REPORT NO. FRA/ORD - 78/38

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IMPROVED PASSENGER EQUIPMENT EVALUATION PROGRAM

Computer-program documented passenger systems

TECHNOLOGY REVIEW

Unified Industries Incorporated
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MAY 1978

SEMIANNUAL REPORT

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Prepared for

U.S. DEPARTMENT OF TRANSPORTATION
FEDERAL RAILROAD ADMINISTRATION
Office of Research and Development
Washington, D.C. 20590

13-Rail Vehicles &
Components

4412

NOTICE

TECHNICAL REPORT STANDARD TITLE PAGE

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
FRA/ORD - 78/38		

4. Title and Subtitle	5. Report Date
Improved Passenger Equipment Evaluation Program Technology Review	May 1978
	6. Performing Organization Code
7. Author(s) Andre L. DeVilliers, Unified Industries Incorporated; Arthur L. Dow, Unified Industries Incorporated; R.B. Watson, L.T. Klauder & Associates; Dr. Richard Uher, Carnegie-Mellon University	8. Performing Organization Report No.
9. Performing Organization Name and Address: Unified Industries Incorporated, 205 South Whiting Street, Suite 201, Alexandria, Virginia 22304	10. Work Unit No.
	11. Contract or Grant No. DOT–FR–74249
12. Sponsoring Agency Name and Address: U.S. Department of Transportation, Federal Railroad Administration, Office of Research and Development, Washington, D.C. 20590	13. Type of Report and Period Covered: Semiannual, October 1977 to March 1978
	14. Sponsoring Agency Code: FRA/RRD-22

15. Supplementary Notes

Contractor: Small Business Administration
Washington District Office
1030 Fifteenth Street, N.W., Room 250
Washington, D.C. 20590

16. Abstract

The status of two foreign rail technologies is analyzed in this second of four semiannual reports. The two technologies are Trucks and Carbody Construction.

Trucks are discussed from the viewpoint of their function in the rail vehicle suspension system. This function consists of vehicle guidance, vibration isolation, vehicle support, and traction and braking. Several current trucks are reviewed, followed by a discussion of radial-axle trucks.

Carbody construction is reviewed from 1965 forward. French technology, 1965 to 1975, is covered, followed by discussion of two modern trains, the TGV (France) and the ET403 (Germany).

Advanced technologies are then discussed, principally construction with aluminum extrusions followed by a short discussion of composite materials.

17. Key Words	18. Distribution Statement
Rail Trucks; Rail Vehicle Suspension; Radial Trucks; Rail Vehicle Dynamics; Rail Vehicle Guidance; Carbody Construction; Finite Element Analysis; Aluminum Extrusion for Carbodies	This document is available to the U.S. Public through the National Technical Information Service, Springfield, Virginia 22101

19. Security Classif. (of this report)	20. Security Classif. (of this Page)	21. No. of Pages	22. Price
Unclassified	Unclassified		

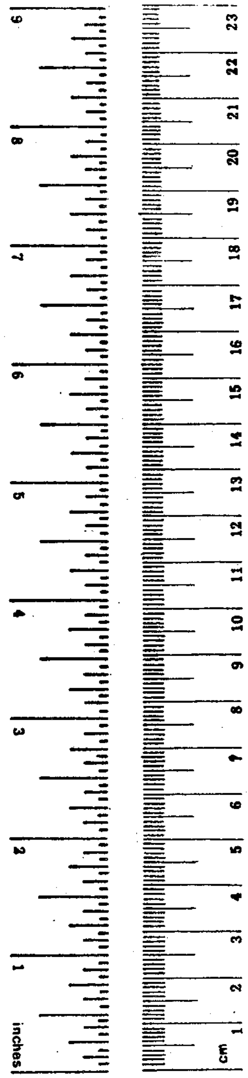
Form DOT F 1700.7 (8-69)

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

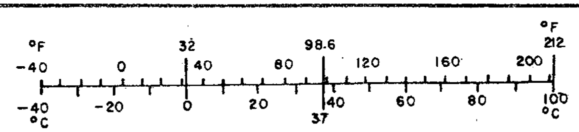


TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

1. TRUCKS

PREFACE

The technology of railway trucks covers a broad spectrum of activity ranging from academic-sector research through the Track/Train Dynamics program of the Association of American Railroads (AAR) to the design details, material applications, and other aspects of new as well as existing truck designs. As a result, it is valuable to an assessment of truck technology to discuss not only the current hardware available, but also the state of knowledge about truck behavior. In this way, the reasoning behind truck designs can be clarified by deeper knowledge of the problems being addressed.

Law and Cooperrider,[1] in their literature survey, have provided a cogent discussion of the purposes behind truck technology and what is known about the dynamic behavior of trucks. The introduction and first two sections of the truck portion of this report (through Guidance Technology) are therefore derived from their paper, although some explanation and modification have been done to suit the present context.

INTRODUCTION

With the widespread acceptance of rails for intercity traffic in Europe, the development of improved train systems appears to be financially rewarding. Many countries have produced improved passenger trains that combine the latest technology in many fields and represent high national pride. Included in this latest technology are improved passenger car trucks and suspension systems.

While the following discussion of passenger car trucks is mainly about European technology, development of improved truck technology in America is also reported.

For purposes of discussion, a truck is a cart-like device which supports one end of a railway vehicle. It is a frame which holds two or more wheelsets or axles. The wheels are rigidly fixed to the axle and therefore rotate together. In addition, the wheels are usually tapered or profiled to provide a self-centering action as the axle traverses the track.

The performance objectives of rail vehicle suspension system design are:

a. To provide guidance with adequate stability margins.

b. To provide effective vibration isolation so that passengers experience a comfortable ride and freight is not damaged.

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[1] E. H. Law and N. K. Cooperrider, "Literature Survey of Railway Dynamics Research," submitted for presentation at Applied Mechanics Division Transportation Symposium, ASME Winter Annual Meeting, 1973. The Law and Cooperrider survey includes several references to other researchers, several of whom are referred to later in this discussion.

c. To provide sufficient traction and braking capability for all operating conditions.

d. To support the vehicles.

It is desirable to minimize the dynamic fluctuations of guidance and support forces to minimize wear and damage to vehicle components and track. Often the objectives for suspension are conflicting; but modern truck design practice is beginning to cope with these conflicts.

Among recent research efforts are those which have centered on determining the details of the process by which guidance is achieved and on the stability problems which may arise in designing for good guidance. These stability problems are generally grouped under the broad term "hunting." As will be discussed later, primary hunting involves lateral, yaw, and rolling motions of the carbody with little truck motion. Secondary hunting is a coupled oscillation in yaw and lateral displacement of the wheelsets and truck frame (with little carbody motion) which is damped below a certain critical value of forward speed. Above the critical speed, disturbances of the truck from equilibrium will result in growing lateral oscillations with wheels flanging, ultimately leading to wheel climb and derailment. As desired operating speeds for passenger vehicles have increased, the problem of determining how passengers judge relative comfort in a vibration environment has received attention.

FUNCTIONS OF RAIL VEHICLE SUSPENSION

For the purposes of this report, the term "rail vehicle suspension" will be used to denote the complete assembly comprising the wheels, axles, truck frame, and any suspension elements connecting the carbody to the truck frame.

The main functions which a railway vehicle suspension must provide in carrying the desired payload are the following:

a. It must provide guidance with attendant dynamic stability and good response on both nominally straight and curved track.

b. The suspension must provide effective vibration isolation in response to track irregularities over the entire speed range.

c. The suspension must adequately support the vehicle under a variety of operational situations.

d. The suspension must provide adequate adhesion over the performance range of traction and braking requirements.

The requirements to fulfill these various functions have always conflicted to various degrees (as will be discussed later). For example, passengers expect the same level of comfort when they ride the Metroliner or Turboliner as they have in their own automobiles on the expressway or in a jet transport flying at 30,000 feet. Since the passengers' comfort is very much affected by the vibration environment, successful design of the vehicle suspension as a vibration isolation device is required. Suspension design can also be further complicated for locomotives by the requirement for high tractive forces to attain high speeds. These forces combined with the forces required for guidance may exceed the adhesion level available between wheel and rail.

GUIDANCE - TANGENT TRACK

The guidance function of the conventional rail vehicle truck leads to the requirement that the truck respond to the required heading changes of the track with minimal steady-state error and sufficiently small overshoot that flange contact does not occur. The reason for the introduction of the coned or tapered wheel was to provide a self-centering action for the vehicle. If the wheelset is initially disturbed from the central position on the track, it will pursue a sinusoidal path about the centerline of the track (assuming no suspension restraints). This type of motion, shown in figure 1, is called "kinematic mode" at low speed. The frequency of this motion for a single wheelset is $\Omega = V\sqrt{\lambda/(LR)}$ where V is the forward speed, λ is the conicity or taper ratio, R is the nominal rolling radius, and L is one-half the rail gage. When there are no primary suspension elements connecting the wheelset to a truck frame, this oscillation is neutrally damped at very low speeds. When the wheelset is mounted in a truck frame such that there are primary suspension elements between the wheelset and truck frame and secondary suspension elements between truck and carbody, the coupled oscillation in yaw and lateral displacement will be damped at low speeds. However, as speed is increased, a speed will be reached (called the linear critical speed) above which disturbances of the wheelset or truck from equilibrium will increase with time. It can be shown that the linear critical speed is proportional to $1/\sqrt{\lambda}$. Some investigators use the term "secondary hunting mode" to refer to the coupled lateral and yawing oscillation. They use the term "kinematic mode" to refer only to the motion of a free wheelset at very low speeds, where inertia effects are negligible. This convention will be used in this report.

For good guidance on straight track, the frequency of the secondary hunting mode should be high to facilitate fast response of the self-centering action. The damping of the mode is a function of the suspension parameters as well as the creep force and moment coefficients which characterize the surface forces and moments exerted on the wheel by the rail. Since the wheel taper ratio, λ, should be high for fast response, there is a fundamental conflict between the requirements of guidance and stability. If λ is too large, the linear critical speed may be within the desired operating speed range of the vehicle.

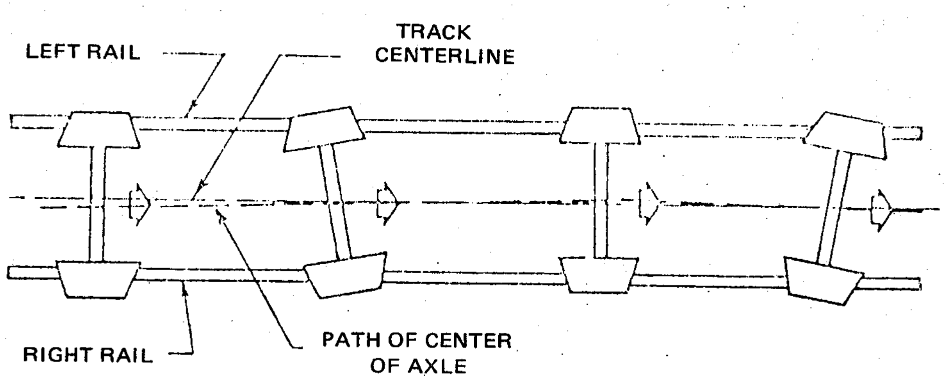

FIGURE 1. KINEMATIC MODE.

GUIDANCE - CURVED TRACK

The objective of good passenger car truck design is to have sufficient conicity to traverse curves without wheel flange contact. Curving without flange contact is facilitated by flexible suspension, large conicity or wheel taper ratio, and short wheelbase. Unfortunately, these requirements conflict with the requirements for stability of the secondary hunting mode. The creep forces required for guidance in curves also cause serious reductions in the available adhesion for traction and braking.

In designing for good guidance performance on tangent track and curves, it is desirable that the characteristics of components do not change materially with length of service or changes in environmental conditions. Both the linear critical speed and the curving performance depend strongly on the wheel conicity or taper ratio and the suspension flexibility. As wheels wear, the lateral profile gradually changes from a straight taper to a curved profile. The "effective conicity" usually increases during this process, and the dynamic performance will thus change with wheel wear. The flexibility of the suspension may also change, thus leading to changed and possibly degraded dynamic response.

VIBRATION ISOLATION

A second function of the rail vehicle suspension is the isolation of the vehicle body from disturbances caused by the train operation over roadbed irregularities. For passenger vehicles this function is met by keeping the motions of the carbody within the range of human comfort. Isolation from roadbed disturbances is also desirable in order to minimize wear and damage to vehicle components such as side bearings, centerplates, bolsters, and the underframe.

Human sensitivity to motion such as passengers experience in a rail vehicle has received considerable attention and investigation. The research in this area has revealed that ride comfort is best measured in terms of acceleration and jerk levels (time rate of change of acceleration) in each of the three linear directions of motion. Acceptable levels for the steady accelerations and jerks experienced in starting, stopping, and traveling through curves are well established. Unfortunately, an accepted method for measuring and evaluating ride comfort under random vibratory motion, such as that caused by rail roadbed irregularities, has not emerged from the multitude of efforts in this area.

Human sensitivity to vibratory motions in the 0.1 to 20 Hz frequency range is the primary concern in evaluating vehicle ride. Comfort criteria for vibratory accelerations are usually given in terms of limits of "isocomfort" curves for vertical and lateral acceleration as a function of frequency. Curves illustrating such criteria are shown in figure 2.

These curves are based on human sensitivity data obtained by oscillating human subjects at discrete frequencies. Unfortunately, the actual motions felt by vehicle passengers do not occur at single, discrete frequencies but rather are random vibrations with frequency content spread over all frequencies. Consequently, it is difficult, if not impossible, to make meaningful comparisons between the comfort criteria expressed in terms of sinusoidal vibrations, and the actual random vibration of the rail vehicle.

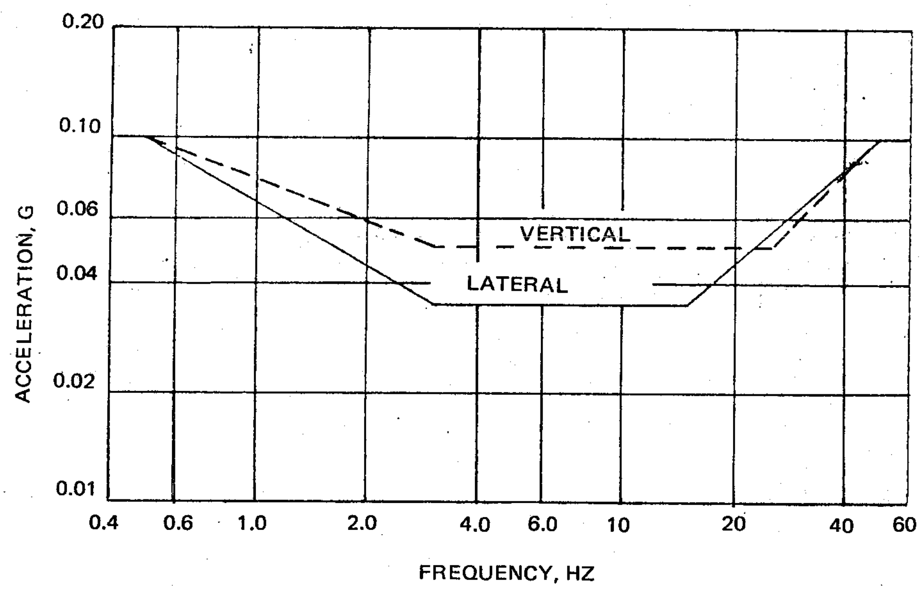

FIGURE 2. TYPICAL ENVIRONMENTAL VIBRATION COMFORT LIMITS (from Law and Cooperrider).

The random vehicle motions can be best expressed in terms of power spectral density (PSD), which describes the intensity of the random vibration at a certain frequency. The area under the PSD curve in any frequency band is the mean square value of the response in that band. Thus, some proposed comfort indexes can be obtained directly from a PSD measurement.

In summary, an analytical study of the riding quality of a railway vehicle entails the computation or simulation of the random motion of the vehicle body in response to the random irregularities of the rail roadbed. The roadbed irregularity inputs to such an analysis can either be in the form of time series data of actual roadbed irregularities or in the power spectral density format, depending on the requirements of the subsequent analysis. The PSD curves of the track irregularities are calculated from actual track measurements.

VEHICLE SUPPORT

A third necessary function of the rail vehicle suspension is to support the vehicle body and its contents. The chief concerns in this area involve the dynamic loads exerted between components or at the wheel/rail interface, as well as the transmission of static loads to the roadbed. The suspension should insure equal and uniform distribution of the loads through the components of the truck to the track and should also minimize dynamic loads. This vehicle support function depends primarily upon the characteristics of the vertical suspension of the truck, in contrast with the guidance function which depends most heavily on the lateral suspension.

One measure of the suspension support performance is the ratio of lateral to vertical loads (L/V ratio) at each wheel. It is desirable to keep this ratio as low as possible. This ratio is also referred to as the derailment quotient, because it has been found to be valuable in predicting the tendency of a vehicle to derail under certain conditions.

TRACTION AND BRAKING

The vehicle suspension system also exerts an important influence on the traction and braking performance of the vehicle. From a traction and braking viewpoint, the objective is to maximize the force available at the rail. Thus, the mechanism for interaction between the suspension system and the traction and braking performance is that of the tangential forces exerted by the rails on the wheels. These forces depend on the geometry, surface conditions, and normal load at the wheel/rail interface as well as on the dynamic behavior of the rail, wheel, and vehicle. The suspension system, through its dominant influence on the dynamic behavior of the vehicle, helps determine the dynamic variation of the normal load at the wheel/rail interface, as well as the level of dynamic lateral forces needed to guide the vehicle.

The traction and braking problems encountered in rail vehicle operation include reductions in adhesion at high speed, in running through curves, and on rough roadbed.

The suspension functions in the vehicle support, traction, and braking performance areas overlap because of the direct influence on the available adhesion of the normal load between wheels and rails. Thus, the static and dynamic wheel load distributions referred to in the preceding section are also relevant to braking and traction performance.

The guidance forces required in curves, more than other factors, cause serious reductions in available adhesion. The contribution of guidance forces to the reduction of available adhesion is, however, an area of ongoing research.

GUIDANCE TECHNOLOGY

As previously discussed, the requirements for guidance of the railway vehicle frequently conflict with those for stability of response, traction and braking, and vibration isolation. A railway vehicle should not respond to high-frequency roadbed irregularities, but should respond quickly and with minimal steady-state error to low-frequency heading changes of the track. The response should always be stable. If the wheel conicity is large, the suspension soft, and the wheelbase short, the vehicle may have excellent curving performance, but it may not isolate the passengers or payload from rail irregularities and it may be unstable.

Research has been concentrated on attaining understanding of both secondary hunting and steady-state curving. After determining how these aspects of the dynamic response are affected by various design parameters, trade-off decisions necessary for good guidance can be made.

STEADY-STATE ERRORS

Tangent Track

A conventional railway vehicle with profiled wheels mounted rigidly on a solid axle should, in the absence of roadbed irregularities, center itself between the rails when traveling on tangent track. (Free-wheeling trucks with cylindrical treads do not have a centering ability and tend to drag one flange continuously.) There are certain conditions, however, when this will not occur. Above a critical speed, which is characteristic of the vehicle, hunting behavior will occur in which the vehicle will sustain an oscillation in which it travels from side to side. This behavior is discussed in some detail in the next section. Below the critical hunting speed, steady-state centering errors may occur because of asymmetries in suspension or wheel diameter or because of rotational friction between truck and carbody.

The steady-state errors that are possible because of friction and asymmetric wheels for conventional passenger trucks include those due to centerplate or side-bearing friction. When acting in series with a spring to provide yaw stiffness, the truck can be caused, when leaving curves, to continue along tangent track in a crabbed orientation. Under certain conditions (small wheel conicity, high friction, and stiff primary suspension), the error may be large enough to cause one or more wheel flanges to drag on the rail. This behavior also occurs when wheel radii differ.

Curved Track

As discussed previously, it is desirable to negotiate curves without flange contact. As a free wheelset rolls around a curve, it is displaced radially outwards such that the outer wheel rolls on a larger radius than does the inner wheel. Therefore, the linear speed of the outer wheel is greater than that of the inner wheel. To generate the lateral creep forces required for steady rolling in a curved path, the wheelset adopts a yawed position. Longitudinal creep forces act on the wheelset and generate the moment necessary to maintain the yawed position. Increasing the longitudinal stiffness (which increases yaw stiffness) leads to greater lateral displacements before sufficient longitudinal creep forces are produced to yaw the wheelset by the required amount to achieve dynamic equilibrium. If the creep forces required for equilibrium are greater than the maximum friction available, slipping will occur and the flanges must come into play. Thus, flanging may occur either because of the lateral displacements exceeding the available (flange) clearance or because of slipping occurring and the wheelsets sliding into flange contact. The lateral displacement from the track centerline (or tracking error) of the wheelset or truck is thus a function of the suspension stiffness. There exists a minimum radius curve which may be negotiated by a given wheelset or truck without gross slipping of the wheels on the rails and subsequent flanging. Therefore, proper suspension design is essential to good curving performance. Since good curving is promoted by flexible suspensions, large taper ratio or conicity, and short wheelbase, there is a trade off necessary to insure stability of secondary hunting on tangent track. These curving requirements all conflict with the requirements of high linear critical speed for secondary hunting on tangent track.

STABILITY

Linear

In providing good curving performance so that tracking errors are sufficiently small that flange contact does not occur, wheelset and truck design parameters may be chosen in such a manner that a potentially more dangerous condition is likely. The possibility exists that the linear critical speed for secondary hunting falls within the operating speed range.

Hunting involves sustained oscillation of the vehicle components in which the wheel flanges bang from rail to rail, the axles and truck yaw about a vertical axis as well as move from side to side, and the carbody responds in yaw and roll modes. This hunting behavior would occur even if the rails were perfectly alined and perfectly level. These oscillations severely degrade vehicle riding comfort and impose large and often damaging impacts on vehicle and track. Hunting occurs only above certain critical forward velocities. Although hunting will inevitably occur with all conventional railway vehicles, the critical speed at which this behavior first occurs can be increased beyond normal operating speeds by proper control of such vehicle design parameters as wheel tread profile, suspension characteristics, truck geometry, and vehicle weights.

Two very different modes of hunting behavior can occur. The most common mode is body hunting, or primary hunting, which can occur at speeds as low as 25 km/h and is common at speeds between 80 and 160 km/h. This mode is characterized by motions of the carbody and less distinct truck motions. Body or primary hunting usually occurs in a limited speed range, with both lower and upper bounds on the vehicle speeds at which hunting is present. This hunting mode is similar to a resonance phenomenon in several ways. First, the hunting usually begins when the frequency of the truck motion equals one of the natural frequencies of carbody motion. The dominant truck frequency is caused by the coned or hollowed profile of the wheel tread, and it increases nearly proportionally with vehicle speed. Thus, when this dominant frequency reaches one of the carbody frequencies, hunting may occur. Like resonance behavior, body hunting can be controlled by damping. If the carbody motion is sufficiently damped, body hunting can be eliminated entirely.

Truck hunting or secondary hunting, however, is inherent in the vehicle design. Unless the wheel profile is cylindrical, or the axles are free wheeling (without rigid connection between the two wheels on the same axle), truck hunting will always occur above a certain critical speed. However, each of these alternatives for avoiding secondary hunting has serious drawbacks. This hunting mode is characterized by the onset of diverging oscillations, growing to more violent truck motion and less pronounced carbody motion. Once truck hunting begins, it will continue to grow more violent as vehicle speed increases.

Approximations for the critical speeds of hunting have been found from stability analyses of the linear equations of motion for many different railway vehicles. In Great Britain and Japan significant effort has been devoted to correlating these linear stability analyses with actual rolling stock behavior. This work has utilized roller stands on which scale-model and full-scale vehicles are placed. The vehicle wheels are driven by the rollers at speeds equivalent to speeds in the operational speed range. Qualitative agreement has been obtained in many cases, but quantitative agreement between analyses and tests has not often been achieved. There have been few successful attempts to correlate the developing theory with actual performance on rails.

A brief discussion of the results of the numerous parametric studies will help clarify the nature and causes of hunting. Consider primary or body hunting first. This phenomena occurs when the frequency of truck motion coincides with one of the natural frequencies of the carbody on the secondary suspension. Two approaches can be taken to avoid such behavior. Like other resonance phenomena, body hunting can be controlled by introducing damping into the secondary suspension. The amount of damping needed to eliminate hunting depends on the vehicle design and the wheel/rail configuration. The available creep force also influences the required damping. Increasing the secondary suspension stiffness to increase the natural frequencies of body motion will achieve this objective. Increasing the wheel radius or reducing the wheel conicity will decrease the rate at which the kinematic frequency increases with speed, and consequently will increase the speed where this kinematic frequency of the truck or wheelset coincides with the body's natural frequencies. This approach is limited to low-speed vehicles since riding comfort dictates that the body's natural frequencies be kept fairly low. As the frequency of the secondary hunting mode increases with speed, in many cases it may coincide with the body's natural frequencies at speeds within the desired operating range.

Secondary hunting cannot be avoided, but by proper design it can be pushed beyond the operating range of railway vehicles. The linear critical speed of truck hunting is insensitive to large variations in creep coefficients. The suspension stiffness, wheel conicity, and truck geometry all have relatively large influences on the linear critical speeds.

The linear critical speed of secondary hunting for conventional, dual-axle trucks increases with increasing suspension stiffness. It has been found that the primary stiffness has a more dominant effect than the secondary on the critical speeds of truck hunting. The critical speeds for a dual-axle vehicle increase initially with suspension stiffness but beyond a certain point will decrease as stiffness increases further.

It should be noted that the concept of a rigid-frame vehicle, in which the wheelsets are held without play in the frame, is considered unrealistic for practical vehicles. Increasing wheel conicity always decreases the critical speed of hunting, while variations in mass, damping, and geometry within practical ranges have a somewhat smaller effect on the critical speed.

Nonlinear

The phenomenon of secondary hunting places important constraints on the maximum speeds attainable by railway vehicles. Designing for stable and highly damped secondary hunting directly conflicts with the design requirements for good curving performance. To insure the attainment of the desired characteristics of the secondary hunting mode, it is necessary to understand the influence on the response of wheel flange/rail contact, tapered wheel profiles, nonlinear creep, and nonlinear suspension characteristics.

No single analysis is known to have considered all the nonlinearities listed above, nor to have considered the degrees of freedom corresponding to perturbations in the longitudinal displacement and rotational motion of the wheelset about the axle. These degrees of freedom may be important when the wheel profiles are highly tapered, in the case of curve negotiation, and in the case of braking and/or traction. With few exceptions, most nonlinear analyses have been concerned with the secondary hunting of trucks and wheelsets.

The ordinary differential equations which describe the motions of the wheelset or truck during secondary hunting are nonlinear. Generally speaking, the equations are time-varying as the coefficients describe physical parameters that may vary with time. However, the assumption is made implicitly that the variations in the coefficients are sufficiently "slow" with respect to the natural frequencies of the vehicle that the coefficients may be considered to be constant. The equations are homogeneous or nonhomogeneous depending on whether forcing caused by rail irregularities is considered.

When the equations are linearized and small perturbations about the equilibrium point are considered, the solution of the resulting eigenvalue problem indicates the stability or instability of the equilibrium. It has long been recognized that the sustained oscillatory motions which characterize secondary hunting are limit-cycle oscillations rather than merely forced oscillations in response to track irregularities. However, it has been only recently that analyses have been made incorporating some of the more important nonlinearities that lead to the occurence of these limit-cycles.

In most cases, investigators have formulated the equations of motion using a Newtonian approach and have addressed the kinematics of wheel/rail contact in separate analyses. The results of these kinematic analyses have been incorporated into the dynamic analyses. A LaGrangian approach (i.e., equations of motion derived by LaGrange method) has also been used, with the kinematics of wheel/rail contact treated by considering two holonomic[2] constraint relations that express wheelset roll and vertical displacement as functions of the lateral displacement and yaw.

Analytical solutions (as opposed to computer solutions) of the nonlinear equations of motion of trucks and wheelsets are few. In one case, stability was examined for various initial conditions of a single, "free" railway axle not connected to a car or truck frame. Subject to various assumptions and approximations (the most important of which is that the forward speed is small), it was shown that the wheel and rail head profiles have a marked effect on the stability of the system. Another study analyzed the motions of a single wheelset with worn or curved wheel profiles. The lateral force exerted on the wheels by the rails under "flange contact" conditions was modeled as a linear spring with a deadband equal to the nominal flange clearance. The wheelset was considered to be connected to a truck frame by a primary suspension system consisting of linear springs that opposed lateral and yaw displacements of the wheelset with respect to the truck frame. The results of this analysis show that both stable and unstable limit-cycles may occur at speeds both above and below the linear critical speed, depending on the particular combination of design parameters. As previously discussed, the linear critical speed is that speed above which a linearized analysis predicts that small perturbations from the equilibrium position will increase with time. Below the linear critical speed, small perturbations from the equilibrium position will decay with time.

All of the above nonlinear analyses have considered the dynamics of the unforced system. Thus, the rails have been assumed to be perfectly straight and smooth. There have been a number of investigations of nonlinear truck and

[2]A holonomic constraint on a dynamic system is one which is represented by an algebraic relationship between the coordinates of the system.

wheelset dynamics that have used digital or analog computers to integrate the equations of motion. These investigations (discussed in Law and Cooperrider) have treated both the unforced case, where the rails are considered perfectly smooth and straight, and the forced response case, where various types of rail irregularities have been considered. Matsudaira analyzed the effects of side bearer friction on the stability of the secondary hunting mode for a dual-axle truck. Side bearer friction was modeled as dry or Coulomb friction in series with a linear spring that opposes yaw motions of the truck with respect to the carbody. To evaluate the effects of severe flange/rail contact, an analysis of secondary hunting was conducted by Matsudaira where repulsive action of the rail against the wheel flange was replaced by an equivalent spring force. The spring constants for these springs were considered to be functions of the amplitude of motion of the wheelset. At a certain speed above the critical value, the truck hunting, once generated, will build up its amplitude until the wheel flanges hit the rail, and finally the vibration will settle to a limit-cycle provided that no destruction of track occurs. Cooperrider, however, found that the influence on stability of yaw friction in the secondary suspension is much less significant for a six-degree-of-freedom truck than for the two-degree-of-freedom rigid truck.

Cooperrider also investigated the effects on truck dynamics of nonlinear creep, flange contact, and centerplate or side-bearer friction. This last nonlinearity was modeled as a linear yaw spring in series with a dry friction element. Both a simple truck with a rigid primary suspension and a complex truck frame were considered. Among results indicating the effects of forward speed, flange clearance, and adhesion level on hunting, it was found that flange contact can lead to sustained hunting at speeds well below the linear critical speed. The tracks were considered to be perfectly smooth and straight.

Stassen's study covered the effects of random lateral rail irregularities on the lateral dynamics of a dual-axle truck with rigid primary suspension between truck frame and wheelsets. Linear creep forces and spin creep effects were considered. The method of statistical linearization was used to obtain linear estimates of the nonlinear wheel/rail kinematic relations. These linear estimates are functions of the root mean square (rms) value of the lateral displacement of the wheelset relative to the irregular rails, the flange clearance, and the mean value of the difference between the actual gage and a nominal gage. The rails were considered to be rigid. The study concluded that the combined creep and spin components can be given by a linear function of the reduced tangential forces, and that introduction of nonlinear functions is not necessary to an adequate model.

The effects of flange contact on secondary hunting have been explored under the assumptions that the flange is vertical and that no relief of the load on the tread occurs during flange contact. The longitudinal component of the flange frictional force was also considered. For the configurations examined, no evidence of a sustained limit-cycle motion was found at subcritical speeds. Frictional resistance to yaw rotation of the truck was examined with the resultant finding that yaw stiffness of the secondary suspension can only be employed to raise the critical speed if adequate frictional yaw resistance is incorporated.

TRUCK CONCEPTS

INTRODUCTION AND DEFINITIONS

The fundamental objectives for rail vehicle truck and suspension design have been discussed earlier. They are guidance, shock and vibration isolation, traction and braking capability, and vehicle support. These objectives are accomplished principally through the mechanism of a truck which consists of a frame, two or more wheelsets (rigidly axle connected), and primary and secondary suspensions. The primary suspension is the elastic connection between axle journals and truck frame. The secondary suspension is between the carbody and the truck frame.

Shapiro[3] has extensively documented the engineering data of a significant number of high-speed trucks around the world. During this process he has developed a list of engineering parameters and their definitions valuable in reporting on truck technology. Therefore, definitions of some basic design parameters are reprinted below:

a. The conventional yaw pivot design involves a truck with a center pin arrangement between the truck and the carbody which allows the carbody to rotate with respect to the truck.

b. A soft primary suspension is a primary vertical suspension having the vertical bounce resonance (of the truck on its springs) of 8 Hz or less. The calculation of this resonant frequency considers the two-degree-of-freedom system of the carbody on its vertical springs coupled with the truck on its vertical springs. Typical practice is using a 1 Hz secondary.

c. A rigid truck frame is a truck frame which is considered to have no flexibilities.

d. A powered truck is one which has motors attached to the truck, either axle mounted or frame mounted.

e. A swing hanger design is one which has swing links connecting the truck frame with the bolster, and allows the truck to move laterally. This movement provides the secondary lateral suspension of the vehicle.

f. The term air spring secondary suspension pertains to trucks having air springs in the secondary suspension.

g. A roll bar is a device by which additional secondary roll stiffness can be provided for systems in which the secondary springs do not provide sufficient stiffness.

h. The term active tilt control applies to those vehicles having a system in which a device such as an accelerometer is used to sense acceleration levels. When these levels become too high, the tilting mechanism is activated.

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[3]Stephen M. Shapiro, "Engineering Data on Selected High-Speed Passenger Trucks," prepared for U.S. Department of Transportation, Federal Railroad Administration, 1977 (advance copy, not yet published).

i. The term articulated train applies to those trucks in which the ends of two car bodies rest on a single truck.

j. An equalizer beam truck is one in which load equalization of the truck is provided by the equalizer beam and its springs. This configuration allows the springs to be mounted longitudinally inboard of the wheels.

k. The term electromagnetic brakes refers to trucks which have brakes which react with the rail.

l. An anchor rod is a bar which takes longitudinal loads and is located either between the bolster and carbody or bolster and truck frame. There are two of these per truck.

m. A bolster is a load-bearing crossmember which is not rigidly connected to the truck frame.

n. An equalizer bar is a member located in the center of a truck to which the center pivot connects.

o. An equalizer beam is a structural piece with springs which provide equalization of the truck. There are two equalizer beams per truck.

p. A slide pad is a device which allows rotational motions to occur between the slider and the carbody.

q. A swing link is a link which permits lateral motions to occur and provides the lateral secondary suspension stiffness.

r. Traction linkages are linkages which take braking and acceleration loads.

Another design option which is presently in the developmental stage is radial action. Radial action is the ability of a truck to adopt and stably maintain a natural radial curving state in any curve, depending upon tread forces rather than flange forces for guidance through the curve. This topic will be discussed below.

In conventional trucks the major performance parameters are those involving suspension and steering. As for most machines, the primary function of the frame is structural containment of other components while they perform their respective functions. The role of the wheels is to guide and carry the burden imposed upon them. This is accomplished by selecting a wheel contour and conicity to match the speed and suspension characteristics of the truck.

Bonded polymer springs are finding increasing application to suspension design. While such springs can be used throughout suspension design, they are very strongly applicable as primary suspension elements in the form of axlebox springs. In general, they consist of alternating layers of rubber and metal, with the rubber layers being concave so as to create an accordionlike appearance for the spring as a whole. Rubber is a nearly incompressible material. Therefore, a compressive strain in the vertical direction will produce a pronounced balloon effect horizontally. The presence of the metal plates serves to control this effect in conjunction with the surface concavity of the rubber. The net result is that the spring is substantially stiffened in the direction of normal loads but still has shear flexibility. This leads to the chevron form of axlebox suspension, which is developed by bending the individual laminations into a chevron shape during manufacture. By installing two such springs in opposition to each other in an axlebox, with the springs vertically inclined to each other, the rubber is placed in bidirectional shear and monodirectional compression simultaneously. The compression serves as a stiffening influence as well as a means of preventing the spring from being torn apart in shear. Besides the compactness of design, the rubber also provides an energy-dissipation capability, with the result that the spring is really a spring and shock absorber combined.

The chevron technique has also been expanded for use as an air spring construction method for secondary suspensions. Two sets of cylinders are formed with the metal and rubber lamination technique, one using concentric laminations and the other a vertical stack of "donut" laminations. The cylinders are then connected in such a manner as to provide a sealed air void. The result is a secondary spring with vertical flexibility, lateral stiffness (not rigidity), and shock-dissipation capability in both directions.

Several high-speed truck descriptions, taken directly from Shapiro,[4] are presented below: the Y-28, Y-32, Y-225, and Y-226, all French designs; the Minden Deutz (German); the DT200 (Japan); the Pioneer-III (United States); and the BT10 (Britain). The ET403 description is based on Shapiro's work as well as on data gathered as a part of IPEEP. Of the remaining trucks, the HST (Britain), the LRC Coach (Canada), the SIG Type III (Switzerland), the Fiat ETR401 (Italy), the ASEA X-15 (Sweden), and that for the TGV-PSE trainset (France) are described on the basis of data gathered under IPEEP. The discussion of the TEP-70 (Russia) comes from Khlebnikov.[5]

Y-28 (FRANCE)

The Y-28 truck is used on the Trans Europe Express (TEE) coaches for 200 km/h service. The primary suspension is composed of four groups of coiled springs and four groups of silentbloc materials of considerable size. Each group is located outboard of the wheels. The silentbloc material which connects the axle to the truck frame provides sufficient stiffness in the lateral and longitudinal direction to keep the axles relative to the truck frame. In the vertical direction it adds to the stiffness of the coil springs.

The secondary vertical suspension is made up of two groups of two coiled springs per truck and two hydraulic shock absorbers to control the vertical movements. There are two swing links between the body and bolster which are articulated at the lower end to two body brackets represented by a ground connection. These swing links are located in line with the lateral axis of the truck and provide the lateral suspension. There is a lateral shock absorber to

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[4]Shapiro, ibid.

[5]J.V. Khlebnikov, "The New U.S.S.R. TEP-70 Type Passenger Diesel Locomotive," Rail International, Nov., 1976.

control the lateral displacement between the body and the truck. A roll bar helps to control the rolling motions between the truck frame and bolster. It is supported under the truck frame crossmember and connected by links to the bolster.

The Y-28 truck has no conventional yaw pivot for steering. It is driven by means of traction linkages arranged longitudinally between the truck frame and two crutches forming part of the body represented by ground connections. The connection is more or less at axle level. These traction linkages have resilient attachments to relieve the linkages of shock loads. The resilient attachment includes helical spring and Belleville washers, resulting in a progressively varying degree of elasticity. Stops engage if force on the wire rope reaches 12 metric tons, which is rare. The ultimate tensile strength is 24 metric tons.

The vertical and lateral carbody loads are transmitted through carbody brackets to the swing links into the secondary suspension. Then the load passes through the truck frame and the primary suspension. The longitudinal load path proceeds from the carbody crutches or brackets, through the traction linkages to the truck frame, and finally goes out through the primary suspension.

Y-32 (FRANCE)

The Y-32 truck is designed for the European standard truck and for speeds in excess of 200 km/h. The primary suspension is composed of four steel coiled springs for vertical stiffness. Flexible connections between the axle box and truck frame result in three-dimensional flexibility which gives longitudinal and lateral stiffness. In the vertical direction this flexibility adds to the stiffness of the coil springs to provide the required vertical spring rate. There are four hydraulic shock absorbers on the truck to restrict vertical motions.

The secondary suspension consists of two long helical springs for vertical stiffness. In addition, these helical springs provide for lateral suspension of the body and enable the truck to rotate. There are hydraulic dampers controlling the vertical and lateral movements of the coach in relation to the truck.

The truck has a unique feature because there is no conventional center pivot. The connection between truck and body is made by traction linkages located at axle level and resiliently attached to the carbody to permit vertical and lateral motions of the truck across the secondary suspension.

The truck has a roll bar fitted between the truck frame and the carbody to provide additional roll stiffness. Another feature of this truck is a yaw damping arrangement relying on the truck which rotates in relation to the body and damps only yaw rotational motions.

The carbody vertical and lateral loads are transmitted through the long helical springs into the truck frame and go out through the primary suspension. The longitudinal loads proceed from the carbody brackets, through the traction linkages, into the truck frame, and out through the primary suspension.

Y-225 (FRANCE)

The Y-225 truck, used on the French turbotrain TGV-001, is designed for speeds in excess of 200 km/h. In a test run in December 1972, the TGV001 attained a speed of 318 km/h. By December 1974, TGV-001 had run more than 16,000 km at speeds greater than 260 km/h and had made over 100 runs at more than 300 km/h. The TGV-001 is composed of two power cars with three trailer cars between them. The five cars are supported on six trucks, all of which are powered. The TGV is an articulated train.

The primary suspension consists of eight sets of helical springs per truck, resting on the axle box brackets and in series with rubber bearers which insulate the body from sound vibrations. This suspension system provides vertical, lateral, and longitudinal stiffness. Four hydraulic antipitching dampers complete the system.

The secondary suspension consists of two Sumiride air springs per truck. One air spring is on each side of the truck and rests on a bearer on the lateral suspension. Two lateral links per bearer insure that the air springs operate vertically only. The vertical damping comes from the air springs. The air systems for the two springs are connected by a differential valve to insure that in the event of failure of one of the springs, the carbody will drop vertically on two rubber stops.

The secondary lateral suspension is made up of four Kleberman Colombes metal rubber sandwiches per truck installed in sets of two giving a frequency of 0.8 Hz. The deformation of the sandwiches in shear permits the rotation of the truck in relation to the body. Two stops, each exerting its effort gradually, limit the total lateral displacement between car and body to ±80 mm. A hydraulic shock absorber provides control of lateral movements.

A yaw damping arrangement in the secondary suspension, composed of a bar with hydraulic dampers connected to the carbody, is located on both sides of the truck. This device also keeps the truck on the line bisecting the angle between two adjacent carbodies.

The vertical load is transmitted from the carbody through the air springs into the rubber sandwiches to the truck frame, and then is taken out through the primary suspension. The lateral load goes from the carbody, through the lateral links and the rubber sandwiches to the truck frame, and out through the primary suspension.

The longitudinal load goes from the body through the T-shaped member in the center part of the truck into the truck frame central crown member and then out through the primary suspension. This T-shaped member is resiliently mounted within the truck central crossmember and is restrained laterally and longitudinally but allows vertical motions.

Y-226 (FRANCE)

The Y-226 truck (figures 3 and 4) is used on experimental motor coach Z7001. Since starting tests in April 1974, motor coach Z7001 had run 125,000 km by the end of January 1978, and had made over 100 runs at speeds between 250 and 360 km/h.

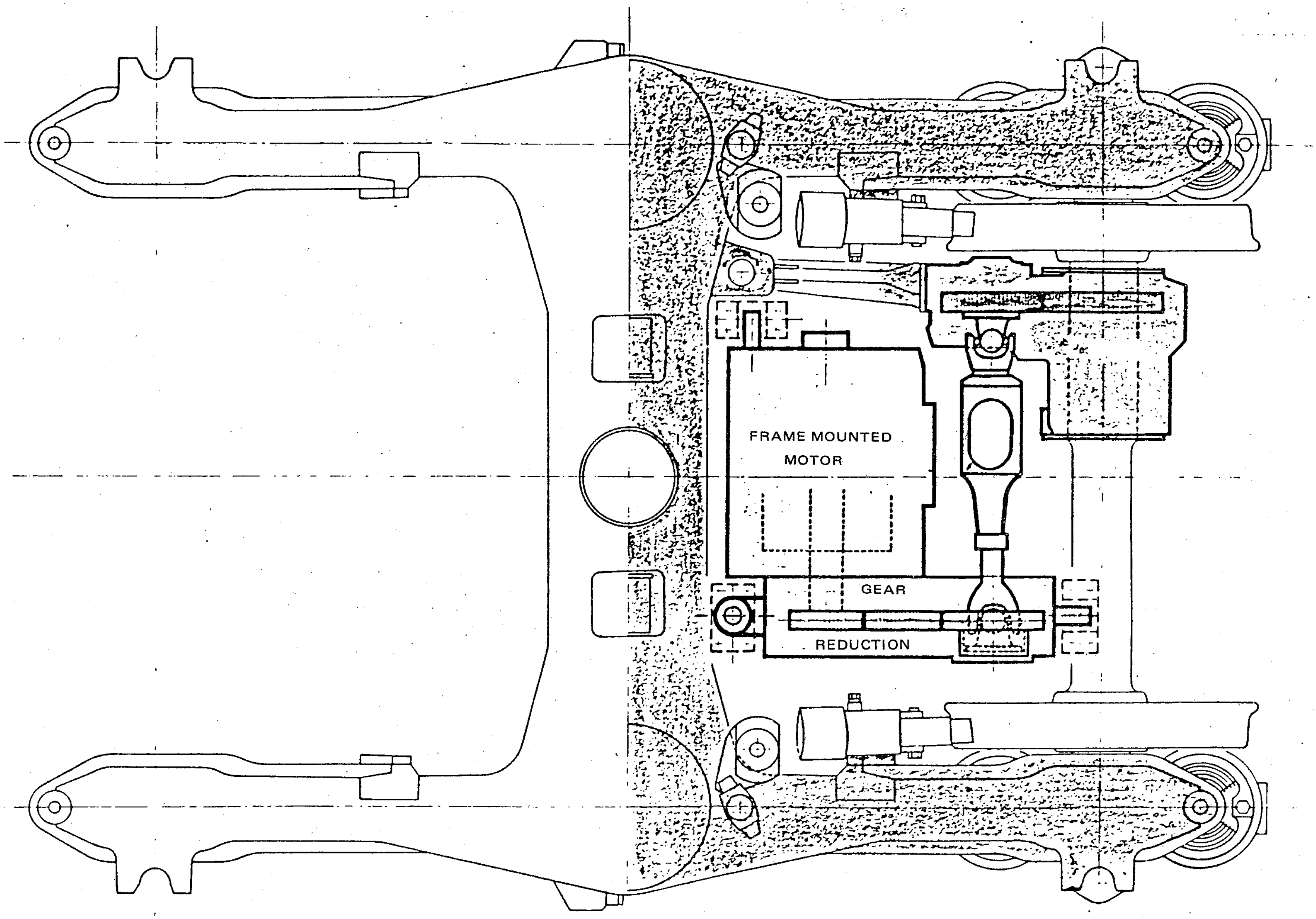
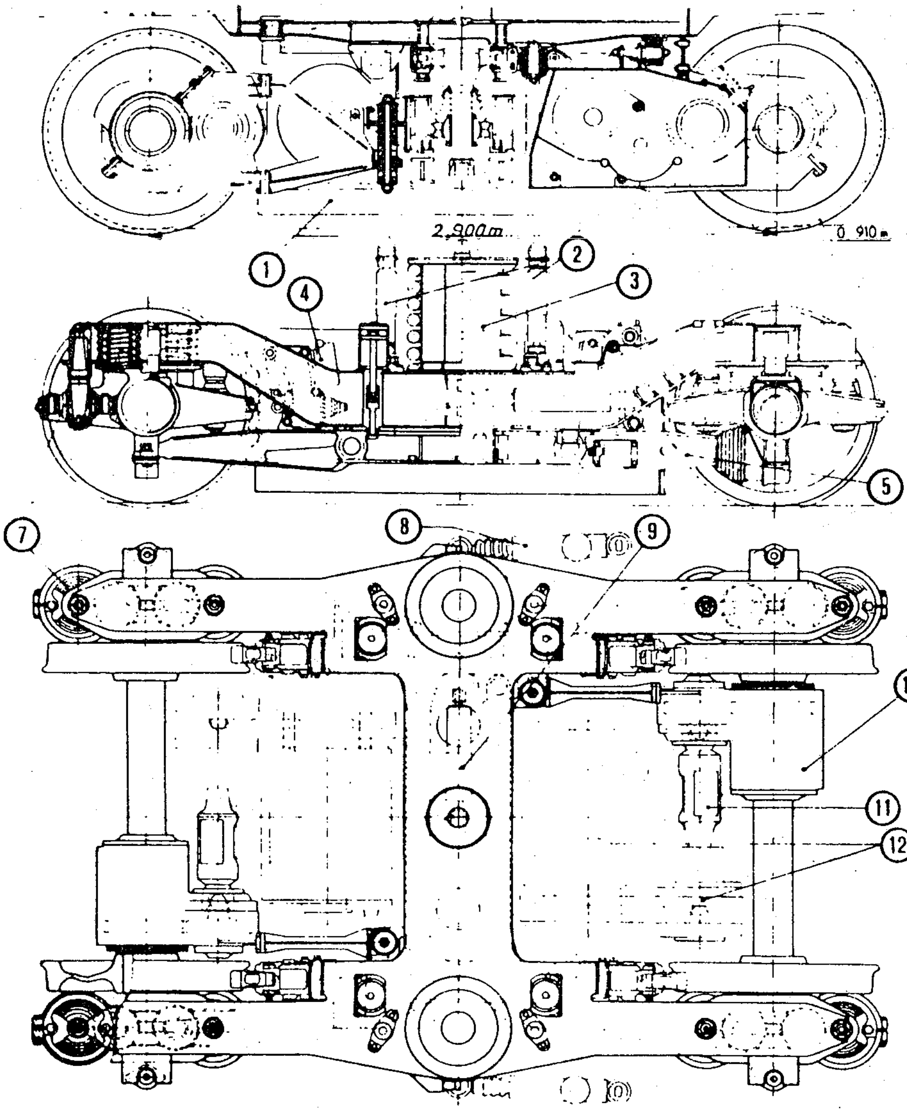


FIGURE 3. FRENCH NATIONAL RAILWAYS TGV-PSE TRAINSET POWERED TRUCK (Y-226)



1. LINEAR EDDY CURRENT BRAKE
2. DAMPERS IN THE SECONDARY SUSPENSION
3. SECONDARY SUSPENSION
4. BRAKE EQUIPMENT
5. BEAM SUPPORTING THE LINEAR BRAKE
6. MEMBER TRANSMITTING FORCES BETWEEN VEHICLE BODY AND TRUCK
7. PRIMARY SUSPENSION
8. ANTIHUNTING DAMPER
9. TRANSVERSE DAMPER
10. DRIVING GEARING (REDUCTION RATIO: 1.15)
11. SLIDING TRIPOD TRANSMISSION
12. TRACTION MOTOR AND REDUCTION GEARING (MOUNTED, BENEATH THE VEHICLE BODY, REDUCTION RATIO: 1.39)

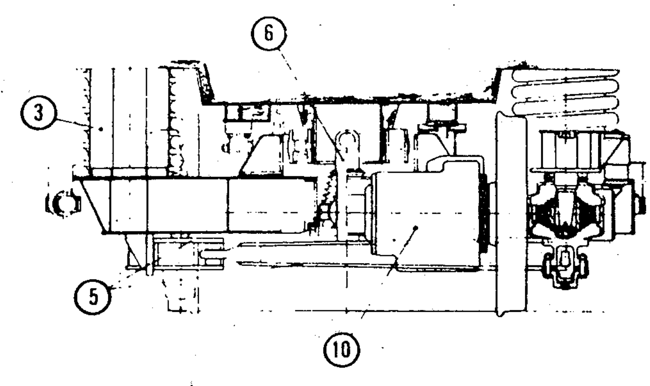


FIGURE 4. Y-226 TRUCK ASSEMBLY

The primary vertical suspension system consists of eight helical springs and four vertical hydraulic dampers. Two vertical guides on the Y-226 fit into brackets in the axle box casting, and the primary longitudinal and lateral suspension consists of alternate steel and rubber rings around these guides. This arrangement allows the stiffness to be varied in the longitudinal and lateral directions.

The secondary suspension is composed of two large helical springs enclosing a rubber cylinder located at the ends of the truck frame central crossmember. This system allows vertical, lateral, and rotational motions. Four hydraulic dampers restrict vertical motions. Rubber stops are arranged to provide progressively increasing resistance so that the lateral truck body movements are limited to 70 mm. There are two longitudinal and one lateral damper per truck. The truck rotation is limited by four stops.

The vertical and lateral loads are transmitted from the car through the long helical springs with rubber inside into the truck frame and out through the primary suspension. The longitudinal loads are taken through the center pivot, into the truck frame, and out through the primary suspension.

MINDEN DUETZ (GERMANY)

This particular Minden Deutz truck is designed for speeds of 200 km/h or higher and was one of the four designs submitted to the International Union of Railways (UIC) for use as the European standard truck. The truck is a swing-hanger-type design.

The primary suspension is composed of eight helical coil springs for vertical motions and eight leaf links for lateral and longitudinal motions. There are four hydraulic dampers for restricting vertical motions.

The secondary suspension comprises four helical coil springs mounted between the spring plank and the bolster. There are two vertical hydraulic dampers. The lateral stiffness comes from the swing links which are supplemented by rubber stops. There is one lateral damper located between the bolster and spring plank.

An added feature of the truck is rotational restraint, which consists of link rods to guide the bolster and prevent it from moving longitudinally. In addition, the truck has a roll bar with spherical joints to allow free lateral motion of the bolster.

The vertical and lateral loads are transmitted from the carbody through the center pin into the bolster down through the coiled spring secondary to the spring plank into the swing links. The loads then go through the truck frame and finally out through the primary suspension. The longitudinal load is transmitted through the center pin into the bolster, through the long link rods into the truck frame, and out through the primary suspension.

DT200 (JAPAN)

The DT200 is a truck which is used on the Japanese New Tokaido Line and is designed for speeds of 200 km/h between the cities of Tokyo and Osaka. The track is standard gage 1,435 mm.

The primary suspension is composed of eight helical coil springs for vertical movements, and eight leaf links, similar to the Minden Deutz truck, with rubber bushings at the end for lateral and longitudinal motions. Four vertical dampers restrict vertical motions.

The secondary suspension consists of two air springs which provide vertical and lateral stiffnesses and also restrict vertical motions by way of air damping. This vertical air damping is supplemented by two hydraulic dampers. There are two lateral hydraulic dampers in the secondary. The longitudinal stiffness comes from two anchor rods which connect the bolster to the carbody.

The vertical load is transmitted from the carbody through the air springs, into the side bearers and out through the primary suspension. The lateral load is transmitted through the air springs into the center pivot, and out through the primary suspension. The longitudinal load is transmitted from the car, through the anchor rods into the center pivot, and out through the primary suspension.

PIONEER-III (UNITED STATES)

One version of the Pioneer-III (PIII) truck was designed for use on the Amcoaches. The Amcoaches are locomotive hauled and capable of speeds up to 193 km/h.

The P-III primary suspension consists of four rubber rings between the axle and the side frame and four side bearers between the side frames and the bolster. There are no hydraulic dampers in the primary. The primary suspension is relatively stiff in comparison to the other high-speed trucks described.

The secondary vertical suspension consists of coil springs in series with air springs. The lateral stiffness is obtained by the shearing of the coil springs in series with lateral stabilizing rods having rubber bushings at the ends. There are two Houdaille rotary shock absorbers in both the vertical and lateral directions for restricting these motions. Two anchor rods connected between the bolster and the carbody restrict longitudinal motions. The P-III has an articulated frame with independent side frames.

The vertical load is transmitted from the carbody through the air and coil springs to the side bearers and out through the primary suspension. The lateral load goes from the carbody through the coil and stabilizing rods, through the center pivot to the truck frame, and out through the primary suspension. The longitudinal load goes from the carbody through the anchor rods, into the center pivot to the truck frame, and out through the primary suspension.

ET403 (GERMANY)

The ET403 is a 200 km/h, four-car, electrically propelled train with all trucks powered.

The truck is fabricated steel, weighing a total of 12 metric tons, 4 tons of which comprise the traction motor and gearing, while another 4 tons are contributed by the wheelsets and journal bearings. Thus, the truck frame and its suspension apparatus proper weigh only 4 metric tons. The truck utilizes a rigid welded-steel frame. Primary suspension is by means of four coil springs and four vertical hydraulic dampers. The longitudinal and lateral stiffnesses are provided by two leaf-spring-type linkages and elastic boxes.

These linkages and elastic boxes can be changed to alter the longitudinal and lateral stiffnesses during experimental determination of 200 km/h stability. If a spring leaf should break, there is an auxiliary guidance device which maintains location of the wheelset. The traction motors are fastened to the truck frame. The drive to the wheelset is by means of a cardan shaft going through the center of the gear box.

The secondary suspension is made up of two M.A.N. air springs supported on laminated hollow rubber block springs which provide vertical and lateral stiffnesses and also allow rotation. The air springs are used in shear to swivel the truck because there is no bolster. After 15 mm air spring lateral displacement during negotiation of curves, the air spring suspension is supplemented by additional transverse rubber springs which give a progressively increasing lateral spring rate. The secondary suspension has two hydraulic shock absorbers for damping vertical motions and one shock for damping lateral and rotational motions by being located a distance from the lateral truck centerline. The air springs are also used for load weighing for the brake system.

The vertical and lateral loads are transmitted from the carbody to the truck frame by the air springs and then go out through the primary suspension. The longitudinal load goes through the center pin and the center pivot into an equilizer bar through rubber-bushed rods, then into the truck frame, and out through the primary suspension.

The design of the truck is evolutionary, based on the truck used on the ET420 train, which has been in service for some time. The truck of the ET420 was developed for speeds of up to 120 km/h, and a relatively stiff guidance was used for the wheel and axle set in the x-direction. For higher speed, the ET403 was provided with a softer wheel and axle set guidance in the longitudinal direction which, in conjunction with concave tread profiles, allows a certain radial alinement on curves and enables side forces between rail and wheel to be reduced.

HST (BRITAIN)

The British Rail High-Speed Train (HST) consists of a seven- to eight-car train with a power car at each end. The following description is for the power car truck (figure 5), developed by British Rail Engineering, Ltd.

The trucks are of all-welded construction, with a service life goal of 25 years. Much computer design work was done to achieve the lowest possible weight. The primary suspension uses coil springs with Alsthom links. The secondary suspension is of the flexicoil type, with direct-acting hydraulic dampers used to control primary and secondary vertical movements, lateral movements, and truck yaw. Traction and braking forces are transmitted between the truck and body by two laminated rubber and steel plate assemblies in the truck center well, located in the front and in back of the center pin.

To reduce the unsprung weight, the traction motors are carried completely on the truck frame at three points, with the gearbox attached to the frame by a rubber reaction link. A flexible drive train is used to accommodate up to 65 mm movement between traction motor and axle-mounted gearbox. In addition, hollow axles and lightweight solid wheels are used. The wheel profile is the RPH hollow tread. No flange root wear and only 4 mm tread wear have been experienced in 400,000 km of use.

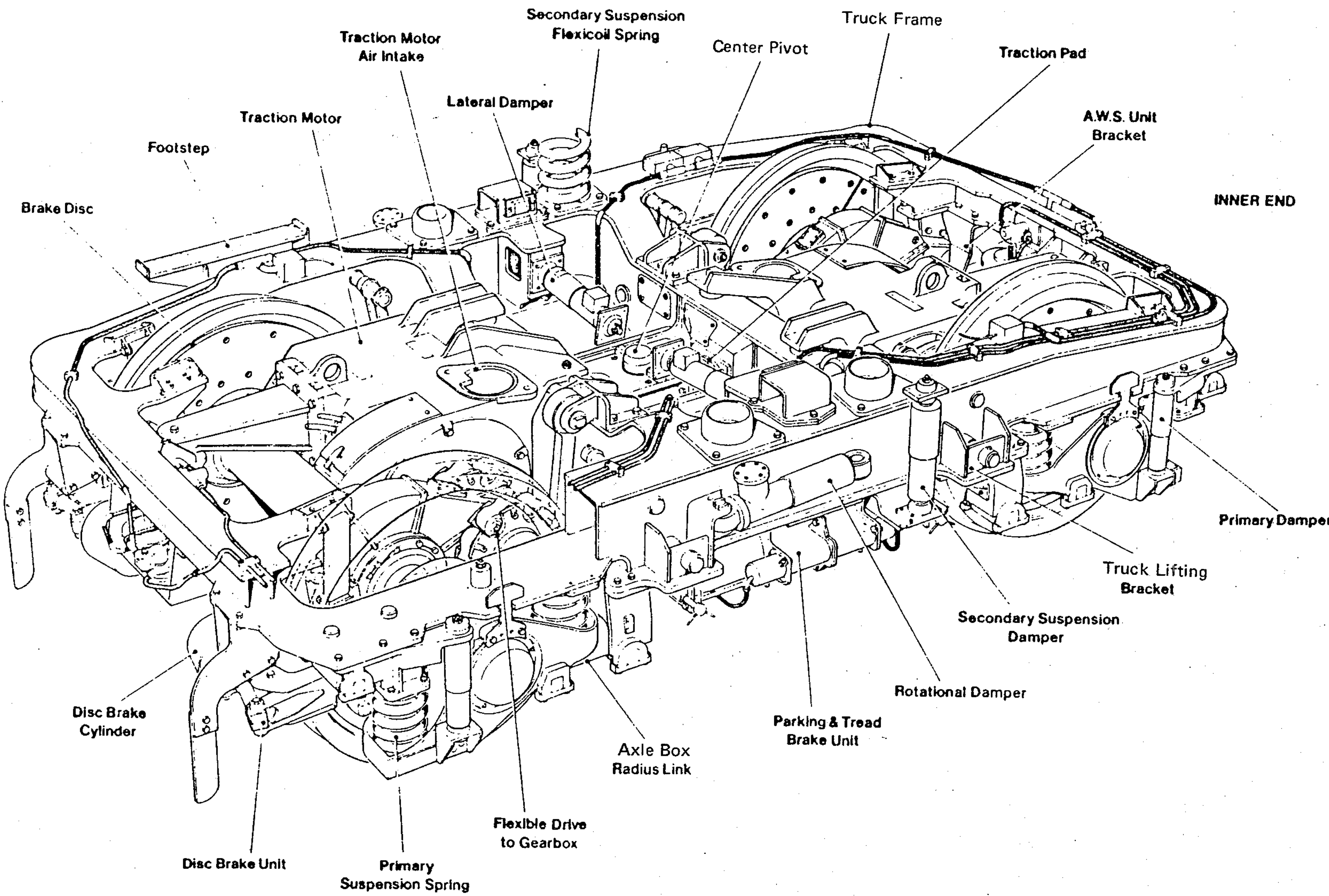


FIGURE 5. HST POWER CAR TRUCK

The braking system uses Girling disc brakes operating on brake discs bolted directly to the wheel cheeks of each wheel. In addition, a cast iron brake shoe is used as a "scrubber" on each wheel tread to maintain good adhesion and for use as a parking brake.

British Rail claims that this truck, which has a design speed of 200 km/h, is in fact stable to 230 km/h and that it appears to be able to operate 400,000 km between overhauls. The design principles of the unpowered trucks used on the HST coaches are somewhat the same as those described above for the power truck. The basic difference is that the unpowered trucks are of swing-hanger design and make use of air springs for the secondary suspension.

BT10 (BRITAIN)

The BT10 truck (figure 6) has been used on British Rail's High-Speed Train passenger coaches. This high-speed diesel train was tested at a speed of 225 km/h in June 1973 and is composed of two power cars and several intermediate passenger coaches.

The primary suspension consists of four helical springs for vertical stiffness. Axle box radius arms, which are pinned through rubber bushings to the frame, provide the lateral and longitudinal stiffnesses. Four hydraulic dampers restrict the vertical motions between the frame and the axle.

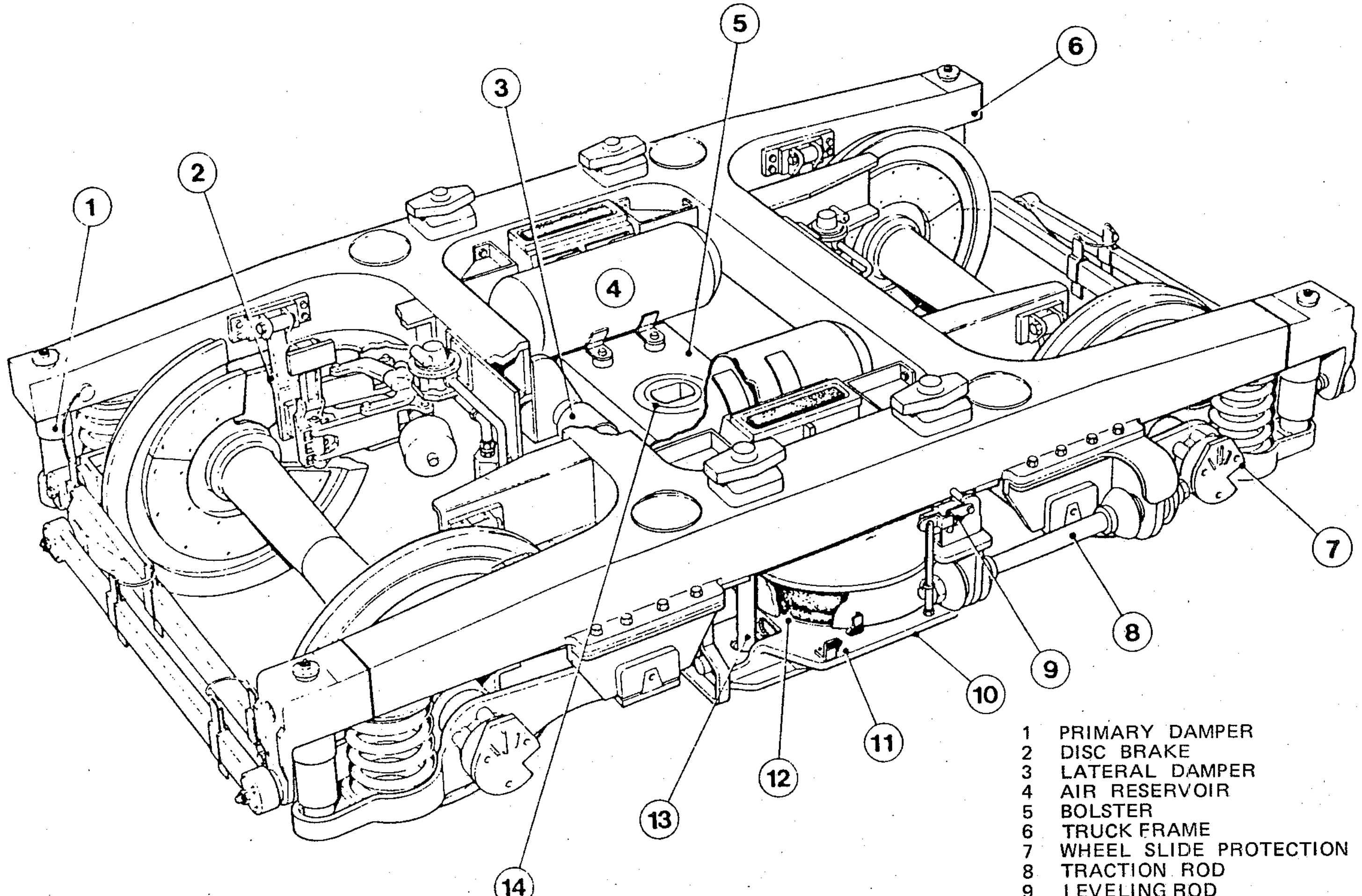
The secondary suspension is composed of two diaphragm air springs for vertical motions. These are located between the spring plank and bolster. Four swing links which connect the truck frame to the spring plank provide the lateral stiffness. The longitudinal secondary stiffness comes from two anchor rods connected between the bolster and the truck frame.

The vertical load path is transmitted from the body through the slide pads, into the bolster, through the air springs, up the swing links to the truck frame, then out through the primary suspension. The lateral load path is carried from the body through the center pin and the bolster. The load goes down the air springs to the spring plank, up the swing links, then out through the primary suspension. The longitudinal load is transmitted through the center pin out to the sides of the bolster, through the anchor rods to the truck frame, and out through the primary suspension.

TEP-70 (RUSSIA)

The TEP-70 is a 160 km/h, 4,000 hp, ac/dc passenger diesel locomotive built by the USSR Kolomna Diesel Locomotive Plant and specifically designed for use where heavy freight and high-speed passenger traffic operate over the same lines.

Such conditions call for a locomotive capable of high maximum speed on a periodic basis while slower freight trains are not blocking the way. In this manner, average passenger train speed can be increased. The trucks are the same as those used on the TEP-60, an older 3,000 hp locomotive. The TEP-70 is a single-power-section, two-cab locomotive with an ac-to-dc drive. The major design problems are keeping the weight at the TEP-60-type constant while increasing horsepower, service efficiency and reliability, and reducing maintenance costs. Continuous speed is 50 km/h; continuous tractive effort is 17,000 kg. Service weight with two-thirds capacity of fuel and sand stored is 129 metric tons plus 3 percent.



1 PRIMARY DAMPER
2 DISC BRAKE
3 LATERAL DAMPER
4 AIR RESERVOIR
5 BOLSTER
6 TRUCK FRAME
7 WHEEL SLIDE PROTECTION DETECTOR
8 TRACTION ROD
9 LEVELING ROD
10 SPRING PLANK
11 AIR SUSPENSION INDICATOR
12 AIR SPRING
13 SWING LINK
14 CENTER PIVOT

FIGURE 6. HST Mk. III COACH, BT10 TRUCK.

The description that follows of the truck and suspension system of the TEP-70 is from Khlebnikov.[6]

The truck of the TEP-70 Passenger Diesel Locomotive, with individual axle drives, has a balanced primary spring suspension and a nose-frame-suspended traction motor. The body is supported on the truck frame through helical springs which take up oscillations caused by rolling, and through rubber dampers of the central supports which allow vertical movement. Each truck has two central (pendulum) supports along the longitudinal axis and four helical springs of the secondary suspension. The problem of improving the horizontal dynamics has been solved by subdividing the mass of the body and trucks in the horizontal plane by using pendulum supports, elastic cross movement of all axles with axle boxes, and free cross movement of middle wheelpair axle with respect to the axle box.

The central supports are made in the form of a rod, the ends of which are fitted with rubber shock absorbers that cushion the vertical and horizontal jolts and reduce high-frequency vibrations transmitted to the body. Spring braces pretensioned to 1,500 kg keep the central supports in a vertical position.

The secondary suspension stage comprising the rubber elements and springs renders more damping of rolling action than of the longitudinal oscillations; the static spring deflection of side bearings under working load is 98 mm.

The pendulum supports transfer the traction and braking forces from the truck frames to the body. The midparts of the rod supports have recesses for accommodating stops made of wear-resistant steel and mounted in the sockets of body frame bolsters.

The load from the body and truck frames is transferred to the wheelpair axles through axle boxes with rubber shock absorbers in the link hinges. The rubber and metal shock absorbers in the hinged connection of the axle box with the truck frame insure a practically free movement vertically and an elastic movement (up to ±8 mm) laterally.

There is practically no axle box movement in relation to the truck frame. To limit the amplitude of vertical oscillations, the truck frame is supported on the axle boxes through a system of coil and laminated springs. The primary suspension stage on each side of the truck consists of two laminated and six helical cylindrical springs joined together longitudinally by three axle box rocker arms and two spring rocker arms. The helical springs are mounted on both ends of the axle box balancer, its center hinge suspended from the bottom point of the axle box housing. The laminated springs are joined by hangers to the spring rocker arms, the ends of which are supported on the central helical springs mounted on the axle box rocker arm. Rubber shock absorbers are used to absorb high-frequency and large short-time vertical forces in those assemblies which directly transfer the load (i.e., between the supporting parts of the truck side frames, the extreme helical springs and buckles of the laminated spring are fitted with rubber shock absorbers).

[6]Khlebnikov, ibid.

The static deflection of the primary spring suspension is 94 mm. The rigidity of the primary suspension, taking account of the rigidity of the link elements of the axle boxes, is 758 kg per cm. To improve the horizontal dynamics of the locomotive, the middle wheelpair of the truck, besides having an elastic lateral because of the rubber shock absorbers in the axle box link, can move freely laterally 14 mm on both sides of the axle box. This is made possible by eliminating the thrust ball bearing used in the extreme axle boxes where the axle is positioned with respect to the axle boxes. The lateral travel in this case is limited by flat thrust rings of the radial roller bearings. The traction motor is fixed to the truck frame at three points through rubber-metal blocks on which the motor is supported, with washers for positioning it with respect to the wheelpair.

LRC COACH (CANADA)

The LRC is a Canadian high-speed train made up of a diesel locomotive and tilting-body coach. While it is based on present-day technology, it has an advanced suspension system for both locomotive and coach. The following information concerns the coach truck (figure 7).

While a major feature of the LRC is the body-tilting system, the basic truck is also quite refined. Having a design speed of 200 km/h, the truck is based on a rigid cast steel H-frame. The frame ends are inboard of the wheels. Primary suspension is by angled laminated-rubber chevron springs, which provide elastic cushioning of axle motions in yaw and lateral translation. Rotary hydraulic dampers mounted at the axle boxes control frame bounce and pitch motions. Forged steel axles and 176 mm diameter rolled steel wheels are used.

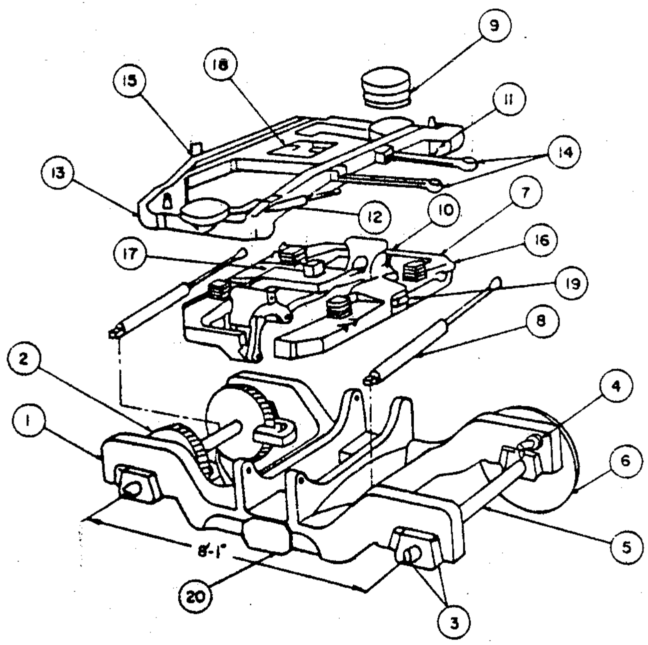
Secondary suspension is by large air springs, spaced at the maximum width possible. Direct-acting hydraulic dampers are used. The air springs rest on a transverse locating link. The spring plank is supported on the tilting bolster by four laminated rubber bearing pads. The truck center post projects upward from the tilt bolster and engages precompressed laminated rubber traction pads attached to the spring plank. Lateral suspension is provided by the combined shearing of the traction and spring plant support pads which provide an elastic restraint of truck location, since the spring plank is not free to pivot.

Inboard axle-mounted disc brakes are used, with two discs per axle. No tread brake is used.

SIG TYPE III (SWITZERLAND)

The Type III car is a new lightweight vehicle built by the Swiss Industrial Company (SIG) for the Swiss Federal Railway. It is an updated version of the existing Type III rolling stock, the major change being the addition of a tilting suspension system. Both types of cars are locomotive hauled.

The Type III truck (figure 8) is of welded fabrication. The primary suspension is connected to the journal boxes by means of bellcranks. The primary springs are made of rubber but may be replaced by coil springs lying horizontally under the truck side frame. Main pivot bearing friction is used for vertical damping.



1. FRAME
2. DISC BRAKES
3. CHEVRON SPRINGS
4. ROTARY DAMPERS
5. AXLE
6. WHEEL
7. TILTING BOLSTER
8. HYDRAULIC CYLINDER
9. AIR SPRING
10. SWING LINKAGE
11. VERTICAL DAMPERS
12. LATERAL DAMPERS
13. SPRING PLANK
14. TRACTION BARS
15. TRANVERSE LOCATING LINK
16. BEARING PADS
17. CENTER POST
18. TRACTION PADS
19. ACCELEROMETER
20. PREVENTER

FIGURE 7. LRC COACH TRUCK

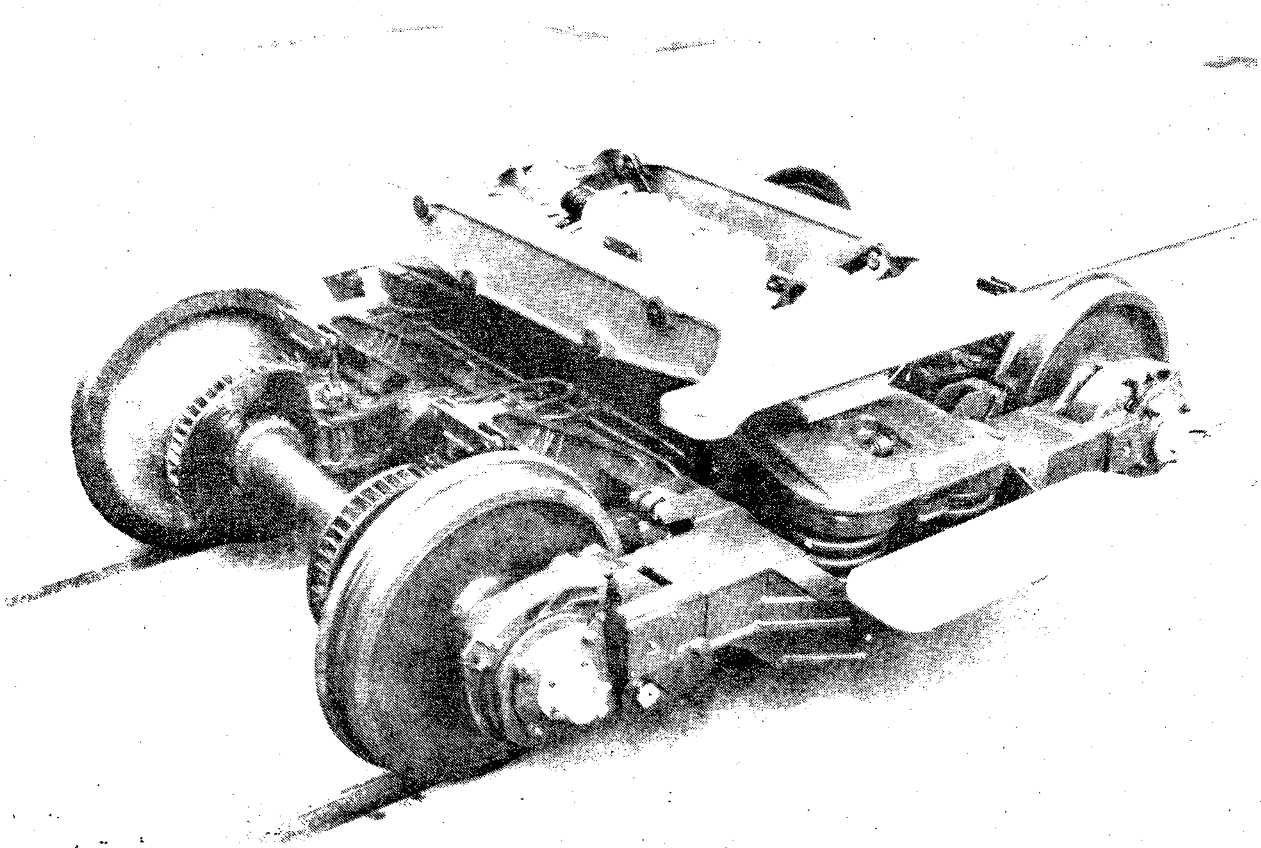


FIGURE 8. TYPE SIG MQS-sk TRUCK FOR STANDARD CAR TYPE III OF SWISS FEDERAL RAILWAYS.

Heavy-duty bearings are used for the rollers of the tilting bolster and handle both vertical and longitudinal loads. No center plate is used; the weight is carried by the rollers. Secondary suspension is by means of flexicoil springs, which are sheared to allow the truck to turn. Thus, there is little resistance to yaw. Vertical and lateral movements are controlled by hydraulic shock absorbers. The Swiss Federal Railway worn-wheel profile is used and is said to give an acceptable ride. The design speed of the Type III truck is only 140 km/h.

A combination disc/tread braking system is used. Two disc brakes are used per axle, and a tread brake shoe is used at each wheel.

FIAT ETR401 (ITALY)

Fiat has developed a tilting-body trainset commonly called the "Pendolino." It is an electric MU (Multiple Unit) train, and its official designation is ETR401. The truck used on the ETR401 is shown in figure 9.

The truck frame is made up of a linked chassis, connected by helical springs and rubber elements to the axle boxes, with spring rates designed to give maximum lateral stability. One of the highlights of this truck is the use of lateral air springs to provide some radial steering for the trucks. The normal movement of these air springs, plus the slewing of the truck on its secondary suspension, totals ±110 mm. The body is supported by swing hangers from a bolster which rests on the flexicoil springs of the secondary suspension. Traction and braking efforts between the truck and the bolster are transmitted by rubber buffers on the longitudinal axis of the truck.

To reduce unsprung weight and allow a softer primary suspension to be used, all traction motors are mounted on the carbody. The inner axle of each truck is driven by a cardan shaft; the outer axle is unpowered. Two disc brakes are used per axle, along with an electomagnetic track brake for emergency use. Dynamic braking provides most of the braking effort.

The pantographs of the trainset are supported on a frame which is mechanically attached directly to the bolster. This frame extends the height of the car and is independent of the body. This allows the pantograph to be centered at all times on the catenary, regardless of the tilt of the carbody.

ASEA X-15 (SWEDEN)

The ASEA X-15 is an experimental three-car MU train, using 30-year-old equipment as a test bed to experiment with new concepts in truck design and tilting-body suspension. The results from these tests will be used as the basis for the X-2, a high-speed trainset presently under design by ASEA for the Swedish Railways. Details are rather unclear, and the following is what is known of the truck design.

The truck frame is a fabricated assembly. The primary suspension is by chevron springs, which are mounted on the journal boxes. A combined vertical and lateral damper is used, mounted at a 45° angle between the journal box and truck frame. There are two bolsters: The truck bolster and above that, the tilting system bolster, which is supported on its bearings. A long traction

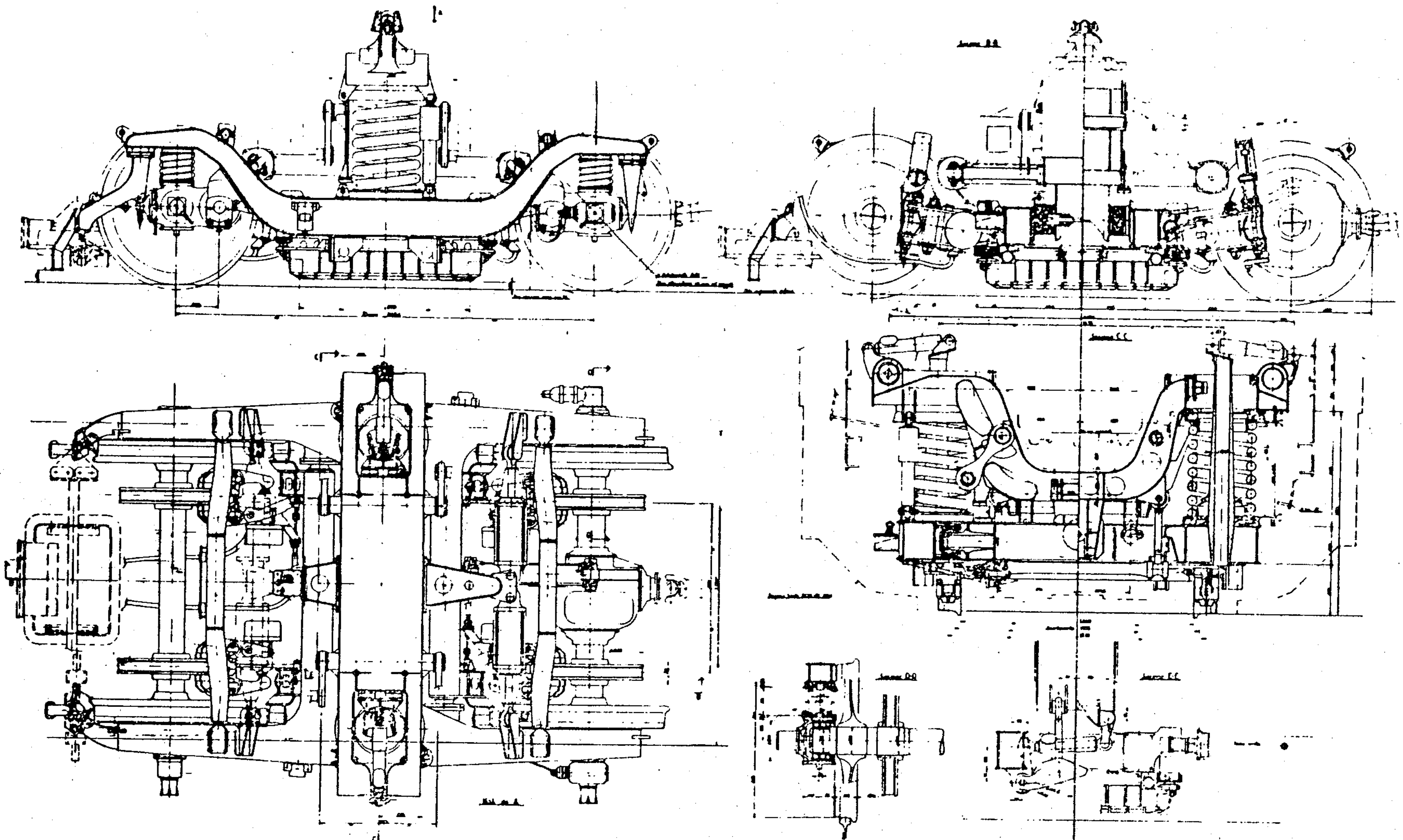


FIGURE 9. FIAT ETR401 TRUCK.

rod is mounted between the bolster and the side sill of the car, and yaw dampers are connected between the bolster and the truck frame.

Secondary suspension is provided by air springs mounted on the tilting bolster, with lateral and vertical dampers used, along with rubber lateral stops. The tilting system is hydraulic, with controls provided to operate a separate tilt mechanism for the pantograph. While the present trucks are unpowered, provisions have been made for mounting traction motors on the trucks in the future.

A combination of disc and tread braking is used. Two inboard-mounted disc brakes are used on each axle, and tread brakes are located at each wheel.

ASEA feels that this truck is able to steer itself with very low track forces. This radial steering is done by means of the chevron springs, which allow fore and aft movement, while providing the vertical and lateral primary suspension.

TGV-PSE (FRANCE)

The TGV-PSE is a new electric trainset being built for the French National Railways (SNCF) for high-speed service between Paris and Lyon. Each train will consist of eight articulated coaches with a power car at each end. Both trucks on each power car, and the coach truck next to the power car, will be equipped with traction motors; the other trucks, while unpowered, will be of similar design.

The trucks to be used, SNCF Type Y-229 (figure 10), are based on the SNCF Type Y-226 truck. A refinement of past designs, the Y-226 has a low, unsprung weight, with as much weight as possible along the truck centerline. The secondary vertical suspension is very soft, with two long flexicoil springs located in an articulation frame between carbody sections. This permits a high support and minimizes body roll caused by curving forces. Hydraulic dampers are used to prevent yaw, and special plastic bushings are used at the spring ends to prevent transmission of noise and vibration to the carbody. SNCF believes that coil springs can provide as smooth a ride as air springs, with a higher degree of reliability. The mounting of the springs on the carbody is as high as possible. The design speed is 300 km/h.

To reduce unsprung weight, the powered trucks have the traction motors mounted on the carbody as close as possible to the center pin. A gearbox is located on the traction motor, and also on the truck frame, just inboard of one wheel. A cardan shaft which can slide ±120 mm and flex 7 degrees is used to transmit power. Both the traction motor and the cardan shaft are parallel to the axle.

The braking system is a combination of dynamic, disc, and tread brakes. Each wheel of the train has a tread brake. In addition, each unpowered axle has four disc brakes.

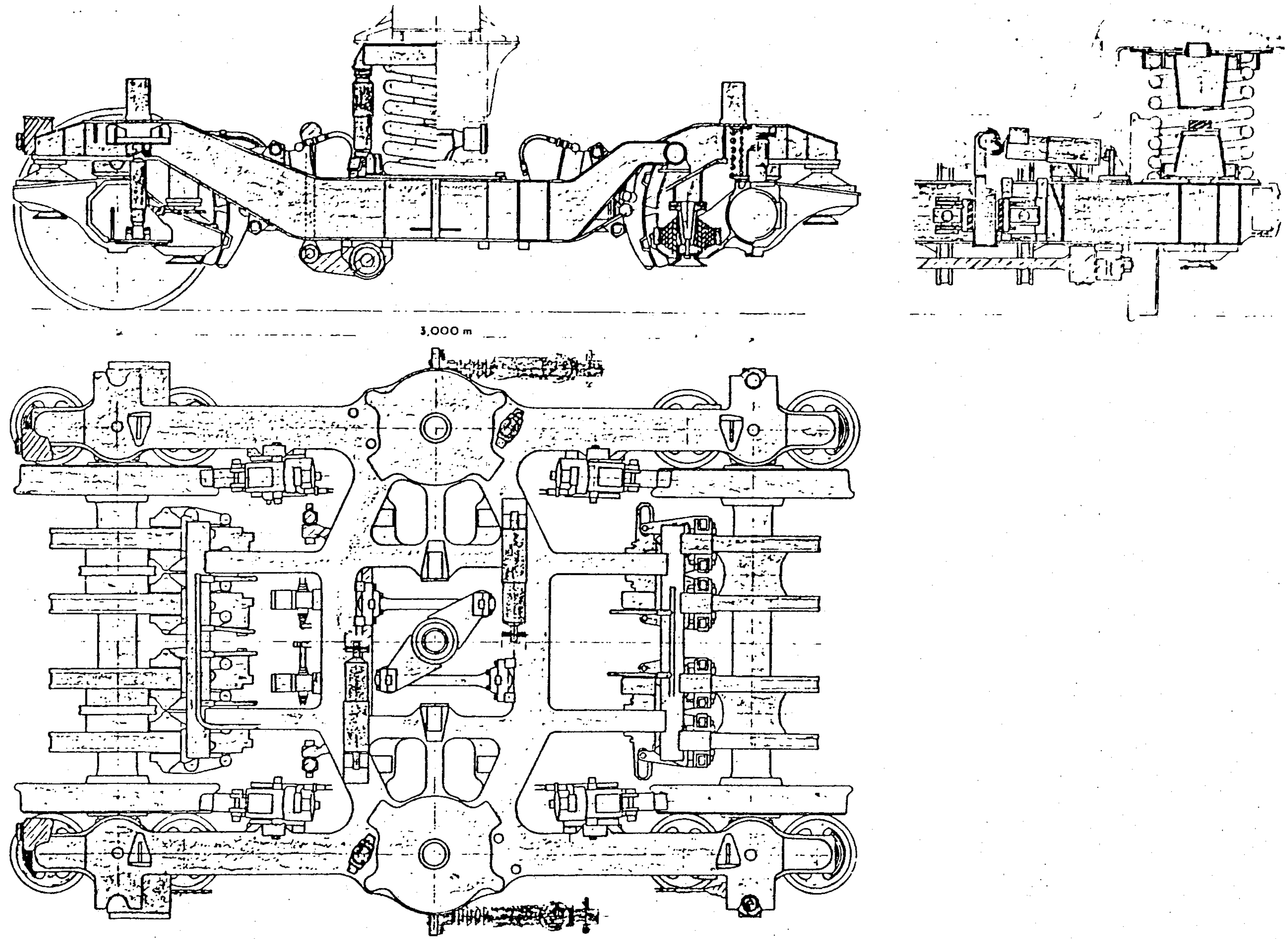


FIGURE 10. FRENCH NATIONAL RAILWAYS TGV-PSE UNPOWERED TRUCK (Y-229)

PREFACE

Bullock,[7] as manager of research and development at Standard Car Truck Company, has presented a very readable discussion of radial truck action, the problems it solves, and the problems it creates. He culminates this discussion with an explanation of the Anchor Truck which his firm is under license to produce in the United States for freight cars. Newland[8] studies the steady-state behavior of a radial truck on curved track and discusses the forces present at the wheel/rail interface. Bullock and Newland are thus the two principal information sources in the Radial Truck section. In particular, the analysis of truck behavior, comprising figures 11 through 17 and accompanying discussion, was taken from Bullock, with some modification and addition. Scheffel's[9] paper is the fundamental background reference for the Anchor Truck. While Scheffel presents extensive mathematical analysis, only Bullock's interpretation of that work is utilized here. General Steel Industries is the American licensee for the Scheffel truck passenger car application.

The remaining trucks are described on the basis of the referenced material which was acquired under IPEEP.

INTRODUCTION AND BACKGROUND

Much effort has gone into the improvement of ride quality of railway vehicles, the aim being to improve maximum speed and comfort on existing track. This work has dealt primarily with vehicles having conventional profiled steel wheels and emphasis has been placed on avoidance of their characteristic dynamic instabilities (hunting). The main problem has been to design a suspension system which is sufficiently soft to attenuate roadbed irregularities, while still maintaining an adequate margin of dynamic stability. Until recently, not as much attention has been paid to the guidance characteristics of profiled wheelsets and their ability to steer through curves.

Much study has been done on curve negotiation of trucks having stiff primary suspension whose axles are forced to remain parallel to each other. In this case, wheel slip occurs with only moderate curvature.

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[7]Robert L. Bullock, "Modified Three-Piece Truck Reduces Hunting and Improves Curving - Status Report," 12th Annual Railroad Conference Proceedings, Effect of Heavy Axle Loads on Track," Pueblo, Colo., Oct. 1975.

[8]D.E. Newland, "Steering a Flexible Railway Truck on Curved Track," submitted for presentation at the IEEE-ASME Joint Railroad Conference, Montreal, 1969.

[9]H. Scheffel, "Wheelset Suspension Designed to Eliminate the Detrimental Effects of Wheel Wear on the Hunting Stability of Railroad Vehicles," submitted for presentation at the ASME Symposium on Railroad Equipment Dynamics, Chicago, 1976; and H. Scheffel, "Self-Steering Wheelsets Will Reduce Wear and Permit Higher Speeds," Railway Gazette International, Dec. 1976.

The motion of trucks with significant yaw flexibility is now receiving attention. The ultimate objective of this attention has been to develop a truck capable of negotiating substantial curves at speed, with no flange contact and a minimum of sliding. Development of such a truck would represent an achievement of improved energy efficiency as well as a vast improvement in wheel and rail life because of the reduction of sliding friction.

The generic name for this truck concept is the radial truck. It is so called because its axles naturally tend toward adopting positions on radii of the curve being negotiated. The conicity of the wheels allows the axles to displace radially outward, such that the rolling radius of each wheel is appropriate to produce minimum sliding at the wheel/rail interface. Ideally, this produces a condition in which each axle is the axis of a cone whose center is at the center of curvature of the curve being negotiated, and whose surface is in perfect rolling contact (no sliding with a surface containing the wheel/rail contact points).

The principal roadblock to the design of such a truck has been that an ideal radial truck is dynamically unstable. Hence there develops a trade off between dynamic stability and good curving ability. Many investigations into hunting stability have been based on creep theory. Briefly, this theory states that when a wheelset is displaced from a position of pure rolling, forces are generated in the contact area between the wheel and the rail. These forces are proportional to the ratio of creep velocity to forward velocity of the wheelset. The creep velocity is an apparent sliding velocity vector at the wheel/rail interface. In pure creep, however, no sliding takes place. This is due to the combination of rolling motion and elastic deformation simultaneously at the wheel/rail interface. The following results are based on these principles.

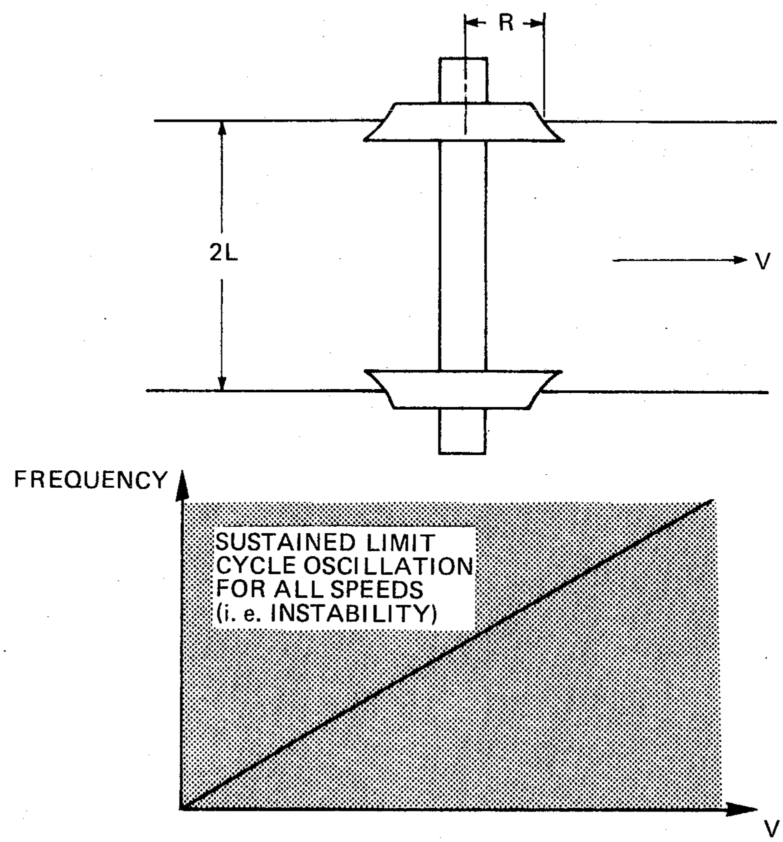
Figure 11 shows a single wheelset rolling along rail. This system is unstable for any speed with its frequency increasing with speed.

Figure 12 shows a single profiled wheelset constrained to ground in the lateral and longitudinal directions. This wheelset is stable up to a critical speed and unstable for all higher speeds. This critical speed is directly proportional to the square root of both the lateral and longitudinal elastic constraints.

Figure 13 shows the profiled wheelset elastically constrained to a mass. This system, like the free wheelset, has no stability whatsoever, regardless of the values chosen for the lateral and longitudinal elastic constants. In other words, the conclusions reached for the single wheelset constrained to ground do not apply to the same wheelset when it is constrained to a mass having lateral freedom.

Figure 14 is a repeat of figure 13 with a second wheelset symmetrically constrained to the same mass. This system is stable below a certain critical speed. In effect, the wheelsets are constrained to each other through the vehicle's frame and obtain their stability in this way.

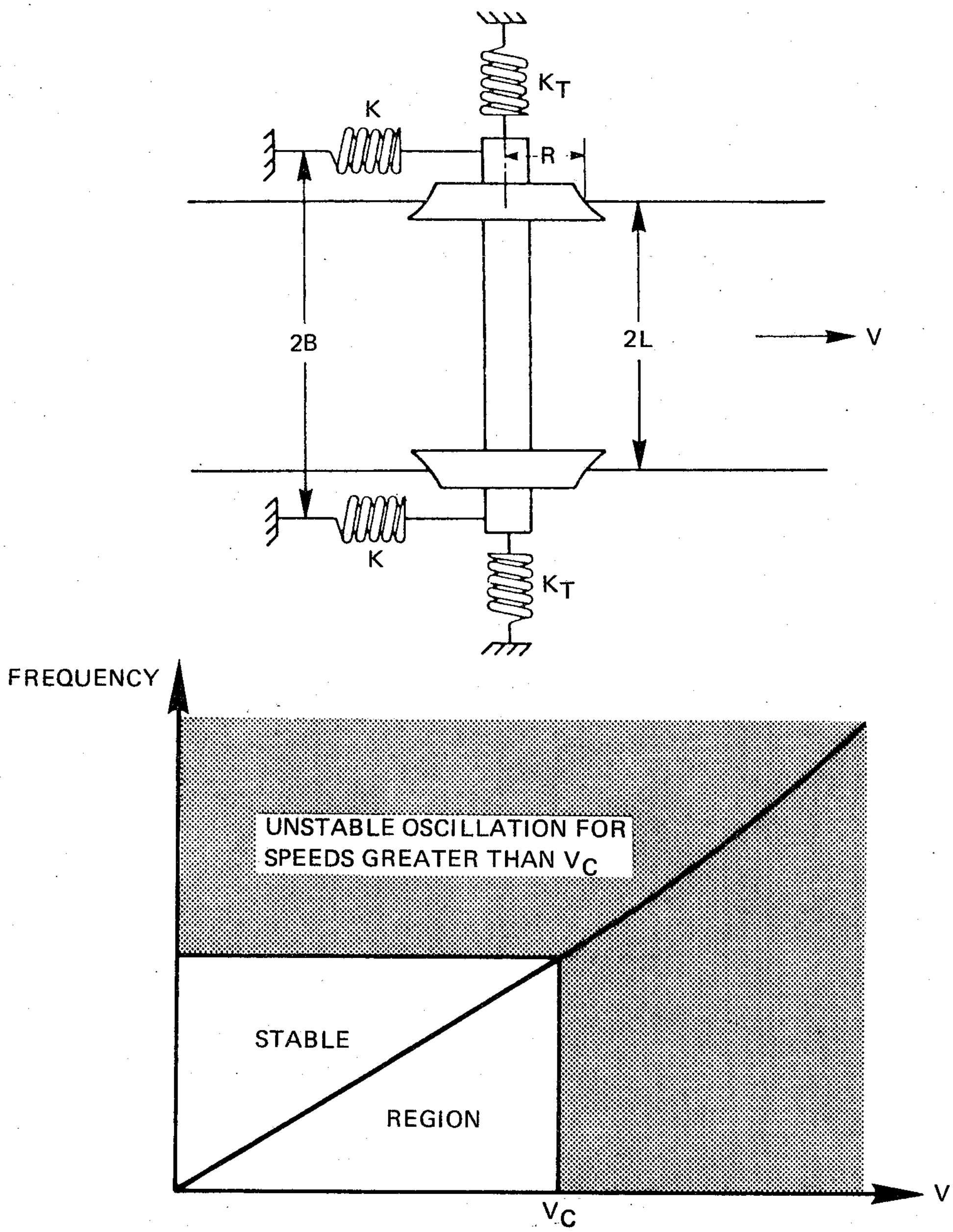
When two wheelsets are constrained to the same mass, their stability is similar to the constraint-to-ground situation in that there is a critical speed and in that stability is improved as yaw constraint is increased. In the former case, however, the improvement is less than in the latter. The ultimate case of



HUNTING FREQUENCY= $V\sqrt{\dfrac{\lambda}{L \times R}}$

WHERE:
V= TRAIN SPEED
λ = CONICITY
R= ROLLING RADIUS

FIGURE 11. SINGLE FREE ROLLING WHEELSET.

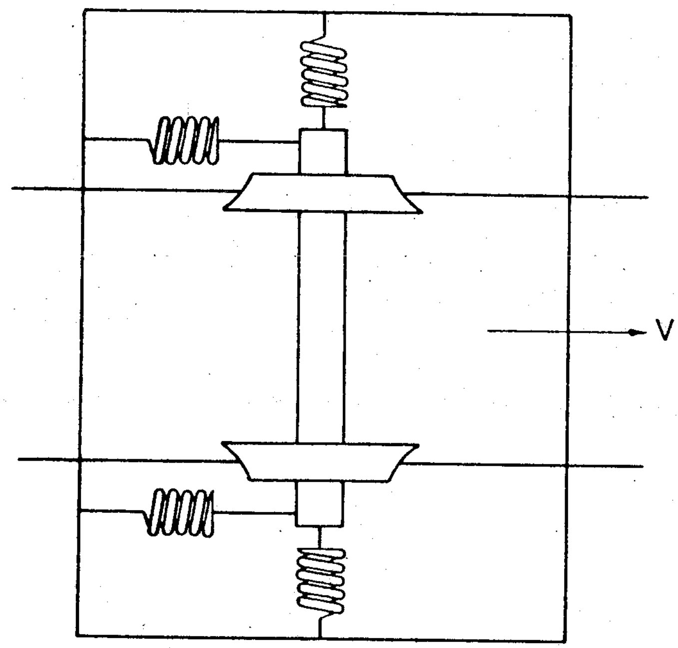


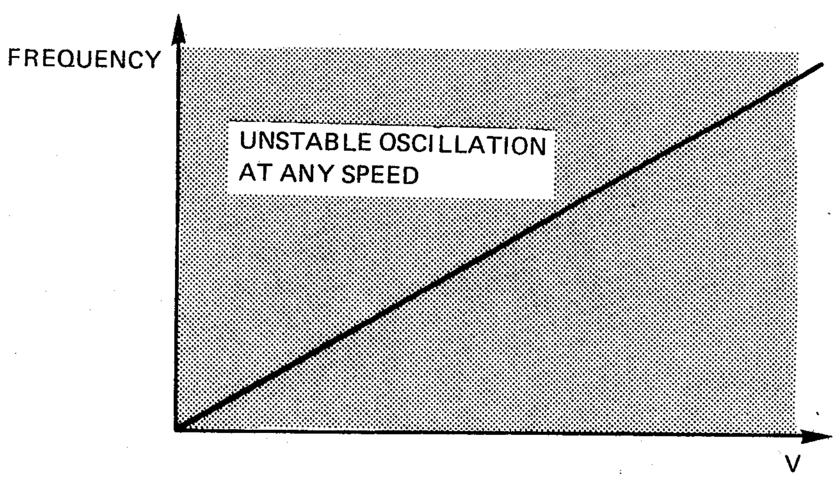
$$V_C = \left[\frac{LR}{\lambda} \left[\frac{K_T}{M} + \frac{KB^2}{ML^2} \right] \right]^{1/2}$$

WHERE:
V_C = CRITICAL VELOCITY
λ = CONICITY
M = MASS OF WHEELSET

THIS IS A HYPOTHETICAL CASE OF RAILS MOVING UNDER CONSTRAINED WHEELSET. HUNTING OCCURS AT SPEED GREATER THAN V_C.

FIGURE 12. SINGLE WHEELSET CONSTRAINED TO GROUND.





V_c = 0 (HUNTING OCCURS AT ALL SPEEDS.)

FIGURE 13. A SINGLE WHEELSET CONSTRAINED BY A FREE MASS.

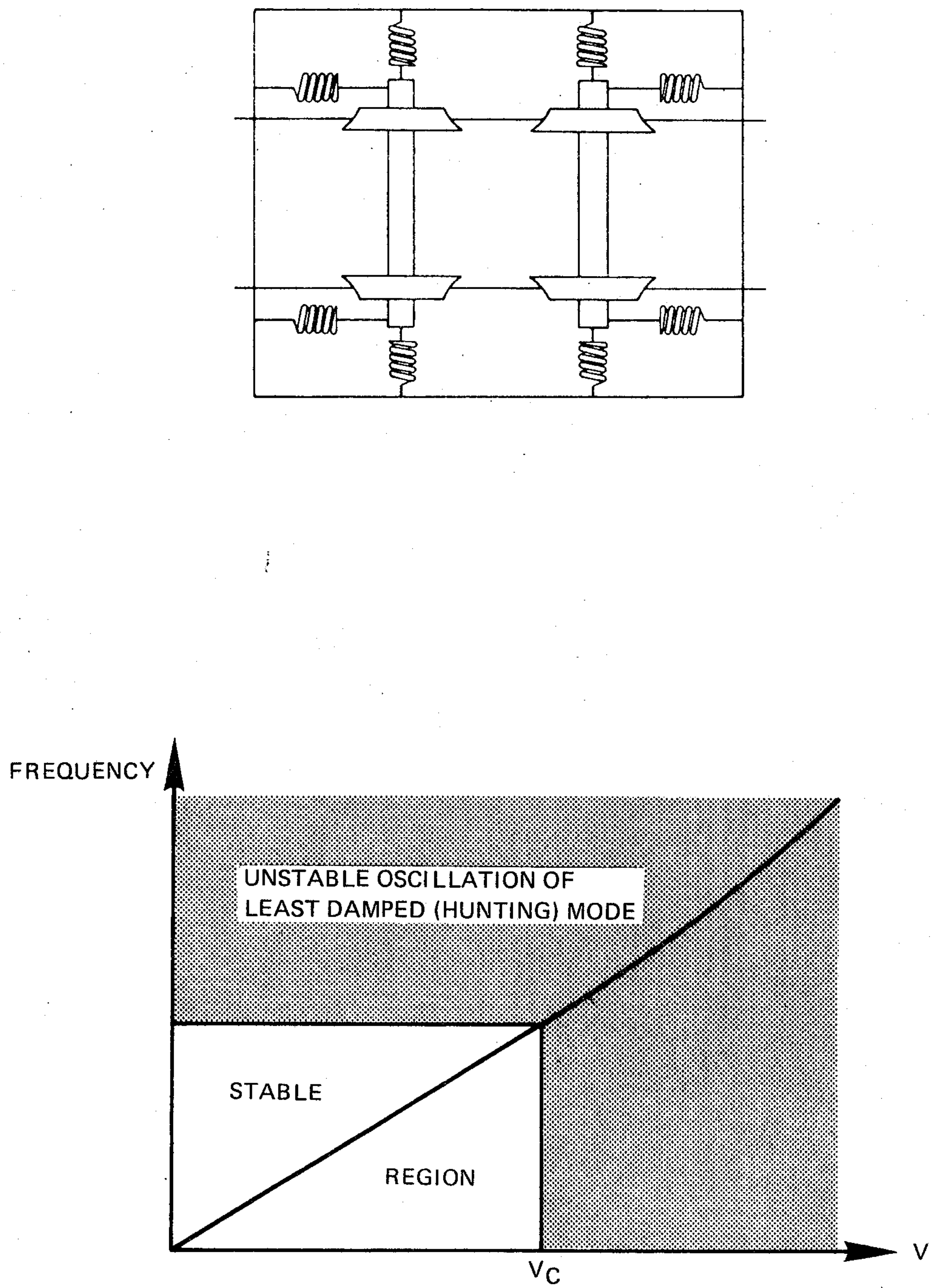


FIGURE 14. A PAIR OF WHEELSETS SUSPENDED TO A MASS GAINS STABILITY.

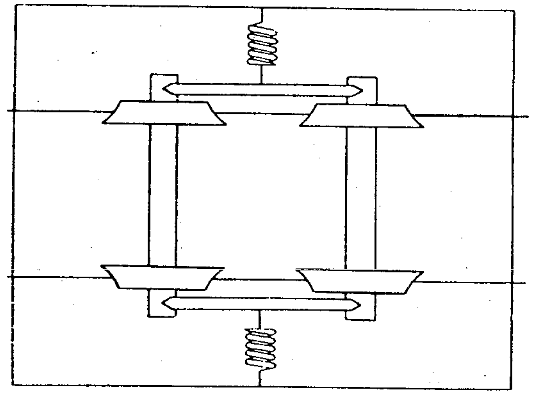


FIGURE 15. CONVENTIONAL TRUCK DESIGNS HAVE HIGH YAW CONSTRAINT BETWEEN WHEELSETS.

yaw restraint on wheelsets constrained to the same mass is shown in figure 15. The principle of the radial truck is completely defeated in the process of achieving high-speed stability. The result is poor curving performance, causing both high tread wear and high flange wear which, in turn, decreases high-speed stability.

ANCHOR TRUCK (SCHEFFEL)

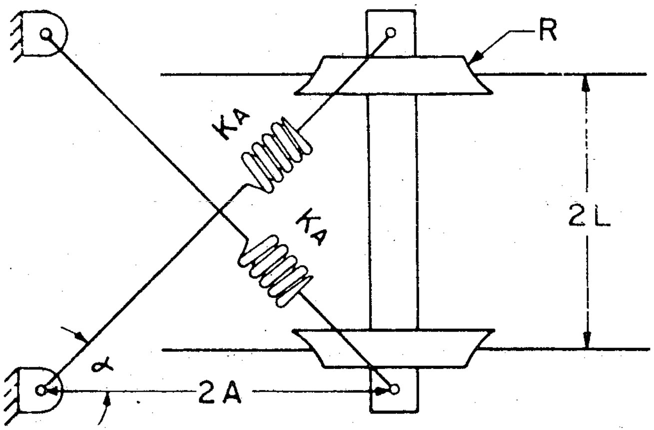
The Anchor Truck represents a very innovative approach to resolving the stability/curving ability dilemma. This truck was invented by H. Scheffel[10] of the South African Railways.

As noted above, a pair of wheelsets suspended to a mass is stable, but less so than a single wheelset suspended to ground. It is also true that one of the pair is more stable than the other. One way of adding additional constraint to the system, which carries with it a virtually inherent critical speed increase, is to add diagonal links.

Figure 16 illustrates that a diagonal suspension could accomplish the same task as the four-spring approach shown in figure 12 for the single wheelset suspended to ground. If the ground connection point is replaced with a second wheelset, and this pair of wheelsets suspended to the truck frame as in figure 14, the desired diagonal connection is provided. Indeed, the stability of the system improves continuously as the diagonal elastic elements are stiffened.

When the diagonal stiffnesses are increased to the point of virtual rigidity, they constitute a holonomic constraint on the original system (i.e., without crosslinks). In this case the wheelsets are constrained to move in a prescribed manner relative to each other. This system, shown in figure 17, is the basic representation of the Anchor Truck. The addition of a holonomic constraint to any dynamic system reduces the number of degrees of freedom of that system by one. The number of resonant frequencies is also thereby reduced by one, and the new frequencies tend to fall between the old ones in value; hence, there is an increase in critical speed for hunting. Because of the direct connection between the wheelsets, the stability of one wheelset must be the same as for the other.

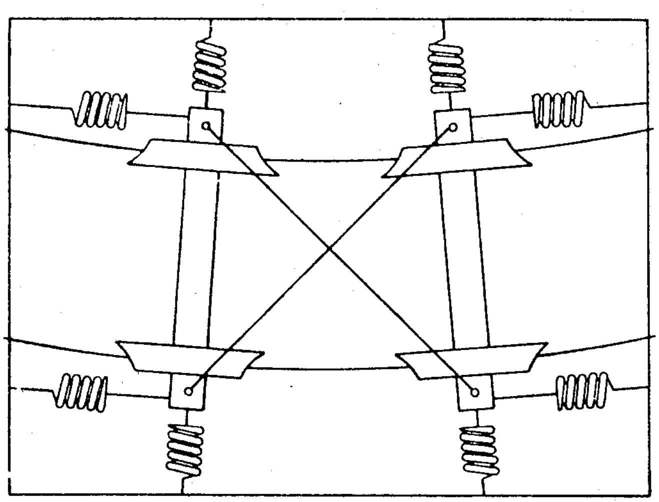
[10]Scheffel, ibid.



WHERE:
A= ½ EQUIVALENT WHEELBASE
M= MASS
L= ½ TREAD
R= ROLLING RADIUS
λ= CONICITY

$$V_c = \left[\frac{LR}{\lambda} \times \frac{K_A}{M} \cos^2\alpha\left(1 + \frac{A^2}{L^2}\right) \right]^{\frac{1}{2}}$$

FIGURE 16. CRITICAL SPEED OF A SINGLE WHEELSET DIAGONALLY SUSPENDED TO GROUND.



LOW WHEELSET YAW CONSTRAINT
ALLOWS WHEELSETS TO ASSUME
A RADIAL POSITION IN CURVES

FIGURE 17. DIAGONAL CONSTRAINT PROVIDES IMPROVED WHEELSET STABILITY.

The advantage of this approach is that the diagonal links are able to restrict the independent oscillation of the two wheelsets in the same manner as the rigid parallel connection of figure 15 and yet freely allow the wheelsets to adopt a radial orientation. In practice it has been found that when the elastic yaw constraints on each wheelset have approximately the same value as the gravitational stiffness, they are low enough for the axles to assume an approximate radial position in curves. The wheels can then have pure rolling in the curves with guidance supplied by the creep forces.

The Anchor Truck has been service tested on the South African Railways and has been implemented in prototype form in the United States by the Standard Car Truck Company.

In South Africa the truck was tested in unit ore train service on 8,000-ton trains operating over sharp curves and steep downgrades from the northern part of the country down to Port Elizabeth. On normal three-piece trucks the wheelsets had usually been condemned after 48,000 km of this service. An experimental Anchor Truck was inspected after 99,000 km with no appreciable change found on its wheel profile. Wheel conicity has been approximately 0.22 (compared to 0.05 on typical United States trucks) with no adverse effect on hunting stability. The low yaw constraints are accomplished by the use of rubber sandwiches which offer the advantages of some vertical flexibility and predictable damping characteristics. As a basic truck concept, the three-piece cast frame was adopted. This concept offers the advantages of good wheel load equalization, low weight, and low manufacturing costs.

The Anchor Truck has been adapted to standard gage for possible production in the United States, and a prototype has been fabricated by Standard Car Truck Company of Chicago for testing on the Union Pacific Railroad. The South African computer model was used for this design effort. During the modeling process it was observed that a standard truck with a 0.22 wheel conicity and a critical speed of 95 km/h had its critical speed raised to 249 km/h by addition of the cross anchors. The modeling also determined that the cyclic loading in the cross anchors was approximately 4,450 daN. This was for a 91-metric ton hopper car. Thus, the cross anchors can be made of relatively light material. The high conicity, of course, would not be used without the cross anchors; but then the increased critical speed would be at the cost of flanging on curves. It was also found during the modeling process that body hunting stability was actually improved with increasing conicity.

Another benefit of the high conicity is a reduction in hertzian contact stress. The standard 0.05 conicity on curved rail causes heavy initial wear. The flange also wears due to two-point contact on curves. This continues until the normal worn wheel profile and its attendant instability are developed. Thus, by virtually eliminating two-point contact and vastly reducing tread wear, the worn wheel profile ceases to be a significant design consideration for suspension systems.

The adaptation of the Anchor Truck principles to the 91-metric ton hopper was accomplished by positioning the anchors below the bolster. This allowed use of the existing bolster, brakes, and much of the side frame pattern equipment. The yaw flexibility was created by placing shear pads above each journal bearing. The loaded test car had a gravitational stiffness of 963 daN/cm using 0.15 conicity wheels. The optimum shear pad design had a spring rate of 1,121 daN/cm. Figure 18 shows projected curving performance as a result of tests. Testing was also conducted on mainline tangent track with speed ranging from 8 to approximately 153 km/h in 8 km/h increments. The critical truck hunting speed was never reached.

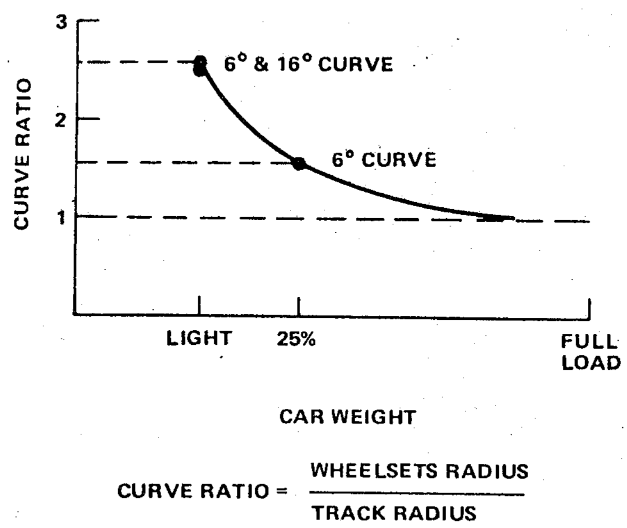


FIGURE 18. ANCHOR CURVING CHART.

LIST TRUCKS (RAILWAY ENGINEERING ASSOCIATES)[11]

Railway Engineering Associates and Canadian National Railway have, in consortium, been developing the radial truck concept both analytically and experimentally. They have been working with three basic proposals.

The first is to simply modify an existing three-piece truck by the addition of a self-steering assembly. The net results of such effort would appear similar to those achieved with the Anchor Truck, since the cross anchors provide precisely this service in the form of a holonomic constraint as discussed above.

The second approach involves redesigning the castings of the conventional three-piece truck, providing space for more steering motion of the steering

[11]Harold A List, "Design System Approach to Problem Solving," 12th Annual Railroad Conference Proceedings, "Effect of Heavy Axle Loads on Track," Pueblo, Colo., Oct. 1975; and Harold A. List, W. N. Caldwell, and P. Marcotte, "Proposed Solutions to the Freight Car Truck Problems of Flange Wear and Truck Hunting," submitted for presentation at the ASME Winter Annual Meeting, 1975, Houston.

assembly and also allowing the steering assemblies to be made large enough to support the brake beams. The latter change is to allow more precise location of the brake shoe across the face of the wheel tread, thus eliminating flange/brake shoe wear and giving improved brake control.

The third approach is a completely new truck design. This would involve a four-piece truck in which the weight of the car is applied directly to the side frames.

List also proposed that trucks with a steering feature be developed for passenger cars, transit cars, and locomotives. For passenger cars, the potential gains are essentially the same as for freight cars, with the addition of an improved safety margin because of reduced flange contact and better high-speed stability. Locomotives would additionally gain in adhesion because of reduction of sliding at the wheel/rail interface. Along with all of the above, a very important factor on transit cars would be the reduction of wheel noise while negotiating very sharp curves.

At this writing a truck of the first type (figure 19) is being tested on the Canadian National Railway.[12] Tests with developmental units of the new truck, operating under load on curves, are said to have shown up to 60 percent reduction of flange force and as much as 75 percent reduction in the angle of attack between wheel flanges and rail. These results, it is held, should mean a substantial reduction in track and wheel maintenance, especially on routes where there is considerable curvature. Operating unloaded, the steering-type trucks were said to demonstrate stability at speeds up to 129 km/h during testing.

The benefits of the steering-type truck can be applied to many existing roller-bearing freight cars. Rubber pads (figure 20) are placed between the bearing adapters and the side frames to facilitate steering motion of the axles. Steering arms are added to handle the exchange of steering forces between the axles. Together, the rubber pads and steering arms permit radial movement of the axles in negotiating curves and increased stability on tangent track.

The Canadian National Railway plans to put the first production models of the steering truck in service on its unit coal trains between the Luscar mines near Jasper, Alta., and Port Mann on the Pacific Coast, a 550-mile route with a high proportion of curves. Annual savings in wheel and rail maintenance are estimated at $950 for each of the cars in service on the route. Reduction in friction drag on curves is expected to add another $225 in annual fuel savings for each of the cars.

The cost of fitting a conventional car with steering arms is said to be about $1,750 per car, plus labor. Dominion Foundry & Steel Co., Hamilton, Ont., is producing the steering arms for Canadian National, while Dresser Transportation Equipment Corp., Buffalo, N.Y., is producing the equipment for the U.S. market.

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[12]"At Last! A Freight Car Truck to Reduce Rail Wear," Railway Track and Structures, Feb. 1978.

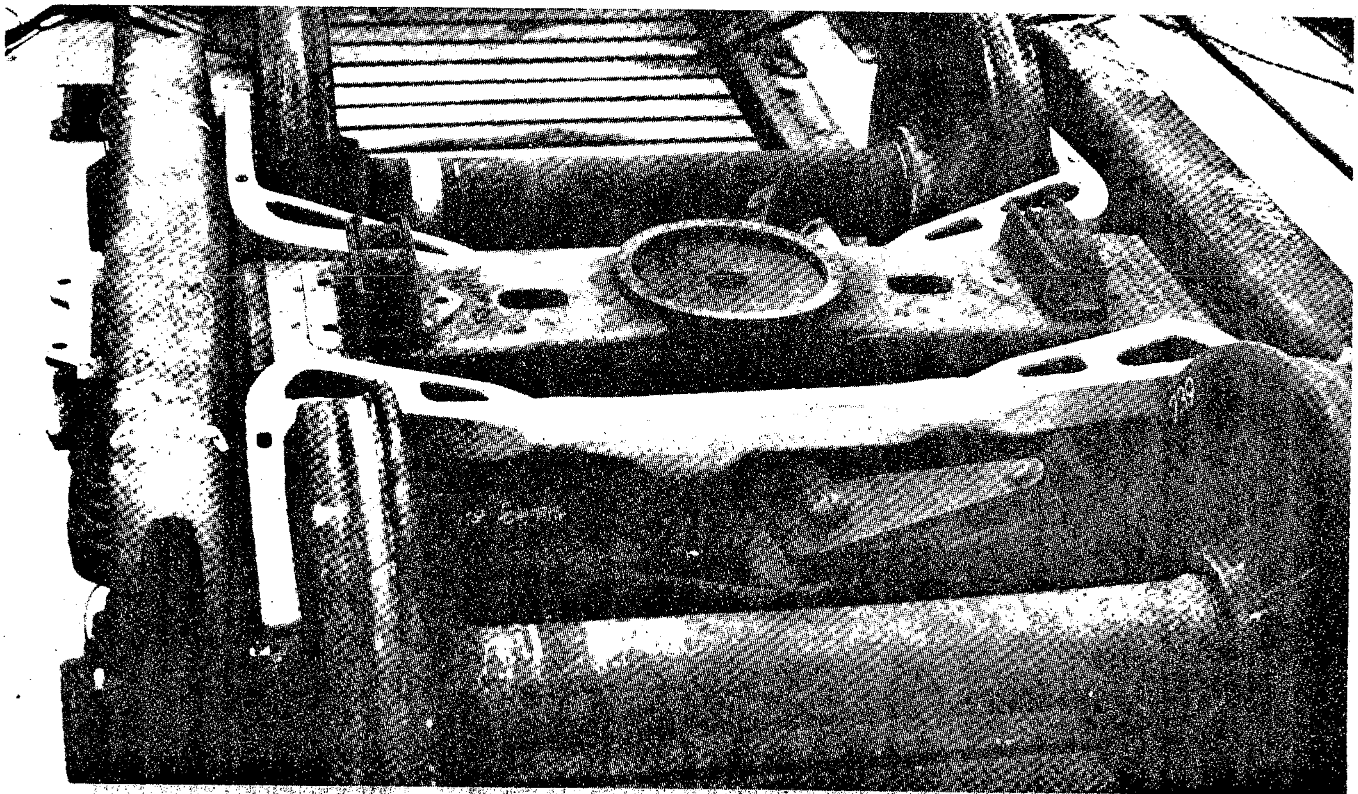

FIGURE 19. LIST TYPE I TRUCK.

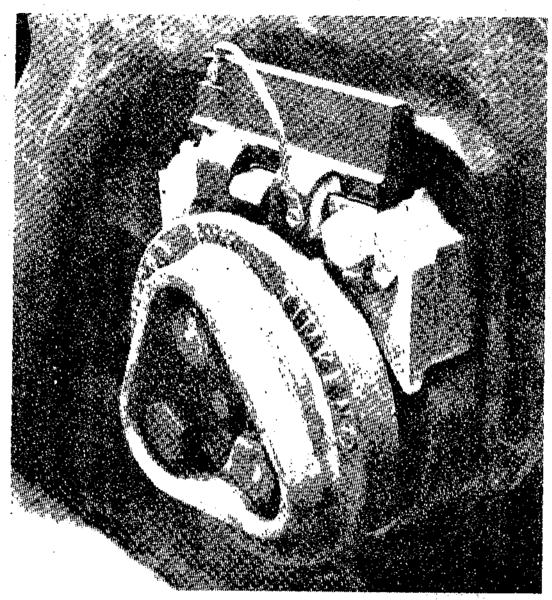

FIGURE 20. RUBBER PAD INSTALLATION.

SCALES TRUCK[13]

The Scales Truck, under development by J. P. Devine Manufacturing Company, is shown in figure 21. Its principle of operation is the forcible steering of the wheelsets into a radial orientation via a linkage driven by the relative rotation of truck and carbody during curving. The Anchor Truck, by comparison, allows the wheelsets to naturally seek a radial orientation, but under mutual constraint to do so simultaneously.

The truck is an H-shaped fabrication with a two-stage suspension system. The primary suspension is formed by rubber shear compression units above the axle bearings. The chevron-type units are located in packets at the ends of the H-frame. The secondary suspension consists of a pair of air bags and hydraulic dampers with rubber overload stops. The air bags are carried on a bolster, which is tied to the carbody by radius arms, and slides on the truck side frame on low-friction pads. A vertical center plate restricts the bags, with control by rubber spring units.

The primary suspension is restrained laterally by the chevron rubber springs and controlled longitudinally by swing arms. The pivot points for the swing arms are connected to outwardly pointing steering arms, which are in turn pivoted on the truck frame. The outer ends of the steering arms are connected by links, fitted with ball joints, to extensions of the bolster which also support the lower ends of the hydraulic dampers. On curves, the steering linkage moves the wheels on the outside of the curve further apart and brings the wheels on the inside closer together.

The primary suspension units shear longitudinally as the wheels move. The truck swiveling and consequent longitudinal wheel movements are in direct proportion to the degree of curvature. Therefore, the wheelsets are held at right angles to both curved and straight track. The arrangement of the radial truck on a curve is shown in figure 22.

DIFFERENTIAL TRUCK (RUSSIA)[14]

Russian engineers are testing a truck with differentials on the axles. This arrangement is such that the left and right wheels can rotate independently of one another. The trucks also have pneumatic-tired wheels fitted with metal flange discs for rail contact.

Such a truck design can only approach one part of the problem addressed by radial trucks: namely, the difference in speed between the outside and inside wheels turning. A true radial truck not only allows the wheels to position themselves to solve this problem naturally, but also provides the transverse forces necessary to turn the vehicle through use of the wheel treads rather than through flange contact.

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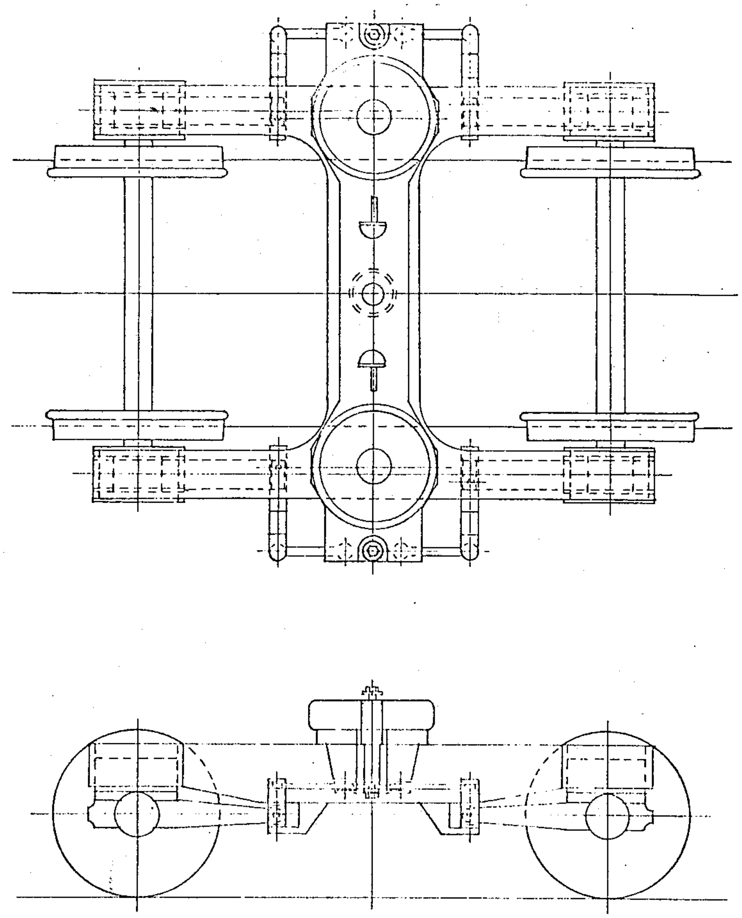
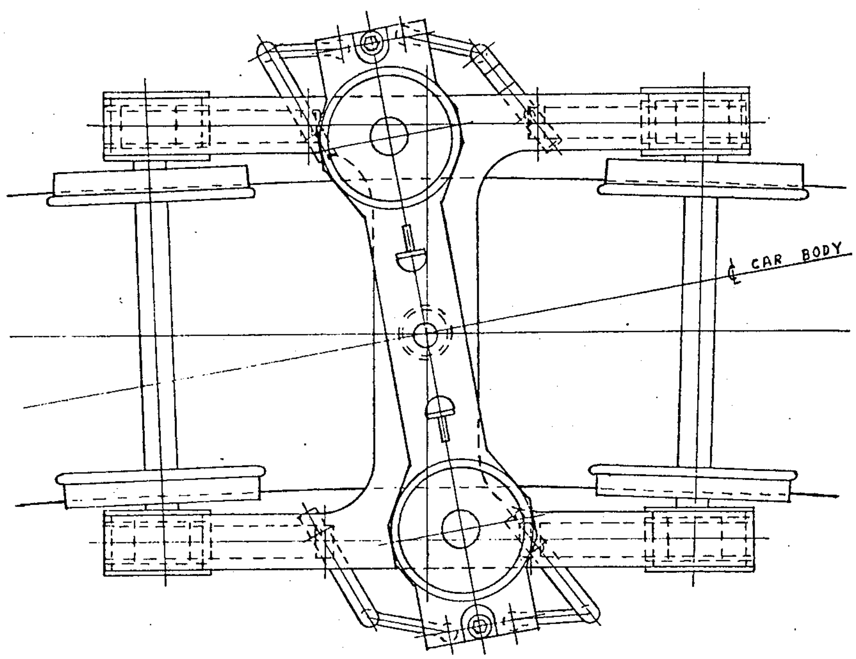
[13]"Radial Trucks for Passenger Cars," J. P. Devine Mfg. Co., Pittsburgh.

[14]"Soviet Tests on Differentials in New Bogies," International Railway Journal, Sept. 1977.



SCALES TRUCK — J. P. DEVINE MFG. CO.

FIGURE 21. RADIAL TRUCK FOR PASSENGER CARS.



SCALES TRUCK — J. P. DEVINE MFG. CO.
DESIGN COVERED BY U. S. PAT. 3,862,606

FIGURE 22. RADIAL TRUCK ON CURVE.

SUMMARY

An overall assessment of the truck technology presented here is made on the basis of the parameters discussed under "Functions of Rail Vehicle Suspension." These are vehicle guidance, vibration isolation, vehicle support, and traction and braking.

Vehicle guidance, with its inherent dynamic stability problems, has been the area in most critical need of improvement. For this reason the recent progress with the development of radial trucks is the most significant technology reported here. It has involved real, yet simple innovation in design which captures the chance to exercise inherent control over a previously poorly controlled aspect of truck behavior.

All of the radial truck concepts presented are apparently not at equivalent levels of development. All do, however, show significant promise toward achieving the goal of good curving performance and reduction of wheel tread and flange wear.

The most promising of these concepts is the Anchor Truck. While the Type I List Truck has the same functional principle and thus should give equivalent performance, the cross anchors of the Anchor Truck are a simpler and lighter steering means than List's steering arms.

but are capable of retro-fit!

2. CARBODY CONSTRUCTION

INTRODUCTION

The status and recent history of railway passenger rolling stock construction are typified by those of the French National Railways (SNCF). The SNCF is currently building a new high-speed passenger railroad from Paris to Lyon with equipment and way tailored to each other. The design and construction of this rolling stock thus embody the best of the materials, design, and manufacturing technology developed in recent years.

Jousserandot[15] has provided a good overview of this history, and as a result, the material that follows (through French Passenger Car Development) has been adapted from his article. Some material on structural analysis (finite element method) has been added. The discussion of the TGV-001 has been adapted in a like manner from Bernard,[16] although a discussion of the Association of American Railroads (AAR) structural requirements has been added. The works by Jousserandot and Bernard were acquired as a result of an IPEEP visit to the SNCF.

As with any industrial product, most progress is integrated gradually into rolling stock as it develops. During the process it is adapted to criteria such as environmental constraints, passenger tastes, and governmental regulations.

From the standpoint of the equipment itself, probably the most stringent environmental constraint is the railroad operating environment per se. With the exception of dedicated trainsets such as the TGV (for the Paris-Lyon line), most equipment must be relatively self-contained; that is, it cannot be assumed that the other equipment in the same train will be of the same design.

Therefore, passenger equipment is not suited to sudden fundamental modifications that would create incompatibilities between vehicles designed at different times and maybe for different original purposes. Interpreted in terms of carbody construction, this means that the basic structural design must be relative to an entire train of variable consist. The loads imposed on a single car through the wheel rail interface and the connection to a specific locomotive are not the same as those loads encountered in a long train of arbitrary consist, a long train of homogeneous consist, or similar short trains.

STRENGTH REQUIREMENTS

In Europe, rolling stock designed for international use must meet the standardized requirements of the International Railway Union (UIC). In the

[15]Pierre Jousserandot, "Mainline Passenger Rolling Stock Developments Since 1965 and Future Prospects," French Railway Techniques, No. 4, 1976.

[16]J. Ph. Bernard, "The Experimental Turbine Train T.G.V.-001," French Railway Techniques, No. 1, 1973.

United States these standards are set by the AAR and the Federal Railroad Administration (FRA).

The UIC requirements, generally less severe than those of the AAR/FRA, state that a vehicle must be able to withstand the following loads without permanent deformation or excessive stresses:

a. A static compressive force of:

200×10^3 daN at the buffing level;
50×10^3 daN diagonally at the buffing level;
40×10^3 daN at 350 mm above the buffing plane;
30×10^3 daN at waist rail and cantrail levels.

(Apart from the cases where it is applied to the buffers, the force is distributed as evenly as possible between all the components contributing to the strength of the coach end wall at the level concerned.)

b. A localized static load (automatic coupling):

200×10^3 daN in compression on the draft gear stops;
150×10^3 daN in tension on the draft gear stops.

c. A uniformly distributed vertical load of $P=K(P_1+P_2)$ where:

P_1=mass of vehicle body in service order;
P_2=twice the number of seats x 80 kg;
K=1.3 -- the dynamic load coefficient.

FRENCH PASSENGER CAR DEVELOPMENT

AS OF 1965

Table 1 shows the characteristics of the main series of passenger cars in service or about to enter service in 1965. The overall car lengths vary between the 24.5 meters for the UIC type Y coach and 25.5 meters for the Trans Europe Express (TEE) coaches. The carbodies are of all-steel welded construction and are self-supporting. There are two categories, differentiated by the type of steel used:

a. Bodies of Martin steel sheet together with, in certain cases, corrosion-resistant steel with a high elastic limit (Corten, for example) for the underframes and body frame members.

b. Bodies of stainless steel, 18-percent nickel and 8-percent chromium, with Martin steel retained for the underframe ends including the pivot bolster. This technique saves some weight and also offers the advantage of good corrosion protection without paint.

1965 TO 1975

Table 2 gives general information on equipment introduced during this period. Most notable was the introduction of 26.4-meter long-bodied passenger cars.

TABLE 1. MAIN TYPES OF FRENCH COACHES IN SERVICE IN 1965.

Types of coaches	Types of trucks	Main Characteristics				Energy supply	Heating or air-conditioning	Main constructional features
		Length (m)	Max speed (km/hr)	Tare (t.)	Seating accom			
Internal Services USI 1960	Y24C	25.9	140	A4t4 36 B10t 34 5	50	HV dual system train line LV axle-driven generator and 72 V battery	Heating by forced air at floor level Summer ventilation Automatic electronic regulation	Martin steel underframe and body U I C folding pivoting access doors Half opening windows Automatic adjustable brake, 1 pipe and cast iron brake blocks
U I C Type Y 1960	Y24A1	25.5	140	A9 41 B10 42 B9c9 43	54 80 54/72	HV four-system train line LV axle-driven generator and 72V battery	Heating by forced air at floor level Summer ventilation Automatic electonic regulation	Underframe and body members in Corten steel Side walls and roof in Martin steel Half-opening windows U I C folding pivoting access doors Automatic adjustable brake 1 pipe and cast iron brake blocks
DEV stainless steel 1963	Y24Z	25.09	150	A9 37 5 A62t 39 A70 37	54 36 41	HV dual-or four-system train line LV axle-driven generator and 72 V battery	Heating by forced air at floor level Summer ventilation Automatic electronic regulation	18/8 stainless steel body with ends of underframe in Martin steel D E V pivoting access doors with moving floor section Half-opening windows Automatic high-power brake, 1 pipe and cast iron brake blocks
TEE PBA	Y24A1S	25.5	150	A8 47 A8D 47 A2D 49 5	48 46	Generator van diesel engine and 300kVA 660 V three-phase alternator 24 V batteries	Air-conditioning with diffusion of treated air from conditioning unit above vestibule supplementary heating by electric radiators at floor level	18/8 stainless steel body with ends of underframe in Martin steel Automatically controlled plug-slide access doors Fixed double-glazed windows with venetian blind between panes U I C loudspeaker installation Automatic high-power brake, 1 pipe and cast iron brake blocks
VRU Restaurant car 1962	Y26C	24.5	150	48 5	52	70 kVA 220 380 V three-phase diesel alternator set	Air-conditioning with diffusion of treated air through ceiling from conditioning unit above vestibule supplementary heating by hot water radiators (diesel cooling circuit in skirtings)	Body in Martin steel with stainless steel at kitchen and service area U I C folding-pivoting access doors Fixed double-glazed windows with venetian blind between panes U I C loudspeaker installation Automatic high-power brake, 1 pipe and cast iron brake blocks Electrically operated kitchen equipment and refrigerators

TABLE 2. NEW TYPES OF FRENCH COACHES BUILT BETWEEN 1965 and 1975.

Types of coaches	Types of trucks	Main Characteristics				Energy supply	Heating or air-conditioning	Main constructional features
		Length (m)	Max Speed (km/hr)	Tare (t.)	Seating accom			
New Mistral 1969	Y28 Y26P with air suspension for rest car	25.5	160	A8u 44 5 A8tu 50	48 46	Generator van 580 kVA 660 V three-phase diesel alternator set	Type TEE PBA air-conditioning (table 1)	TEE PBA fittings with development and improvement of kitchen and bar equipment Electro-pneumatically controlled brake
High comfort standard	Y28	25.5	200	A8u 51 A8tu 50	48	Generator van, 580 kVA 660 V three-phase diesel alternator set	Air-conditioning with diffusion operated air through ceiling from conditioning unit above vestibule supplementary heating by electric radiators at floor level	Underframe and main body members in Corten steel Body to trapezoidal gage capable of 5 " tilt Automatically controlled plug slide access doors Fixed double glazed windows with venetian blinds between panes U I C telecommunications Electro-pneumatic and electro-magnetic brakes
Internal services long body TU 75	Y32	26.4	160 Poss 200	A10tu 40 B10tu 40 5 B52tu 42	58 83 (80) 44 (40)	Static convertor 30 kVA dual or four system 40 kVA dual or four system 24 V battery	In first and second class air-conditioning with air diffusion below windows by ejector-convectors	Corten steel underframe and body Folding-pivoting access doors with automatic closure Internal doors power assisted Electro-pneumatic brake U I C telecommunications
U I C long body Type ZVU 75	Y32	26.4	160 Poss 200	A9u 39 A4B6u 39 5 B10c 10xu 42 6 B6D 36 3	54 60 60 36	30 kVA four system static convertor and 24V battery Axle-driven generator and 24 V battery	Air-conditioning from unit below underframe, air diffusion below windows by ejector-convectors	Corten steel underframe and body Folding-pivoting access doors with automatic closure and four access steps Power-assisted intercommunication doors Electro-pneumatic brake U I C telecommunications
Grill-Express	Y28	24.5	160	53 5	20 to 40	75 kVA diesel alternator set	Mixed heating forced air from roof with electric and hot water heater hot water radiators at floor level	Corten steel underframe and body members 3 folding-pivoting access doors 10 tables for 1 10 tables for 2 or 4 persons Electro-pneumatic brake U I C telecommunications
Sleeping cars T2	Minden Deutz M62B	26.4	160	60 5	36	HV train line and four-system equipment LV axle-driven generator and 110 V battery	Air-conditioning with diffusion of air under windows by ejector-convectors the heater unit is supplied with hot water from two boilers, one electric and one liquid-fuel fired (for autonomy)	Underframe and body members in copper bearing steel sheet Arranged as 18 two-bed compartments (9 upper, 9 lower) 2 folding-pivoting access doors with 3 or 4 access steps Half-opening lockable windows Electro-pneumatic brake Service area with cold cupboard and water heater

While continuing to follow the UIC rules, vehicle structural designs and production methods were aimed at several objectives:

a. Weight reduction. This produces both original and operational cost savings. The latter are particularly relevant to high-speed trains and trains which stop frequently.

b. Corrosion protection.

c. Improvement of thermal and sound insulation.

Some worthwhile weight reduction had already been achieved with 1965-vintage cars by designing closer to the UIC strength standards and by introducing high-strength steel for certain components. This has become firm practice along with the use of copper-bearing corrosion-resistant steel for the protection of the underframe as well as the main members. Because the resulting corrosion loss is small, it is then possible to reduce the original wall thickness which contained a corrosion allowance. Because it is necessary to maintain sidewall smoothness, however, the material for these components is often still the traditional carbon steel.

Parallel with this progress has been the upgrading of design-analysis methods made possible by the increase of general-purpose computer power available. Better structural analysis makes possible significant weight savings through better utilization of material. In earlier years SNCF designers regarded the carbody as two Vierendeel trusses made up of the roof as the upper chord and the underframe as the lower chord with the pillars forming the vertical members. This structure is a statically indeterminate system of adjacent frames and was analyzed by a number of approximate methods. These methods have since been implemented on computers and improved to the point of yielding quite good results.

The premier structural analysis technique now in use, however, is the finite element method. It is apparently in use by the SNCF. The finite element method models any structure by establishing "nodal points" throughout the structure and then determining the displacements and/or hypothetical concentrated forces at those points. In the case of a three-dimensional continuum, the nodal points become the vertices of compatible three-dimensional elements (tetrahedrons, parallelapipeds, etc.). When displacement relationships between the vertices of a given element are assumed (i.e., linear, quadratic, etc.), strains within the element can be computed based on the displacements at the vertices. Given a stress-strain relationship for the material (which can be as complex as desired, such as for composite materials) the strain energy within the element can be computed based on the vertex displacements. By mathematically recombining the elements (forcing coincident vertices to have the same displacement) and minimizing the total work done by the internal stresses (strain energy) and the external forces, the vertex displacements can be solved for simultaneously. It can be seen that this is basically the classic Rayleigh-Ritz method in modern dress. A large body of computer software for the finite element method now exists, the most prominently known program being NASTRAN which was developed during the space program.

As indication of the progress achieved by the SNCF in weight reduction, table 3 below shows some underframe body-mass values for several lengths of carbody designed in different years.

TABLE 3. UNDERFRAME BODY MASSES.

Type of stock	Year built	Overall length (m)	Mass of underframe body assembly (t)
USI (Int. Ser).	1960	25.09	10.5
UIC type Y	1960	24.5	12
VTU 75	1975	26.4	10.3
VU 75	1975	26.4	9.3

Stainless-steel construction has yielded an appreciable saving (about 20 percent of the mass of the body), but this is limited by the fact that the ends of the underframe, which are large, heavy assemblies, are still made of traditional steel. The weight reduction achieved with traditional steel by way of improved design and manufacturing methods combined with the prospects offered by light-alloy construction techniques is weakening the competitive position of stainless steel. Ultimately, however, weight reduction is limited not by strength requirements but rather by body stiffness such that the lowest natural frequency will be higher than any expected service-generated vibrational frequency.

Protection against corrosion is of fundamental importance because it enables the margin on the thickness of members to be reduced with corresponding weight reduction. At the same time it avoids future maintenance on the body framework. Progress in welding techniques has reduced the need for lap joints, which are a source of corrosion if they are not perfectly watertight, and has enabled connections to be made by edge welds with the bead deposited by arc welding.

TGV (FRANCE)

INTRODUCTION

The TGV is the SNCF articulated trainset being developed for the new Paris to Lyon high-speed line. This new railway, to be completed in 1981, will be reserved exclusively for passenger trains which will cover the 440 km distance in 2 hours with a speed of 260 km/h. Top speed of the line will be set at 350 km/h.

The new line will be electrified. However, since the trainset development is taking place at the same time the line is being built, the prototype trainset, the TGV 001, is gas-turbine-electric powered. Strictly speaking, the discussion that follows covers this prototype unit only. The electrified version, the TGV PSE, from the standpoint of criteria such as carbody construction and running gear, will be virtually the same trainset. The power cars will simply have electric propulsion equipment in place of the gas-turbine generator set and only six trucks are powered.

POWER CARS

The two power cars are identical, one on each end of the trainset. The power cars are different from conventional locomotives in that they do not provide all of the tractive effort of the train. Since all axles of the train are powered, the purpose of the power cars is to function as a power supply to these axles. Because of their location they also contain the operator's station.

The power carbody is of the stressed-skin type and is made of high-elastic-limit, corrosion-resistant steel. It is designed to support an axial compressive load of 200 metric tons. This does not meet the AAR interchange requirements for passenger railcar structures.

The frame (figure 23) is made of welded components. Its two sides are composed of vertical and longitudinal members connected at the bottom by crossmembers and at the top by roof crossmembers between the cantrails or sills. The external aluminum sheet cladding is stretched and fixed by screws.

The cab structure is designed to rest on the shock-absorbing elements of the body and underframe. Diagonal cross stays are provided at cantrail or sill level. The cab is thus a rigid unit and is completely integrated with the vehicle body structure.

The streamlined nose contains a strong framework supported on the sole bars of the underframe. This framework constitutes both the headstock and a protective shield. In front of this shield is a removable assembly (the streamlined nose of the train) which is designed to act as a collapsible energy absorber in the event of a collision. The protective shield itself is designed to withstand a force of at least 70 metric tons uniformly distributed at the level of the upper waist rail.

TRAILERS

The TGV 001 has three different types of trailing vehicles: second-class car, first-class car, and test car (figures 24, 25, and 26). The test car is, of course strictly relevant only to the prototype trainset.

The body structure is designed to withstand an axial compressive load of 200 metric tons in the passenger portion of the car and 150 metric tons in the raised end portions. The car ends thus become potential collapsible energy absorbers in protection of the passenger compartment. These compressive loads do not meet the AAR interchange requirements for passenger railcar structures.

The body is of tubular, one-piece, stressed-skin construction, its components being stamped or folded from corrosion resistant steel sheet and welded together. The underframe is specially reinforced in the doorstep area in compensation for the structural discontinuity there. The body vertical members are connected together at the top by the cantrails, which in turn are connected by the curved roof members located in line with the vertical members.

The ends of the cars are designed to receive the special frames for access between adjacent cars. The end containing the access door has the part of the frame containing the coupling hook. The other end contains the fixed part of

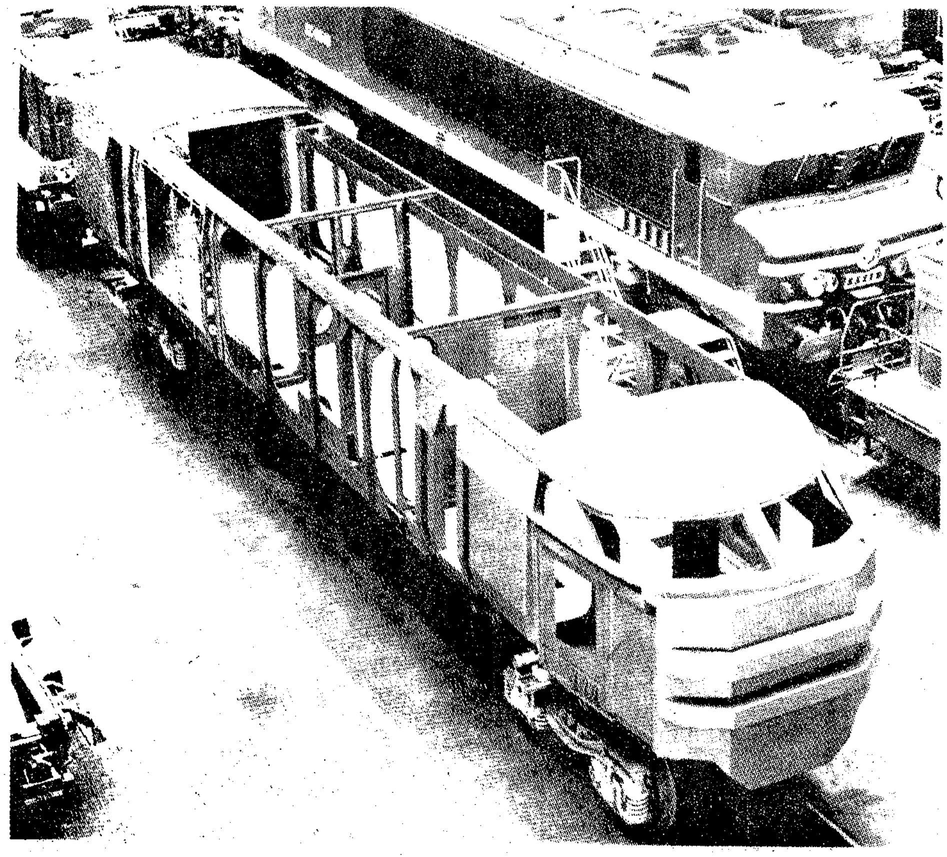


FIGURE 23. BODY FRAME FOR POWER CAR.

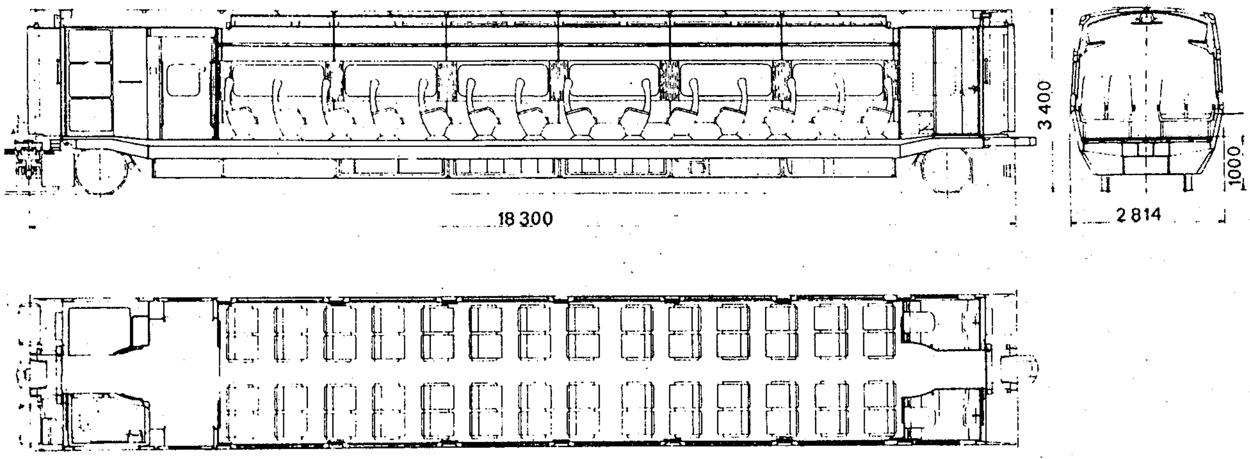


FIGURE 24. DIAGRAM OF SECOND-CLASS CAR.

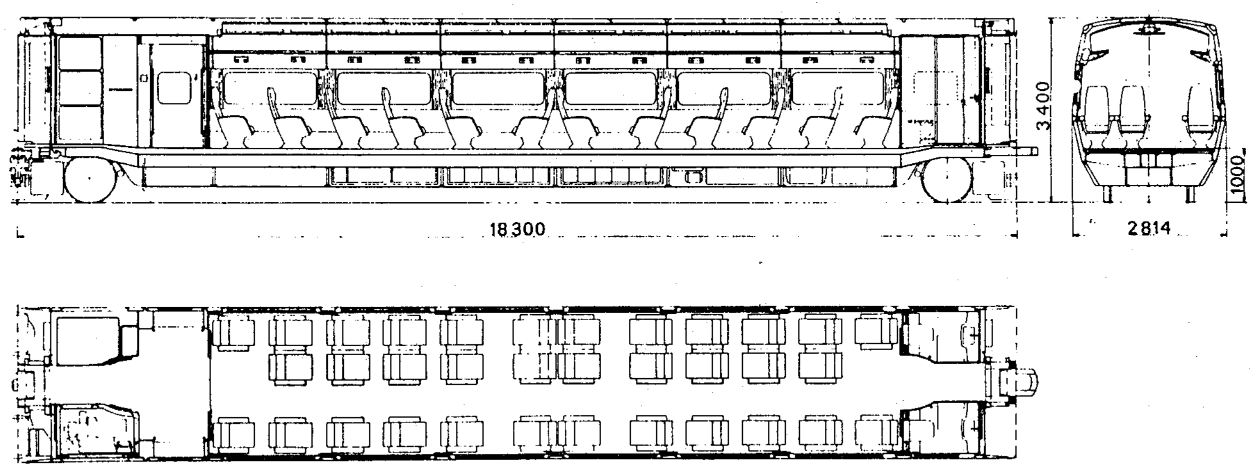


FIGURE 25. DIAGRAM OF FIRST-CLASS CAR.

the frame. The power cars are connected to the trailers and the trailers to each other by these access frames. The frames rest on the air suspension system of the trucks (figure 27) and transmit the tractive and braking effort to the cars.

The frames, assembled by machine welding, consist of a fixed frame bolted to one end of a carbody and a frame free to move, called the "carrying-frame," on which rests the adjacent end of the body of the next car. The part which is free to move carries the drawhook which is designed for a tractive effort of 50 metric tons.

The carrying-frame supports the adjacent end of the next vehicle through the fixed frame, resting on the reinforced rubber spherical joint which provides articulation of one carbody relative to the other. The frames also contain safety devices designed for a tractive effort of 100 metric tons.

AAR REQUIREMENTS

The proposed configuration of the electric TGV trainset (TGV PSE) is a 10-unit trainset weighing 315 metric tons empty. According to this, the buff load would have to be 356,000 daN by AAR standards. The buff load to which the TGV is designed is 196,000 daN per power car and 147,000 daN for the trailer cars. If a shorter consist were used, the power car would meet the 178,000 daN minimum static buff load requirement, but the passenger car still would not.

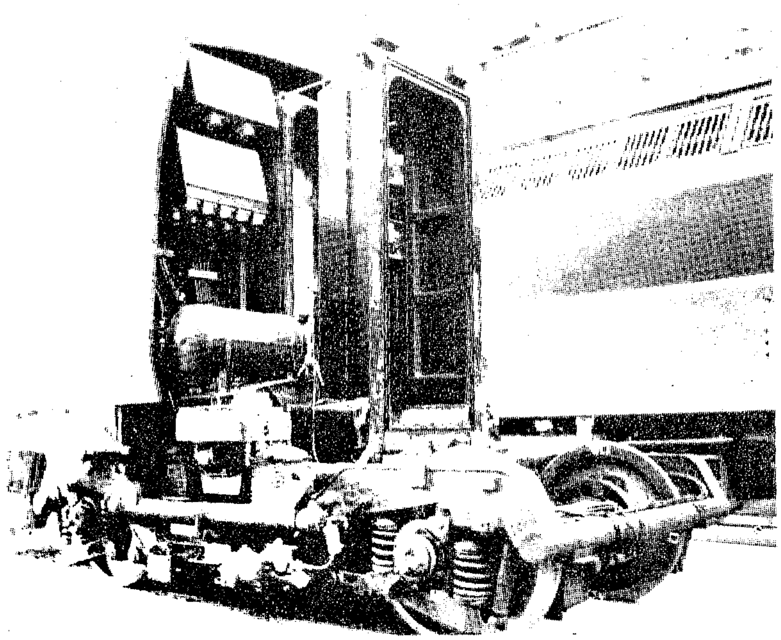
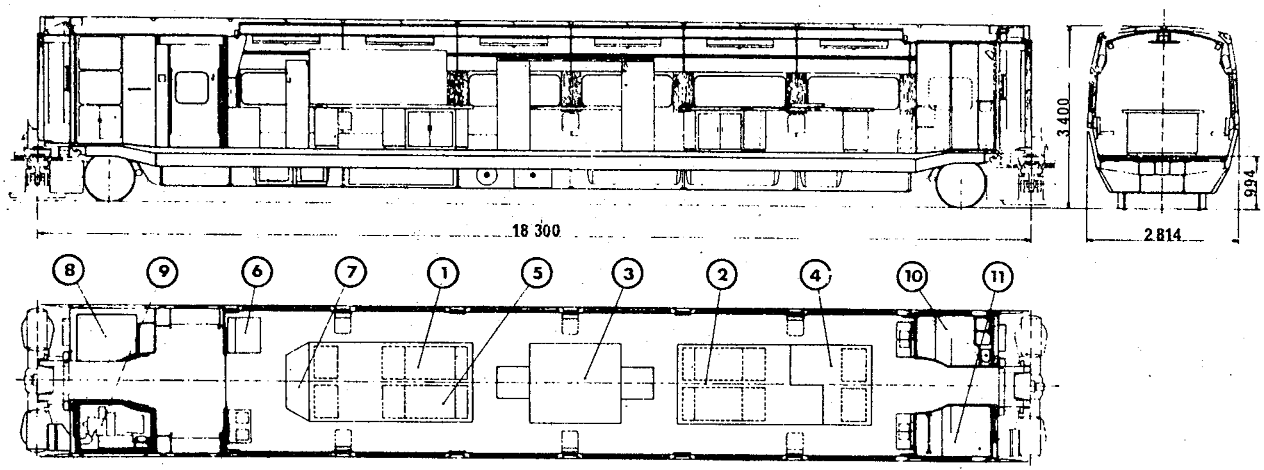


FIGURE 27. TRUCK SUSPENSION COMPONENTS.



STATIONS: 1. TRACK TESTS 2. TRACKS TESTS 3. RAILWAY DYNAMIC MEASUREMENTS 4. STRUCTURAL TESTS 5. BRAKING TESTS 6. CHIEF OF TEST 7. POWER CONTROL 8. CABINETS 9. AIR-CONDITIONING EQUIPMENT 10. PHOTOGRAPHIC ROOM 11. CLOAKROOM

FIGURE 26. DIAGRAM OF TEST CAR.

GERMAN EQUIPMENT - ET403

INTRODUCTION

The ET403 (figure 28) was developed for the intercity link of the German Federal Railroads (DB). The project was started about 1970. Three prototype trains have been built; two are in revenue service, and one is operating in test service only. It serves as a backup to the two in revenue service, which operate between Munich and Bremen.

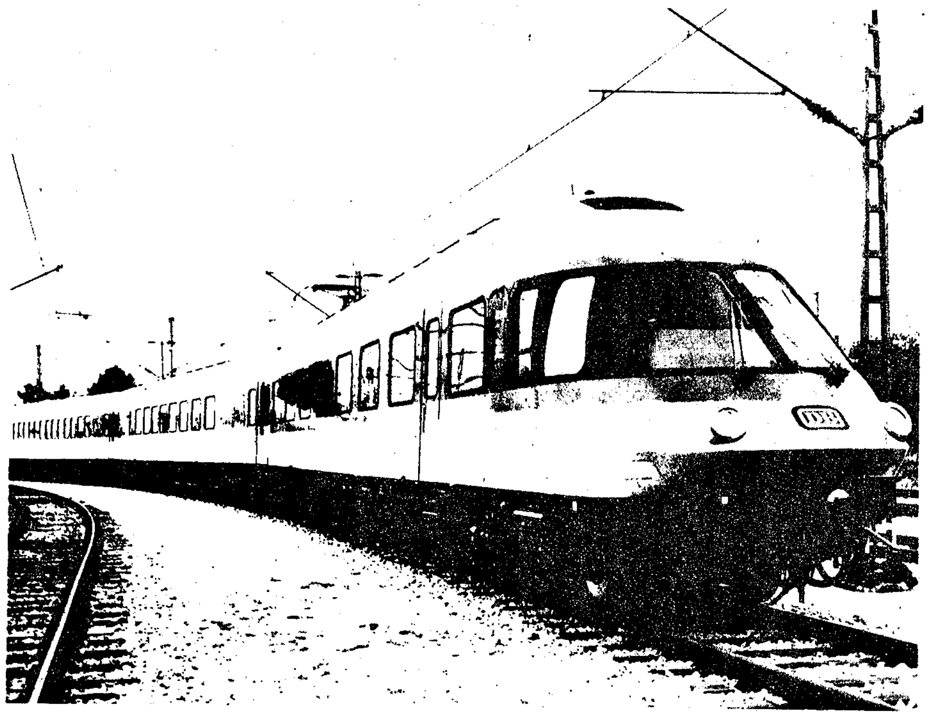


FIGURE 28. EXTERIOR VIEW OF THE SERIES ET403 ELECTRIC TRAIN.

The lightweight design philosophy was based on the need to run on existing track, with many speed restrictions, at speeds up to 200 km/h, but with the same braking characteristics now required for trains operating at 120 to 140 km/h. That is, the stopping distances for the new, faster train must be the same as for the older, slower ones.

High acceleration and deceleration were among the major design requirements. All axles are driven; therefore, the adhesion demand for either braking or acceleration is spread over all wheels, and the vehicle can still be kept light.

The consist includes two first-class cars at the end or controlling cars, a diner-coach combination, and a full coach. The carbodies are made from F6 aluminum alloy. The extrusions run the full length of the carbody, where necessary. They are 58 mm high and 27.5 m long. There is no centersill between the bolsters. The lead carbody weighs 7.62 metric tons unequipped.

The parlor carbody weighs 6.77 metric tons, and the diner weighs 6.80 metric tons. The roof has cross struts, covered by aluminum sheet, to support the dynamic brake grids. Aluminum construction of the ET403 type has been used before in the ET420 equipment, but not with such high and wide extrusions as in the ET403.

Another reason that DB is attracted to aluminum construction is the inherent energy-absorbing capability of the designs. When an accident occurs, the damage tends to be localized at the car ends. Thus, the collision energy is absorbed in the car ends rather than being transmitted into the passenger compartment. Repair is also easier since damaged sections can be cut away and new extrusions spliced in. Collision post strength is 150 metric tons at the underframe connection and 40 metric tons at the roof.

The material that follows was adapted from Forster.[17] Forster's work, acquired as a result of an IPEEP expedition to Europe, was published in German. A translation was performed by nontechnical personnel and as a result the process of adapting a portion of it to the present context included modification for readability by an American technical audience.

Development and construction of the carbody portion of the train was handled by two firms, Linke-Hofmann-Busch (LHB) and Messerschmitt-Bolkow-Blohm (MBB). Among other things, LHB was responsible for the exterior and interior configuration of the entire train, including the introduction of "large-extruded-section construction" for all cars. MBB was responsible for the supporting frame structure, including the choice of aluminum alloys, uniform aluminum section design for all cars, and proof of strength.

CONSTRUCTION METHODS

Lightweight construction was accomplished by using high-strength aluminum alloys and by using the experience gained during the design and construction of the ET420, along with ample computer analysis, for the efficient design of structural members. The structural component assemblies which were proven in the ET420 (the front crossmembers, main crossmembers, and the floor pan design) were used as a basis for the underframe design. In the sidewall construction, however, because of the relatively long sections between the end entrances of the train, the ribbed-sheet technique was discarded in favor of using large extrusion panels of up to 580 mm in height for the side walls below the window ledge and above the windows. The large-section technique was retained for the longitudinal underframe members and for the roof head arc. In the process of totally eliminating ribbed sheeting, extruded sections with pressed connecting sections were used for the installation of the windows.

Strength criteria observed during design of the car frames were:

a. The carbody must be able to withstand without permanent deformation, the following loads, sequentially applied:

[17]Hilmar Forster and Meinrad Liepert, "The Carbody Construction Part of the Rapid Transit Electric Train ET403," Elektrische Bahnen, Vols. 8 and 9, 1973.

150 Mp on the coupler;
40 Mp at the height of the longitudinal roof members;
36 Mp on the truck pivot.

(Note: One megapond (Mp) of force is the weight of one metric ton; one kilopond (Kp) is the weight of a kilogram.)

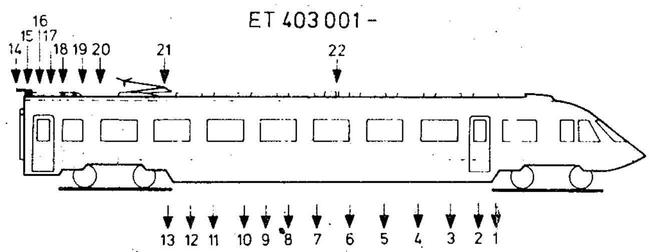
b. The carbody must be able to withstand a vertical static load equivalent to a combination of the service weight, including service load, and a thrust increase of 30 percent as a dynamic equivalent without exceeding design stresses for the respective materials utilized. The maximum loadings for the three different vehicles are grouped together in table 4. Each consists respectively of an evenly distributed load over the entire car length, plus the point loads as shown in figure 29.

The adherence to stress requirements was validated by measurement of stress and deformations on each type of car frame produced.

TABLE 4. MAXIMUM LOADINGS OF CARBODIES.

Maximal Loadings	
End Car	53,400 t
Saloon Car	53,000 t
Dining Car	53,775 t

The structural analysis technique used was a modified approximation method by Biek and Fabry. Its results were validated during stress testing. A cross-sectional view of the completely welded carbody is shown in figure 30 (index numbers refer to the legend on figure 31). The most important extruded sections used in the carbody frame are shown in figure 31 with their dimensions and position.



TOTAL WEIGHT OF POINT LOADS: 11020 kg

DISTRIBUTION

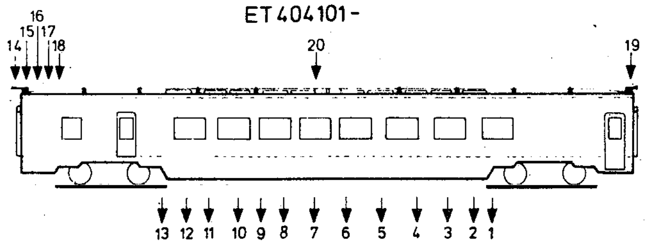
UNDER THE CAR

1	COMPRESSED AIR DEVICES	100
2	ELECTRIC SWITCHING DEVICES	110
3	DRIVE MOTOR CHOKE	1635
4	ARMATURE AND FIELD CURRENT RECTIFIER	270
5	TRANSFORMER	4470
6	TRANSFORMER COOLER	350
7	REFRIGERATION AGGREGATE FOR THE AIR-CONDITIONING SYSTEM	850
8	AIR-CONDITIONING UNITS	550
9	COMPRESSED AIR DEVICES	75
10	BATTERY AND MAIN INVERTER TRANSDUCER CHOKE	1300
11	COMPRESSED AIR DEVICES	370
12	ELECTRIC SWITCHING DEVICES	110
13	COMPRESSED AIR DEVICES	110
	TOTAL	10290

ON THE ROOF

14	HIGH COLTAGE COUPLING	22
15	NONATTENUATED VOLTAGE TRANSDUCER	66
16	ARRESTER	15
17	ROOF END CONNECTOR	17
18	COMPRESSED AIR QUICK RELEASE SWITCH	145
19	DISCONNECTOR SWITCH	25
20	DISCONNECTOR SWITCH	60
21	CURRENT COLLECTOR	260
22	BRAKE RESISTANCES	160
	TOTAL	730

FIGURE 29. CAR WEIGHT DISTRIBUTIONS (SHEET 1 OF 2).



TOTAL WEIGHT OF POINT LOADS: 11287 kg

DISTRIBUTION

UNDER THE CAR:

1	COMPRESSED AIR DEVICES	100
2	ELECTRICAL SWITCHING DEVICES	110
3	DRIVE MOTOR CHOKES	1635
4	ARMATURE AND FIELD CURRENT RECTIFIER	270
5	TRANSFORMER	4470
6	TRANSFORMER COOLER	350
7	REFRIGERATION AGGREGATE FOR AIR-CONDITIONING SYSTEM	850
8	AIR-CONDITIONING UNIT	550
9	COMPRESSED AIR DEVICES	75
10	BATTERY AND MAIN INVERTER TRANSDUCER CHOKE	1300
11	CONVERTER, COOLING CAPSULES, CHOKE	960
12	ELECTRICAL SWITCHING DEVICES	110
13	COMPRESSED AIR DEVICES	100
	TOTAL	10880

ON THE ROOF:

14	HIGH VOLTAGE COUPLING	22
15	NONATTENUATED VOLTAGE CONVERTER	66
16	ARRESTER	15
17	ROOF END CONNECTOR	17
18	COMPRESSED AIR QUICK RELEASE SWITCH	145
19	HIGH VOLTAGE COUPLING	22
20	BRAKE RESISTANCES	120
	TOTAL	407

FIGURE 29. CAR WEIGHT DISTRIBUTIONS (SHEET 2 OF 2).

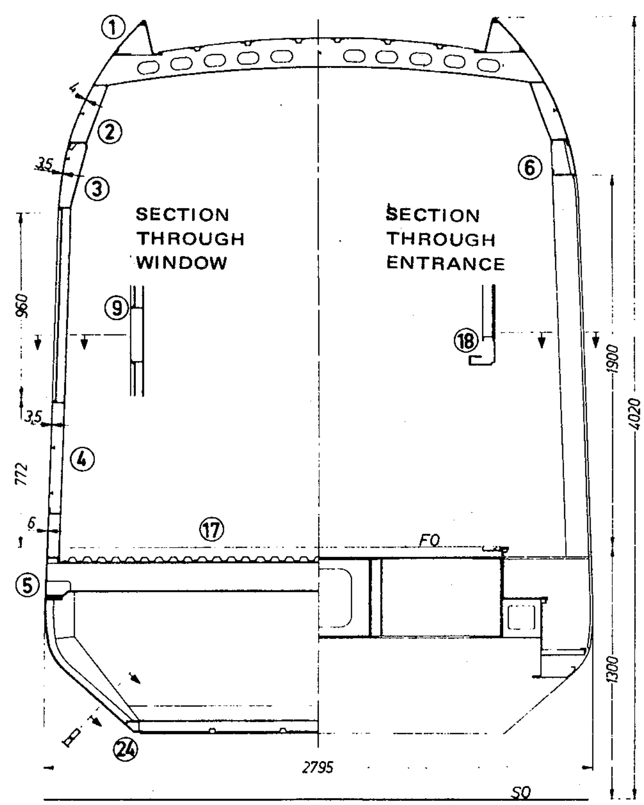
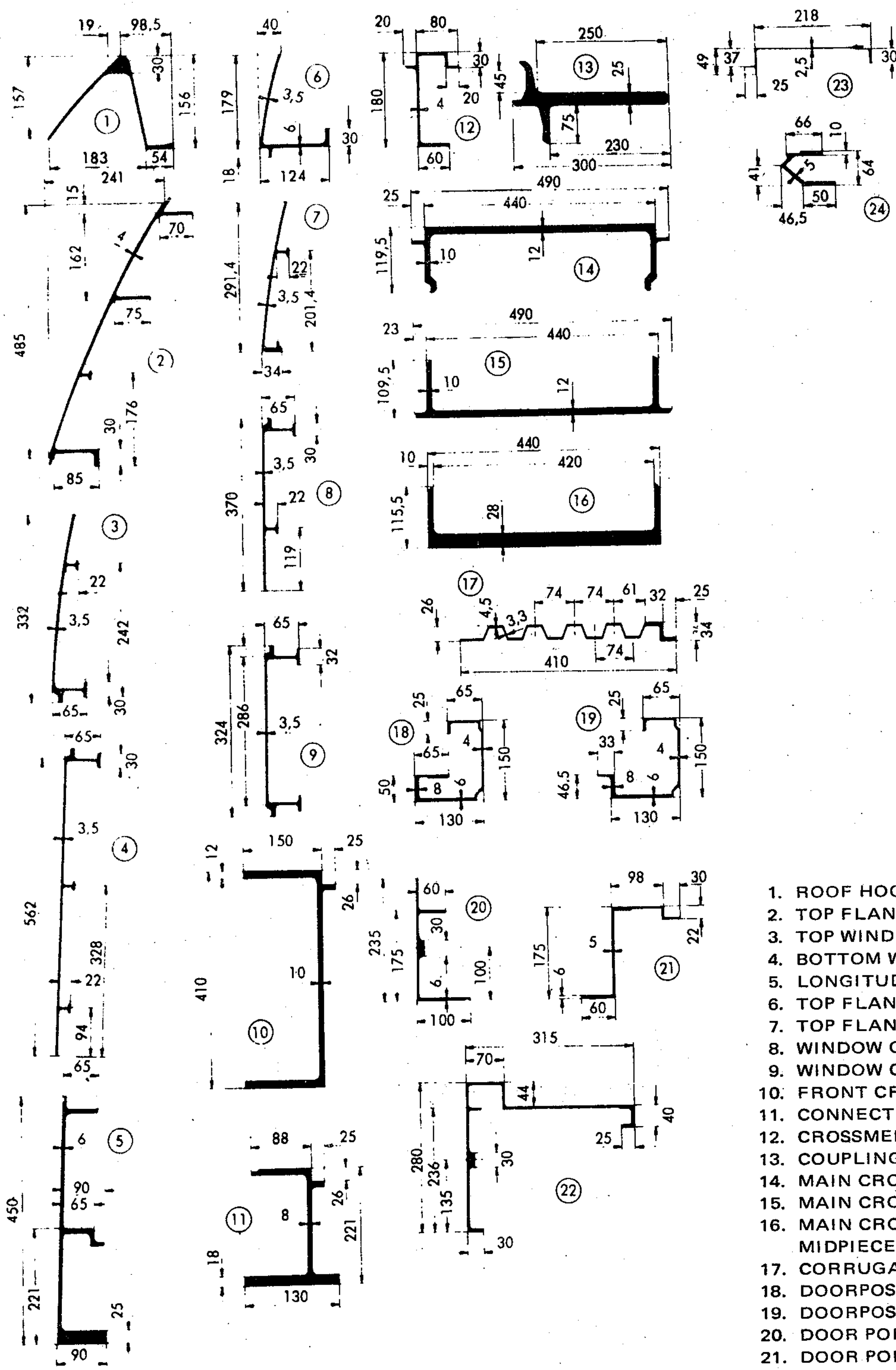


FIGURE 30. CARBODY CROSS SECTION.



1. ROOF HOOD
2. TOP FLANGE
3. TOP WINDOW FLANGE
4. BOTTOM WINDOW FLANGE
5. LONGITUDINAL MEMBER
6. TOP FLANGE (ENTRANCE)
7. TOP FLANGE (LOADING DOOR)
8. WINDOW CROSSPIECE
9. WINDOW CROSSPIECE
10. FRONT CROSSMEMBER
11. CONNECTING MEMBER
12. CROSSMEMBER
13. COUPLING SUPPORT
14. MAIN CROSSMEMBER (TOP SECTION)
15. MAIN CROSSMEMBER (LOWER SECTION)
16. MAIN CROSSMEMBER (LOWER SECTION MIDPIECE)
17. CORRUGATED SHEET
18. DOORPOSTS (ENTRANCE)
19. DOORPOSTS (ENTRANCE)
20. DOOR PORTAL (FRONT WALL)
21. DOOR PORTAL (FRONT WALL)
22. DOOR POSTS (FRONT WALL)
23. PANEL
24. BEAM

FIGURE 31. ARRANGEMENT OF MAIN EXTRUSION SECTIONS USED.

UNDERFRAME

The basic design of the underframe (figure 32) is cleanly arranged by avoiding oblique members and allows easy assembly. Figure 32 shows the underframe of the second-class car from above without the corrugated floor pan. In the middle cars (second-class and dining cars), the underframe consists of the two longitudinal members extending from end entrance to end entrance as well as of the major component assemblies of the floor members with entry pan and main crossmembers. Between these are the normal crossmembers and the supporting crossmembers for the transformers. In the end cars, as a floor plan requirement, the necessary niche for the middle entrances is included, and the projecting member at the front end is adapted to the front-end shape and also for the installation of a Scharfenberg coupling. The underframe design of the end car is shown in figure 33.

To make room for electrical equipment, the air-conditioning and compressed-air systems were mounted between the trucks and the floor pans in the same configuration as used in the ET420 train (figure 34). The lower part of the side wall panel array is integrated into the longitudinal member, designed as a large extrusion section (section 5, figure 30).

In construction of the front crossmembers, the proven modular design of the ET420 was used, but as much as possible with large-extrusion sections in order to minimize welded seams (sections 10 and 13, figure 31). In addition, the requirement that every underframe side be able to sustain a load of 75 Mp applied at the coupling level was realized by reinforcing the treadboard assembly. The main crossmembers are U-shaped extrusion beam (sections 14, 15, 16, figure 31) which, when welded together at the flange ends in the neutral surface of the total cross section, constitute a unitized, distortion-resistant body support member. Built-in reinforcement sections and bushings permit the connection of the truck pivots, air suspension bellows, and emergency guides. For the transformer support members, two U-shaped extrusion sections, similar to those used in the main crossmembers, were employed. Taking into account the fact that the floor pan areas are mutually separated by bulkhead partitions, it was necessary to interrupt the 1.5 mm corrugated floor plate on the underframe at the respective crossmembers (section 12, figure 31). In order to allow maximum beam heights, the corrugated plate, which is utilized to carry compression forces and which is continuously welded to the supporting construction, was interrupted at the main crossmember and transformer supports. Similarly, the corrugated deck was joined to the front crossmembers by means of a butt joint.

SIDE WALLS

As mentioned earlier, an integrally unitized construction approach was used for the side walls, consisting of large extruded panels. Each built-in side wall panel (figure 35) consists of an extruded section extending under and over the window. Similarly, special extruded sections were used for the window posts. In configuring the sections, the crosspieces and flanges necessary for window installation and the attachment of interior panelling were also taken into account (sections 3 and 4, figure 31). The section transition in the window corner roundings between the vertical and horizontal sections was accomplished with machined transition angles. The termination of the side wall panels is provided by an extrusion section designed as a door column (section 18, figure 31). In addition to the longitudinal reinforcement ribs, vertical

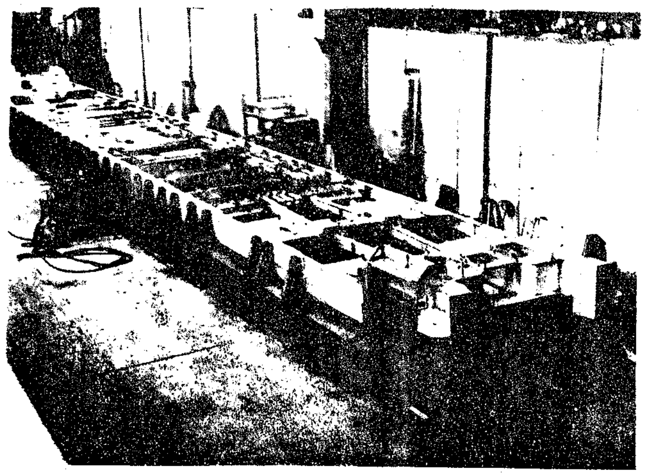


FIGURE 32. SALOON CAR UNDERCARRIAGE.

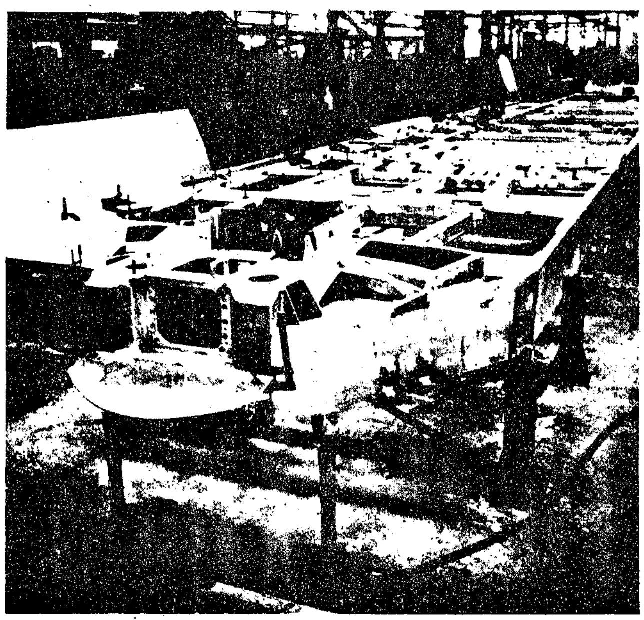


FIGURE 33. END CAR FORWARD UNDERCARRIAGE.

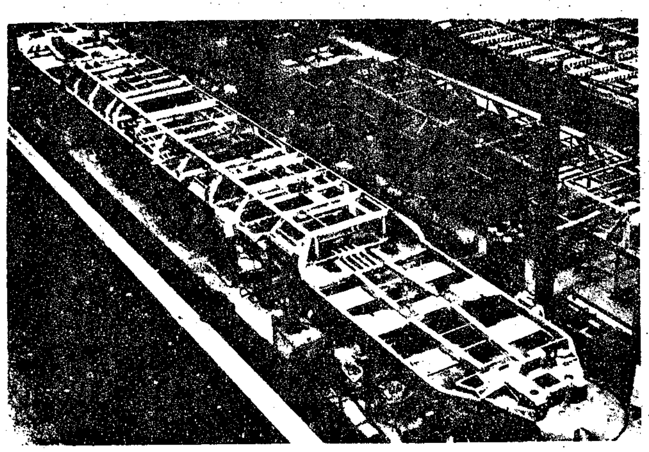


FIGURE 34. END CAR FLOOR PAN UNDERCARRIAGE.

angle sections, which simultaneously provide attachment for the interior paneling, are welded on to stabilize the side wall panels under and over the windows (figure 36).

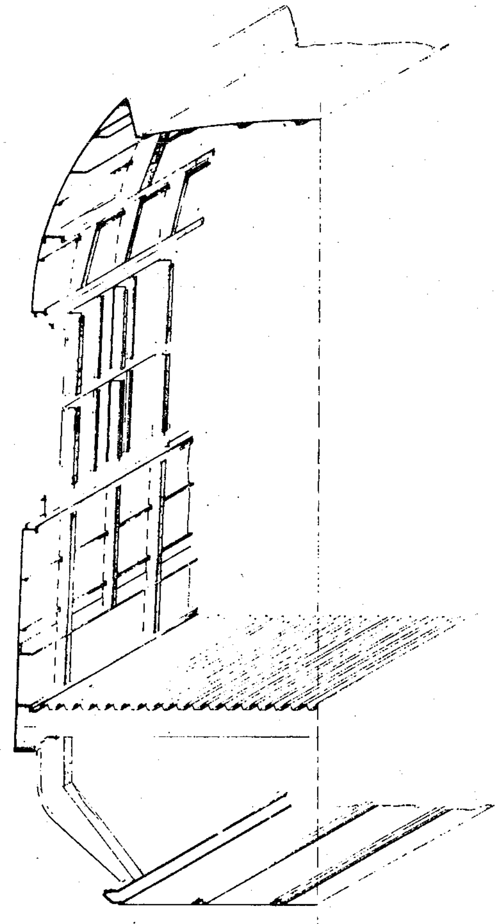


FIGURE 35. PERSPECTIVE OF CARBODY CROSS SECTION.

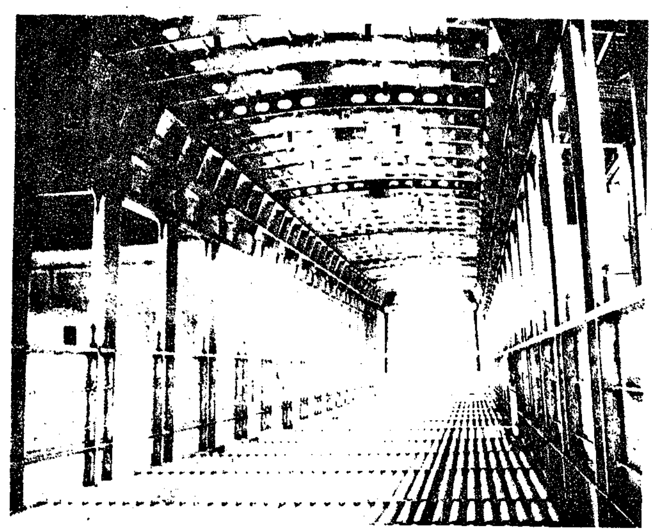


FIGURE 36. INTERIOR BASIC FRAME.

ROOF

The car roof consists of two longitudinal beams, each composed of two large extrusion sections (sections 1 and 2, figure 31) and reinforced by angle sections, and of cross-lying, 2.5 mm thick, extruded roof hoop supports. The covering of the middle section of the roof is provided by roof sheet reinforced by spaced hat sections.

In order to conceal from the station platform the resistors mounted on the roof, the side roof edges (figure 37) are raised in a horned configuration. As an extra benefit, this also reinforces the carbody structure. During the design of the individual structural elements, special attention was paid to the attachment of interior and exterior components. The roof portion under the high-voltage assemblies was also appropriately strengthened.

FRONT WALL

The frame, configured for the aerodynamically designed train head of the end cars (figure 38), is manufactured in the conventional sheet-section manner and mounted on the underframe. Accessibility to equipment in the forward area was facilitated during the design phase by the arrangement of flaps and detachable panel sections. The head walls on the close-coupling ends consist of extruded sections of a 2 mm thick, welded, stressed-sheet covering.

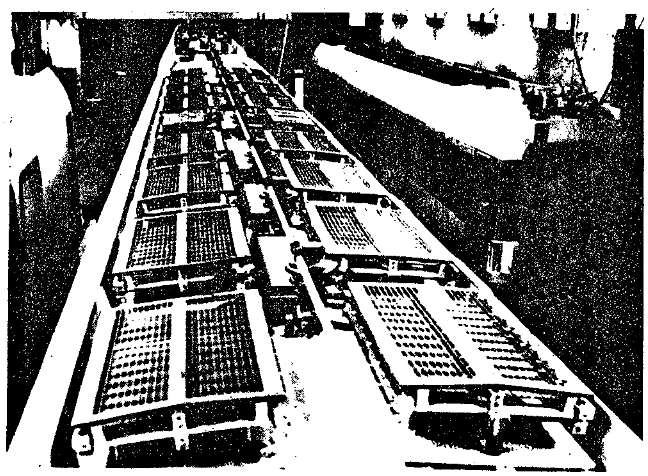


FIGURE 37. CAR ROOF FROM ABOVE.

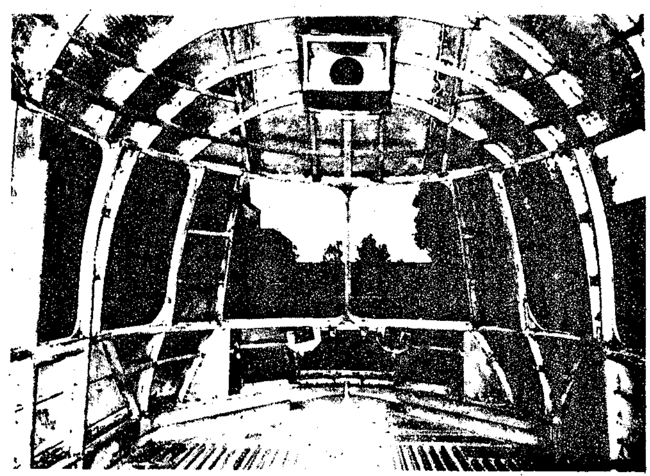


FIGURE 38. FRONT ASSEMBLY OF END CAR.

STRUCTURAL TESTS

Stress, compression and vibration tests were conducted on the prototype car body skeletons. Figure 39 shows a carbody during the tests. The stress tests were effected by loading the car floor and the floor pan with evenly distributed loads along with point loads corresponding to the loading increments shown in table 5. In the stress tests conducted at the point loads with wire strain gages, the permissible maximum stress of 15.30 kp/mm^2 was not even approached. Stresses of between 4.0 and 8.0 kp/mm^2 were measured at the normally highly stressed window corners.

The lower stresses for aluminum, as compared to steel, are characteristic of aluminum construction. Because there is a flexure limitation, the lower modulus of elasticity of aluminum results in lower-stressed cross sections. The measured deflection under load in the center of the car, which in all three car types was less than the permissible value of 19 mm, occurred at load increment 3 of table 5:

a. End car, 10.5 (10.90) mm.

b. Middle cars:

   (1) Saloon (second-class) car, 12.7 (12.85) mm.

   (2) Dining car, 9.98 (11.55) mm.

The values in parentheses indicate the computed flexures based on these displacements. Even at the highest load (150 Mp) on the attachment points of the center buffer-couplings, the measured stresses were consistently below the permissible value of 23.0 kp/mm^2. Figure 40 provides a comprehensive perspective on external stresses at the carbody center at a compressive load of 150 Mp.

The natural frequency (bending mode) of the body is of particular importance in avoiding resonance phenomena emanating from the trucks. In this context, the measured values will differ from the actual values because the incomplete car has the final suspension system installed. Corrections were therefore made on an empirical basis. The tests indicated that the following bending natural frequencies can be expected:

a. End car, 10.2 Hz.

b. Saloon car, 9.4 Hz.

c. Dining car, 10.8 Hz.

Here, as well, the design requirement that the bending natural frequency be greater than or equal to 9 Hz has been met.

Car frame weights were validated at:

a. End car, 62 kg.

b. Saloon car, 6,770 kg.

c. Dining car, 6,800 kg.

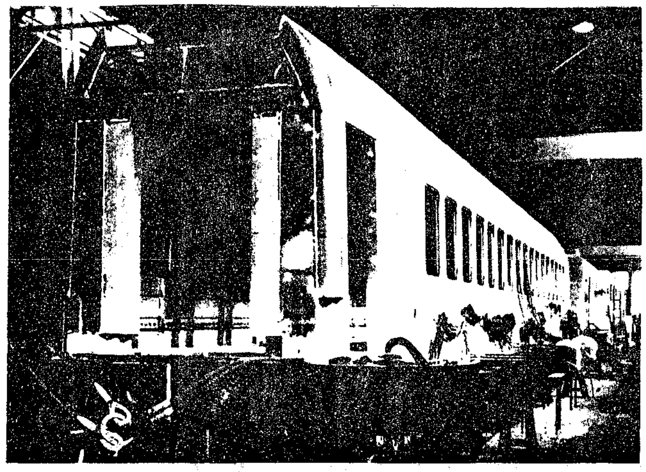


FIGURE 39. CARBODY DURING STRESS ANALYSIS TESTS.

TABLE 5. LOAD INCREMENTS FOR STRESS TESTS.

	Load Increment			
	1	2	3	4
End Car	8,840 t	35,425 t	43,750 t	53,400 t
Saloon Car	6,770 t	33,775 t	42,325 t	53,000 t
Dining Car	8,675 t	37,250 t	42,950 t	53,775 t

Load Increment 1 = Basic Frame Weight + Floor Platform

Load Increment 2 = Service Weight

Load Increment 3 = Load Increment 2 + Payload

Load Increment 4 = Load Increment 3 + 30% Thrust Load

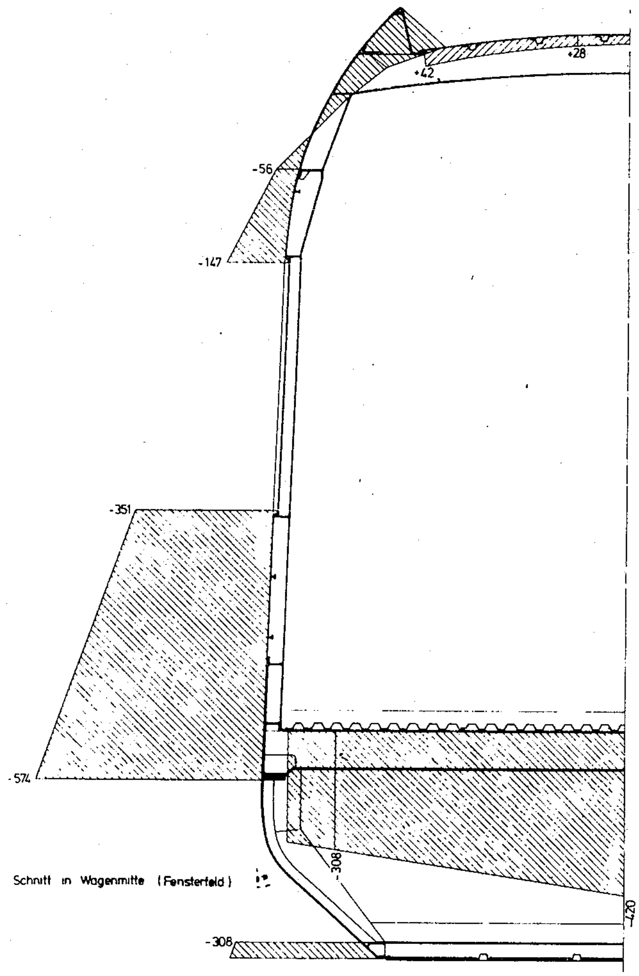


FIGURE 40. STRESS DISTRIBUTION ON BASIC CAR FRAME AT A STATIC COMPRESSION LOAD OF 150 Mp.

ADVANCED TECHNOLOGIES

INTRODUCTION

Two areas of advanced technology in passenger train carbody construction will be discussed: aluminum extrusion and composite materials. The former is already in use, as evidenced by the discussion of the German ET403 trainset. The latter has had broad application in the aerospace field and is beginning to appear on the horizon for the automotive industry. It is only a matter of time before composite materials will find their way into lightweight rail carbody construction.

ALUMINUM EXTRUSIONS[18]

Construction of passenger carbodies with welded aluminum extrusions has been advanced to a fine art by Societe Franco-Belge in conjunction with Alusuisse. When asked to bid on 1,000 MF77 cars for the Paris metro (RATP), Franco-Belge and Alusuisse undertook a detailed study of using aluminum in flow-line production. A factor favoring the use of aluminum was the wide range of extruded profiles already available.

The study showed that a body shell fulfilling all stress requirements and suitable for the Paris metro cars could be built almost entirely from the extruded profiles, which allow modern manufacturing processes such as automatic welding to be employed. Franco-Belge won the order, deliverable in early 1975, over higher bids for steel-constructed cars, making this the first time that aluminum construction became seriously competitive in the rolling-stock market. It should be noted here that the ET403 trainset discussed earlier represented only a few cars, never having been placed in volume production. Shortly, afterward, Franco-Belge won a 100-car order for the Atlanta metro (MARTA) using the same technology as for the Paris cars. After this, another 600 cars were ordered by the Paris metro. This order was for MS79 cars built to UIC standards for use on RATP-SNCF Interconnection services. The result is that Franco-Belge currently has 1,700 cars in production or on order, all to be built using welded-aluminum-extrusion technology.

The body design of the MF77 cars (figure 41), the result of design ingenuity and computer structural analysis, provides more space for passengers in the restrictive conditions of the Paris metro. The design concept of the MF77 rests on the use of long extruded profiles. The floor consists of five longitudinal profiles with no crossmembers; the roof is formed of six further profiles, and two more main profiles act as structural links. There are 15 secondary profiles. When assembled (figure 42) the body forms a self-supporting structure weighing only 2.8 metric tons. The ends are formed of conventionally pressed sheets bolted onto the frame so that they can easily be detached for repairs. Assembly time for a complete body shell is between 300 and 500 hours; for an equivalent steel structure about 1,000 hours would be required.

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[18]"Paris Metro Cars Set Design Standards for Series Aluminum Production," Railway Gazette International, Dec. 1977.

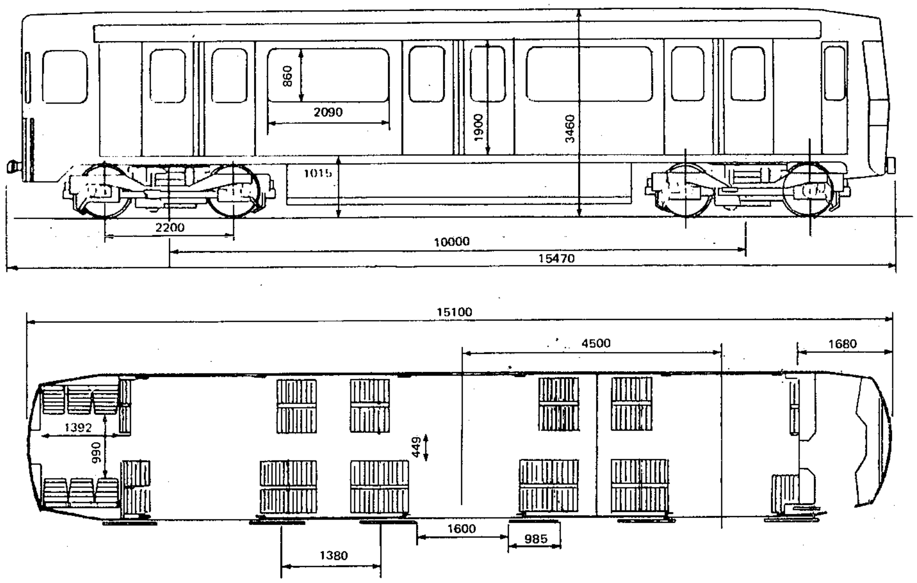

FIGURE 41. MF77 CARBODY

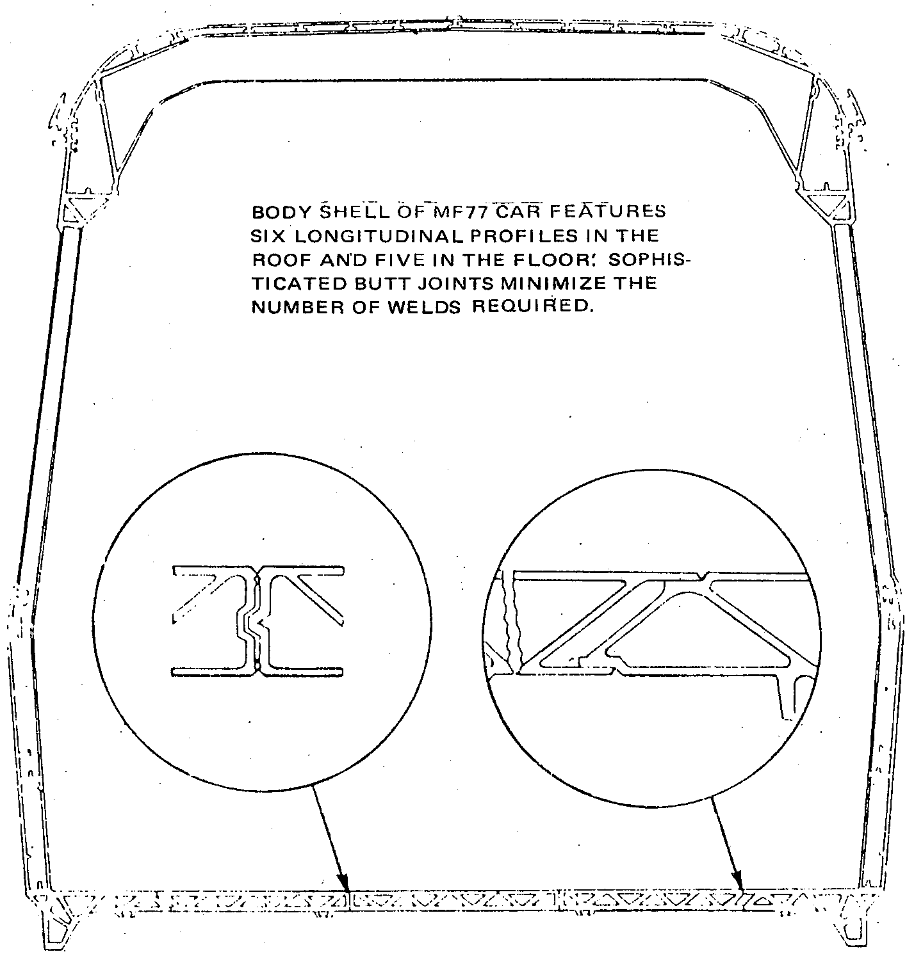

FIGURE 42. CROSS-SECTIONAL VIEW OF MF77 BODY SHELL

The alloys chosen for the MF77 cars are:

a. AlSi 0.7 Mg for extruded profiles;

b. AlMg 2.5 for sheets thicker than 3 mm;

c. AlMg 3 for sheets less than 3 mm thick.

The good mechanical characteristics of AlSiMg are obtained by hardening and tempering. It has an excellent weldability but the thermic cycle affects the hardness of the base metal by partial annealing. The reduction in hardness is accentuated as the metal to be welded is thicker. At butt joints there is a reduction in hardness in an area located 15 to 30 mm from the weld axis. This is compensated for by the fact that the weld is not completed along the entire section. Rapid cooling due to the shape of the profile helps to reduce the significance of the affected zone.

The filler metal employed, AlMg 5, is particularly suitable for the welding duties in the MF77 design and is not subject to cracking.

AlMg 2.5 and AlMg 3 behave well after welding, and the mechanical resistance of a welded joint corresponds to that of the bare metal in annealed state.

Rapid progress has been made in the last few years with Metal Inert Gas (MIG) welding processes, and Franco-Belge considered MIG to be the only process suitable in the MF77 project. The proportion of joints welded automatically is 42 percent both in welded length and in the amount of metal deposited. The equivalent figure for conventional steel structures is 10 percent.

On each car 295 meters of welds are carried out automatically, and a further 402 meters of welds are semiautomated.

Welding is effected using dc supplied from rectifiers or constant-voltage generators. For a small change in voltage, and hence of the arc-length, these allow self-regulation by a considerable variation in current. This is very important for aluminum which is sensitive to variations in the feed of the rod. The welding heads are cooled by air as it is rarely necessary to use currents higher than 200 to 220 A.

Since virtually all welds are carried out on the level, the spray arc procedure is most commonly used. This produces a hot-fusion bath giving well-penetrated welds which are little subject to porosity. Preheating is used for all thicknesses greater than 10 mm.

Franco-Belge set up its own training unit to familiarize its staff with aluminum welding technology. A welder passes through in 4 weeks during which he undergoes theoretical and practical familiarization work and tests. After completing the aluminum welding course, a welder is required to carry out further tests at regular intervals to assure that his standard of welding is maintained. A number of workshops at Raismes have been allocated exclusively for aluminum car construction, and they are thermally insulated to minimize heat dissipation; working areas are protected as far as possible from drafts.

The edges to be welded must be dry and free of oily substance which is often present in aluminum. It is therefore necessary to scour the material after fabrication and then to clean it with benzine or chlorinated hydrocarbons and brush the joints immediately before assembly. The extruded profiles supplied by Alusuisse are fabricated so that the welding edges are shaped in the definitive form to be used in welding. This includes:

a. The welding chamfer;

b. The undercut which allows the aluminum to be deposited without altering the joint at the root;

c. The heel below the undercut which absorbs the transverse shrinkage by limiting deformation;

d. The possibility of rapid dissipation of heat in four directions which helps to minimize reduction in the mechanical properties of the joint.

Faults are likely where piercing may occur. This is avoided by use of clamps which maintain the components rigidly in position during the welding operation; the assembly jig therefore frequently doubles as the welding jig.

As far as possible, straightening is avoided after welding. Instead it is usually possible to bend and preform profiles with the help of clamps. In certain cases hammering is permitted as is the use of blowpipes (to a maximum temperature of 240°C).

When setting up the jigs it was necessary to allow for expansion and shrinkage. A 1-meter bar of aluminum increases in length by 24 mm at 100°C. Rapid cooling is encouraged because of the ribbing of the profiles.

The sequence of welds is also an important factor which required careful study at the planning stage. For example it was found that it was often better to make two rapid welds than a single slow one.

Occurrence of porosity caused for example by dirty edges, poor quality, or lack of gas is best avoided by strict control of welder performance backed by a high standard of training. Cracking emanating from the crater at the end of a weld can be minimized by progressively reducing the current before deenergizing the arc. The welding machine has apparatus fitted to insure this, and automation helps to reduce the number of times arcs are energized and deenergized.

Sweating was used systematically to test the quality of welds on the first 10 bodyshells. Hammer testing is subsequently applied as a function of the faults revealed in the first shells and of the stresses in the welds.

X-ray photography is an essential complementary test, since hammer testing is difficult because of the complexity of the joints and profiles. It is, however, the only way of obtaining reliable information on internal faults. For 3 mm thicknesses a 90 kV X-ray generator is used, and for 10 mm thicknesses a 110 kV unit is employed with exposure times of 95 to 170 seconds.

A prototype body was subjected to a series of static tests in September 1976 at Raismes. A second prototype was sent in October for accelerated fatigue tests at the Brussels laboratory of the Association of Belgian Industries. For 5 weeks the body was subjected to dynamic loads equivalent to a service life of more than 20 years, assuming that annual distance run is 80,000 km and the average number of dynamic stresses or incidents per kilometer is from 4 to 6.

The static and fatigue test confirmed the structural integrity of the bodyshell, and no design changes were necessary.

Once the floor of an MF77 car is assembled from the five profiles, the collision joint and bolster are welded underneath. The curved side walls feature two short, wide extrusions between the three doorways mounted on the floor structure below the windows and two end elements. These are assembled with uprights forming the door frames and a profile above the windows to form the complete sidewall. The roof is fitted with bracing struts before it is attached to the sidewalls. For final welding the complete body shell is placed in a rotating jig which allows the required welding positions to be set up. Some weld details are shown in figure 43.

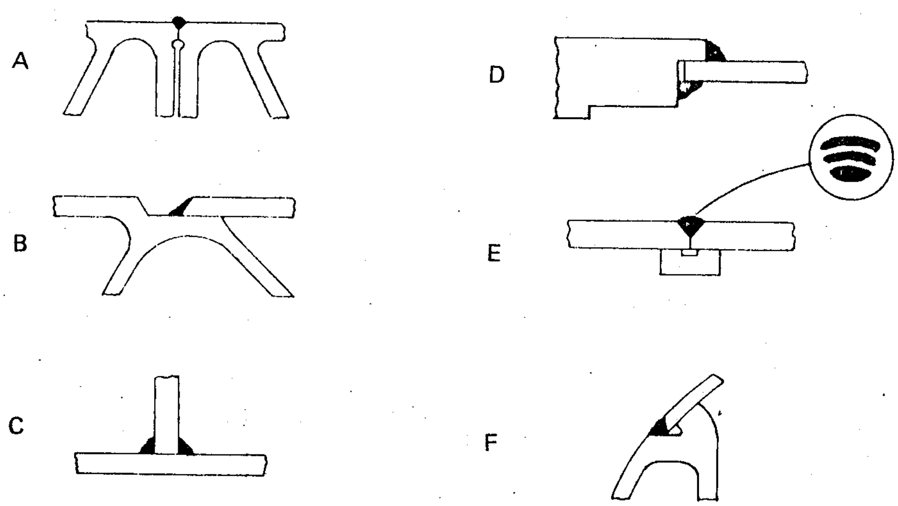

FIGURE 43. WELD DETAILS OF SIX JOINTS ON MF77 BODY SHELL STRUCTURE USE PARAMETERS SHOWN IN TABLE 6.

TABLE 6. PREPARATION FOR WELDING AND WELDING PARAMETERS FOR ALUMINUM BODIES OF MF77 CARS FOR RATP.

Assembly components	Welding process and position	No. of passes	Speed rod feed m/min	Voltage V	Current A	Welding m/min	Gas consump-tion l/min	Preheating C
Floor (Fig. 43A)	MIG automatic, 2 heads level	1	11 to 12	12 to 24	190 to 200	1 to 1•1	24 to 26	None
Floor (Fig. 43B)	MIG automatic level	1	11 to 12	22 to 23	180 to 190	0•8 to 0•9	24 to 26	None
Crossbeam Pivot Collision joint (Fig. 43C)	MIG manual level	2	12 to 14	24 to 25	190 to 200	--	24 to 26	80 to 100
Cross-beam Pivot-ring (Fig. 43D)	MIG automatic fixed angle	2	19 to 20	29 to 30	240 to 250	0•5	24 to 26	80 to 100
T-girder Pivot Collision joint (Fig. 43E)	MIG manual level	3	10 to 15	22 to 26	180 to 220	--	22 to 26	80 to 100
Roof on cantrail (Fig. 43F)	MIG automatic cornice	1	10 to 11	22 to 23	170 to 180	0•75 to 0•8	24 to 26	None

COMPOSITE MATERIALS[19]

As discussed in the introduction to this section, composite materials have had wide use in the aerospace industry but are just beginning to filter out from that beginning. No applications to rail passenger carbody construction are currently known, but they are surely on the horizon since the automotive industry is now moving in that direction. Given the production volume of that industry, composite material technology is likely to emulate aluminum extrusion technology, discussed in the previous paragraphs, and become a highly automated production form. Ford Motor Company is borrowing heavily on the aerospace technology and manufacturing experience in its development of a prototype lightweight automobile using graphite epoxy composite components in the body, chassis, and power train components.

Western Development Laboratories of Ford's Aerospace and Communications Corporation, which has been using the material for years in antennas and other components for space vehicles, is cooperating with the automobile manufacturer in the development program. An aerospace firm has been assigned fabrication of door hinges for the prototype.

The initial feasibility program has involved fabrication of approximately 20 graphite epoxy composite components for a 1978 Granada, including a hood, door, door-guard beam, upper and lower front suspension arms, driveshaft, transmission support and air-conditioning system braces. The experimental graphite epoxy composite driveshaft weighs 5.44 kg compared to 7.89 kg for a conventional steel shaft.

The prototype six-passenger vehicle will utilize considerably more of the composite material, in addition to aluminum and plastics, with weight expected to be about 1,134 kg, approximately 567 kg lighter than Ford's planned 1979 intermediate cars built with conventional materials.

Discussions with graphite epoxy composite material manufacturers have indicated that with a commitment from the automobile industry to utilize the material, it would take 5 to 10 years to increase production from the present level of about 113,400 kg annually to about 45 million kg a year.

Current price of about $55/kg is expected to drop to about $22/kg in the 1980's, but it is believed that cost will fade as an issue over the next few years.

SUMMARY

The most significant technological progress in carbody construction presented here consists of:

a. The transition from corrosion-resistant steel frame and sheath construction to construction with aluminum extrusions; and

[19]"Aerospace Technology Used in Prototype Car," Aviation Week and Space Technology, Dec. 5, 1977.

b. The evolution in analysis technique from classical manual methods through first-generation computer techniques and finally to the powerful finite element method currently in use.

The combination of strong structural analysis capability and production versatility available in aluminum extrusion technology has made possible economically manufactured, lightweight, corrosion-resistant carbodies. Structural optimization analysis makes possible the placement of material in the most efficient configuration. Direct extrusion of the resulting structural forms in aluminum eliminates both the need to create these forms from welded components and the existence of external sheathing.

Welding eliminated by the extrusion process and the elimination altogether of external sheathing represent a direct reduction in labor cost.

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