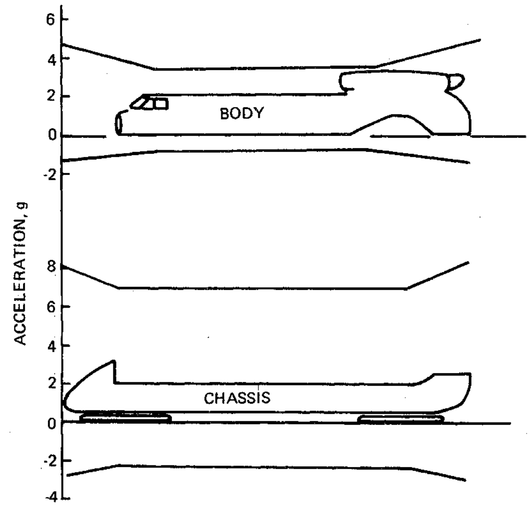

Fig. 5 Vertical acceleration envelope.

Suspension Reactions and Inertial Factors

A number of combinations of external loadings are shown in Fig. 3. The heavy arrow represents application of cushion closure, retention, or arrestment loadings, while the light arrow indicates the normal 1g balance loading. Where the cushion load may be substantially below its normal balance loading (as in retention or lateral oscillations), the cushion load is taken as zero and no load arrow is shown.

A perfect secondary suspension provides complete isolation of the body from all disturbances represented by these external loadings. The unbalance between the applied loads and the body steady-state suspension reactions is absorbed in the inertia of the chassis or cushions, depending upon the suspension configuration. For example, the aeropropulsion, body/chassis configuration with four-cushion closure loading yields a chassis acceleration of

$$a_C = (4 \times \text{cushion closure load} - \text{body weight})/\text{chassis weight}$$

$$= (4 \times 28{,}000 - 21{,}000)/13{,}000 = 7g$$

Similar balances for moments provide pitch and roll accelerations for the unsymmetrical loadings.

Practical limits to the stroke available for the suspension system lead to situations in which the spring or actuator elements bottom and the loads are transmitted directly. The load paths and inertial reaction in body and chassis depend upon the suspension configuration.

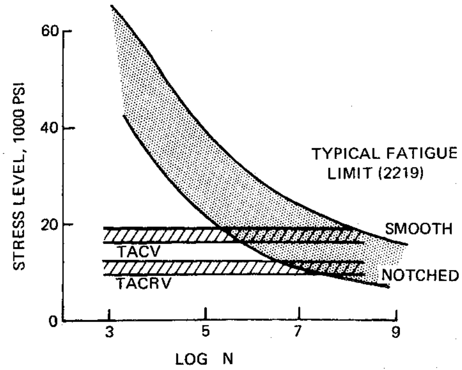

Fig. 6 Fatigue life expectancy.

Body/Chassis

With forward levitation cushion closure, suspension bottoming would occur at the forward end of the body. The inertial balance equations are resolved by equating the acceleration of chassis and body at the suspension location, as shown in Fig. 4. For example, in the aeropropulsion body/chassis configuration with closure of two forward cushions, the effective reduced mass at the suspension is

$$M_{RB} = M_B/3, \text{ for the body}$$

$$M_{RC} = M_C/2, \text{ for the chassis}$$

and the dynamic force, $F_C = 2 \times 28{,}000 - 34{,}000/2 = 39{,}000$ lb

The suspension load at the forward body support is

$$\text{Body weight}/2 + F_B = 21{,}000/2 + 39{,}000/[1 + (13{,}000/2)/(21{,}000/3)] = 30{,}800 \text{ lb}$$

Similar equations can be written for the aft cushion closure case. With all four cushions at closure, the body suspension may be bottomed at both ends or at either end, and the balance of inertial reactions will be different in each case.

Due to the geometry of the suspension and body chassis clearance, there are two modes of bottoming in the roll cases. One, with a low vertical position of the suspension, occurs by contact of bumpers between body and chassis, which apply loads at the same point as the body-to-chassis links used in the independent cushion suspension mode. The other mode is actual bottoming of the roll actuators. In this latter case, it is the condition for equal roll acceleration of body and chassis that is used to resolve the inertial balance equations.

It is inherent in the premise of equal acceleration for the inertial equation balances between body and chassis that the assumed bottoming of the suspension system during testing will be monitored to limit its extent. Because the purpose of testing is to determine the operating limits of the vehicle/guideway system, it is expected that bottoming will occur. However, the tests need not be extended to conditions of excessive bottoming (high relative velocity) with substantial momentum to be absorbed at impact of body and chassis.

Independent Cushions

Because the cushions have relatively low mass and the body and chassis are rigidly linked, the body/chassis balance accelerations may be obtained directly. The link loads are determined in proportion to the body and chassis mass properties.

A partial summary of inertial reactions for the body/chassis configuration is shown in Fig. 5.

Structural Design Considerations

Several primary considerations enter into the final structural design for the TACRV: stiffness, moderate life expectancy, and low cost.

Stiffness in the research vehicle is necessary to give an adequate static reference framework for the systems under test: the cushions and suspensions. In addition, the interpretation of test data is more direct, with a minimum of interaction between structural modes and rigid-body dynamics modes. Because the major dynamic inputs can range up to 10 cps, the stiffness of the structure should provide higher frequencies for the structural response.

As a research vehicle, calendar life may be 5 to 10 years, but total actual running time is expected to be less than 5000 hours. However, even with limited time spent in conditions with substantial dynamic loads, the number of cycles for the applied load levels will be high. For example, 5 minutes/day at 5 cps for 10 years is in the 10^6–10^7 cycle life range.

To facilitate design and fabrication, extensive use has been made of welded aluminum. Because structural weight was not a critical factor, it was determined to keep adequate margins in design in lieu of static structural tests on the one-of-a-kind vehicle.

Safety Factors

Applied (limit) loads for the structural design of the vehicle are based upon severe test or emergency conditions. In lieu of static tests to confirm the analysis of the overall structure, an ultimate factor of 2.0 is applied to the limit loads.

An additional factor of 1.33 is applied in welded areas subject to tension. For local fittings which have limited energy absorption capability, the ultimate design loads are increased by applying an additional factor of 1.15.

Because the vehicle has been designed for severe loads with ample safety factors, resulting structure has an infinite fatigue failure life. Cyclic loading, which feeds through the suspension system, will result in body loads which are about 1g ±0.1g, and result in maximum stress levels on the order of 8000 psi. This is substantially less than the infinite-life stress level of 14,000 psi for the material used in the body structural members. The chassis is subjected to higher levels of cyclic loads, particularly when the cushions are hard-mounted to the chassis. At 300 mph, over a rough guideway, these loads are on the order of 1g ±3g. Because the chassis structure is designed for an ultimate stress of 60,000 psi, which would occur at 20g, the peak cyclic stresses will be on the order of 12,000 psi—also less than the endurance limit.

Typical fatigue strength for the TACRV structure is shown in Fig. 6.

Application to TACV Design

While the previous review of structural design considerations pertain to the particular TACRV design, the general approach should also apply to production design of the TACV. However, several significant differences may be expected.

Availability of Design Data

From experience with the TACRV design and test and other related developments, actual guideway characteristics, cushion loading conditions, and suspension capabilities will be more accurately defined.

Weight Control

In production design, emphasis will be placed on achieving lower structural weight through use of higher-strength alloys and static test proof of the structure.

Fatigue

The net result of closer definition of loads and more efficient structure will be generally higher stress levels. This, in combination with a higher level of utilization, will require closer examination of the life expectancy of the structure. It is expected that the smooth-ride goals for TACV body will be achieved, so that this portion will have long life. However, cushion and chassis structure will most certainly be potential fatigue-critical areas, as indicated in Fig. 6.