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High Speed Rail Tilt Train Technology: A State of the Art Survey

ENSCO, Inc., Springfield, VA

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May 92

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U. S. Department of Transportation

Federal Railroad Administration

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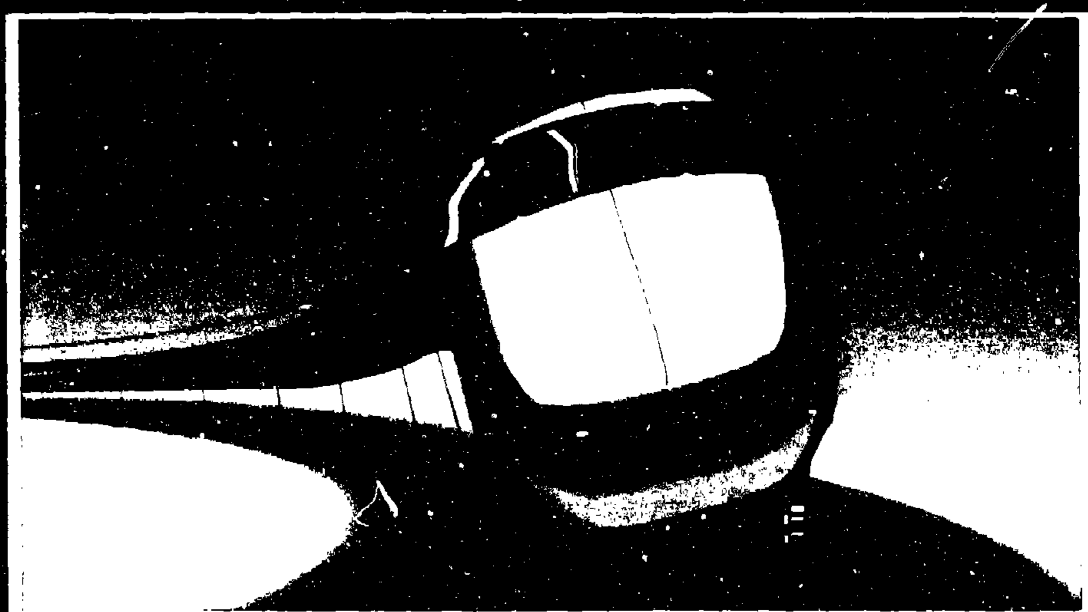

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July 1992

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This report presents an assessment of the technical and operational features of existing and proposed tilt-body rail passenger vehicles. Basic concepts of railroad route selection, track geometry, and curve negotiation are reviewed, and the rationale, advantages and disadvantages associated with body tilting and the techniques used to achieve body tilt are discussed.

An overview of the development status and selected key characteristics of tilt technologies are presented. Issues associated with deployment and operation of tilt-body technologies in the U.S. are identified and analyzed, including a review of U.S. experience to date, areas of incompatibility of foreign tilt technology with existing U.S. equipment and infrastructure, special maintenance procedures and skill requirements, and compliance with FRA and other regulations.

Appendices to the report present discussions on the physics of curve negotiation for conventional and tilting vehicles, the principles of tilting and tilt control strategies and mechanisms, and a description and technical characterization of the principal tilt technologies.

14. SUBJECT TERMS
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PREFACE

Many intercity high-speed train technologies have become an operating reality in recent years. Though mostly of foreign origin, these new trains offer the potential for immediate application in the United States. Each high-speed train was developed to meet the particular operating environment appropriate to the parent country's transportation policy. The resulting diversity in design concepts permits the consideration of a variety of systems in meeting various U.S. application requirements. One particular design concept, the tilt-train technology, offers opportunity for application over the existing rail infrastructure.

This report, one in a series of reports which describe new high-speed rail technologies, presents an overview of the state-of-the-art in tilt-train technology. It is intended to give the reader a better understanding of the unique features of this approach to train design and the variations that exist. Briefly described is the function of the tilting mechanism, whether passive or active, and its performance with respect to passenger ride quality, safety and trip times, which are all influential in passenger acceptance and modal choice. Two trains of the type described in this report, the Spanish Talgo *Pendular* and the Canadian LRC, were previously tested by Amtrak on the Northeast Corridor (NEC), though not used in revenue service. Currently being considered for test and revenue service in the NEC is the Swedish X2000, also covered in this report as well as in an earlier report on the Safety Relevant Observations on the X2000 Tilting Train.

This report was prepared for the Volpe National Transportation Systems Center (VNTSC) in support of the United States Department of Transportation (U.S. DOT), Federal Railroad Administration's (FRA) Office of Research and Development. The authors wish to thank Robert M. Dorer of VNTSC and Arne J. Bang of the FRA Office of Research and Development, for their direction, helpful guidance and input during the preparation of this document.

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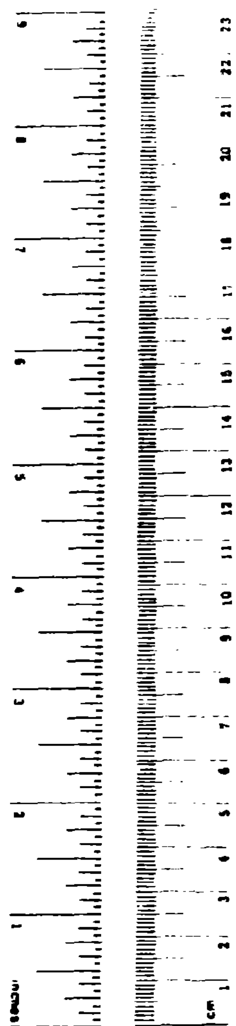
METRIC (SI*) CONVERSION FACTORS

APPROXIMATE CONVERSIONS TO SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
	LENGTH			
in	inches	2.54	millimetres	mm
ft	feet	0.3048	metres	m
yd	yards	0.914	metres	m
mi	miles	1.61	kilometres	km
	AREA			
in^2	square inches	645.2	millimetres squared	mm^2
ft^2	square feet	0.0929	metres squared	m^2
yd^2	square yards	0.836	metres squared	m^2
mi^2	square miles	2.59	kilometres squared	km^2
ac	acres	0.395	hectares	ha
	MASS (weight)			
oz	ounces	28.35	grams	g
lb	pounds	0.454	kilograms	kg
T	short tons (2000 lb)	0.907	megagrams	Mg
	VOLUME			
fl oz	fluid ounces	29.57	millilitres	mL
gal	gallons	3.785	litres	L
ft^3	cubic feet	0.0328	metres cubed	m^3
yd^3	cubic yards	0.0765	metres cubed	m^3

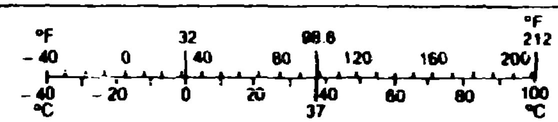
NOTE: Volumes greater than 1000 L shall be shown in m^3.

Symbol	When You Know	Multiply By	To Find	Symbol
	TEMPERATURE (exact)			
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



APPROXIMATE CONVERSIONS TO SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
	LENGTH			
mm	millimetres	0.039	inches	in
m	metres	3.28	feet	ft
m	metres	1.09	yards	yd
km	kilometres	0.621	miles	mi
	AREA			
mm^2	millimetres squared	0.0016	square inches	in^2
m^2	metres squared	10.764	square feet	ft^2
km^2	kilometres squared	0.39	square miles	mi^2
ha	hectores (10 000 m^2)	2.53	acres	ac
	MASS (weight)			
g	grams	0.0353	ounces	oz
kg	kilograms	2.205	pounds	lb
Mg	megagrams (1 000 kg)	1.103	short tons	T
	VOLUME			
mL	millilitres	0.034	fluid ounces	fl oz
L	litres	0.264	gallons	gal
m^3	metres cubed	35.315	cubic feet	ft^3
m^3	metres cubed	1.308	cubic yards	yd^3
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



These factors conform to the requirement of FHWA Order 5190.1A

* SI is the symbol for the International System of Measurements

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1. INTRODUCTION

1.1 OBJECTIVES

This report presents a survey of the technical and operational features of existing and planned tilt-body rail passenger vehicles. It follows the general format of the Federal Railroad Administration (FRA) Report entitled, "Safety Relevant Observations on the X2000 Tilting Train,"[1] but with a slightly broader scope and emphasis.

1.2 DATA SOURCES

In preparation of this report, information was drawn from public sources. Technical and illustrative material was also requested from the developers, suppliers, and operators of the different technologies. The variable level of detail in the technical descriptions and characterizations presented in Appendix C reflects differences in the availability of such information.

The data in the public domain were identified through on-line searches of the National Technical Information Service (NTIS) and the Transportation Research Information Service (TRIS) databases, manual and on-line searches of holdings in the Canadian Institute of Guided Ground Transport (CIGGT), ENSCO, FRA, and National Research Council of Canada libraries, including recent (post-1980) periodicals and journals, and the files of senior researchers at CIGGT and ENSCO. This information was supplemented by materials provided by the FRA Offices of Research and Development and Railroad Development.

To ensure that the developers, suppliers and operators of tilt-body technologies world-wide had an opportunity to provide up-to-date information, requests for data were sent to Bombardier, Talgo Pendulentes S.A., SIG, FIAT Ferroviara, ABB, EB Strommens, and JR-RTRI as suppliers, and to VIA Rail Canada, RENFE (Spain), SBB (Switzerland), SJ (Sweden), FS (Italy), DB (Germany), NSB (Norway), OBB (Austria), and JR-SHIKOKU (Japan) as operators.

1.3 APPROACH

The most significant implications of tilt-body technologies are for the trade-offs and compromises that have been, and continue to be, made between the "best" track for freight operations and the "best" track for passenger operations or where space and/or economic constraints limit options for performance improvement.

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[1] Safety Relevant Observations on the X2000 Tilting Train, prepared for the FRA Office of Research and Development, DOT/FRA/ORD-90/14 (NTIS: PB 91-129668), December 1990.

The nature of the differences between optimized track for freight and passenger traffic are explained and the potential advantages and limitations of tilt-body technologies noted. Since many of the important characteristics of these technologies could lead to requirements for waivers, regulatory revisions, and/or new rulemaking should there be a desire to operate such technologies in the United States, it was felt that these basic issues must be understood.

1.4 REPORT ORGANIZATION

Section 2 of this report presents and discusses basic concepts of railroad route selection, track geometry, the physics of curve negotiation, the rationale for body tilting, the advantages and disadvantages associated with body tilting, and the techniques used to achieve body tilt.

Section 3 provides an overview of the development status and selected key characteristics of the tilt technologies examined in this assessment.

Section 4 contains an overview of U.S. experience with tilt-body technologies to date.

Section 5 identifies and discusses issues associated with deployment and operation of tilt-body technologies in the United States. This section includes an examination of areas of incompatibility with existing U.S. equipment and infrastructure, special maintenance procedures and skill requirements, and compliance with FRA and other regulations.

Appendix A presents a detailed development of the physics of curve negotiation for conventional and tilting vehicles, while Appendix B presents a technical discussion of the principles of tilting and tilt control strategies and mechanisms. Appendix C contains a more-or-less detailed description and technical characterization of each of the technologies summarized in Section 3.

2. SOME BASIC CONCEPTS

To understand the advantages and disadvantages of tilt-body rail vehicles, the reader requires an appreciation of some basic concepts that are fundamental to railroad design and operation. This section of the report addresses this requirement through simplified examples and illustrations. A more rigorous and comprehensive treatment of these concepts is provided in Appendix A.

2.1 GEOMETRY OF A RAIL ROUTE: CURVES AND GRADES

From an operational viewpoint, the essence of any railroad route can be captured in terms of its three-dimensional geometry: how the plane of the track is located vertically and horizontally relative to the three reference axes. This geometry poses the fundamental constraint on railroad operations. **Figure 2.1** illustrates the elements of horizontal *route or alignment* geometry. Tilt-body technologies are an engineering response to the limitations of route geometry.

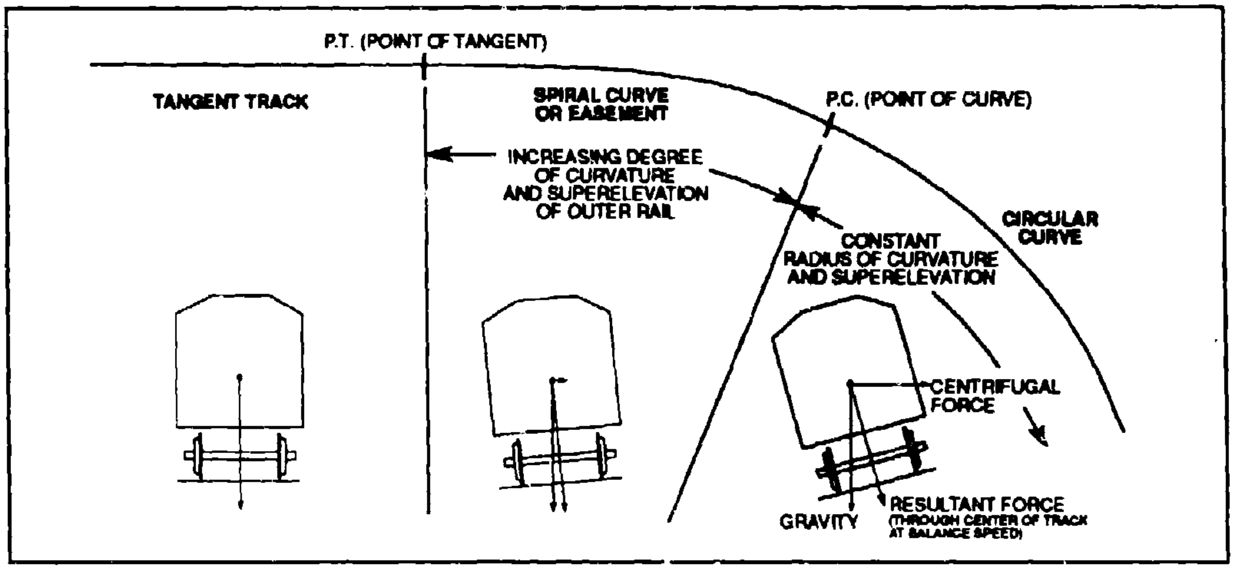

Figure 2.1: Elements of Horizontal Route Geometry

A straight section is referred to as *tangent* track. Curves may be described either in terms of the number of degrees (the smaller the value, the shallower the curve), or in terms of the length of the radius (the larger the value, the shallower the curve). In North American practice, the *degree of curvature* is defined as the central angle subtended by a chord of 100 feet between two points on the centerline of the curve. The lateral acceleration or force experienced by a vehicle traversing a curve at a given speed increases as the degree of curvature increases.

To reduce the rate of change in lateral acceleration (and thus force) between tangent and curved sections of track, all railroad tracks incorporate what is known as a *transition spiral* or *spiral* *easement*, also shown in **Figure 2.1**. The transition spiral permits a gradual and controlled increase in curvature and superelevation (discussed below), which serves to make the rate of change in acceleration (and force) less noticable, both by riders and in terms of the forces imposed on the track. In conventional (i.e., freight) railroad practice, the length of the transition spiral is driven primarily by the allowable rate of change in superelevation rather than curvature, and in some cases, by physical limitations of the track layout. The objective is to match superelevation with curvature throughout the transition.

In terms of vertical geometry (**Figure 2.2**), the key measures are *slope* or *gradient* (the rate at which the elevation of a track changes), and the radius of curvature at crest or trough. The gradient is usually measured as a percent (a 1.5% grade means a change in elevation of 1.5 feet for every 100 feet of horizontal distance). Vertical curves may be described by degrees or radius of curvature, just as for horizontal curves. Vertical curves, whether at crests or in troughs, also require transition spirals. Conventional railroad practice in North America is to design vertical curves as parabolas, rather than circular curves; parabolas provide an inherent transition from the uniform gradient line.

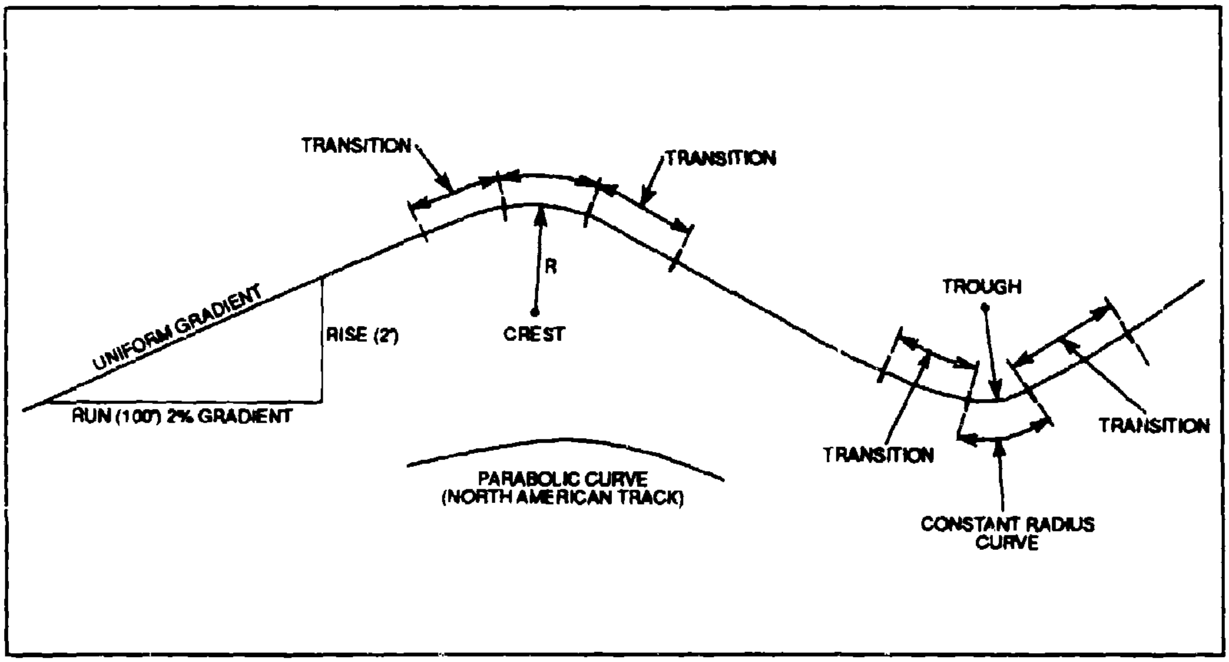

Figure 2.2: The Elements of Vertical Alignment Geometry

The expectations of travellers, with respect to comfort, are the basis for most geometric limits. These limits are the levels of lateral and vertical acceleration, expressed as a proportion of gravitational acceleration (g), that have been shown to be acceptable to the majority of passengers - 0.08g to 0.10g for lateral and downward vertical accelerations, 0.05g for upward accelerations. Most passengers cannot detect accelerations of less than 0.04g.[2] Passenger comfort is also a consideration when designing transition spirals. An acceptable level for the rate of change in lateral acceleration, termed lateral *jerk*, has been shown to be around 0.03g per second (for levels of acceleration up to 0.1g).[3]

Because passengers are typically more sensitive to the unweighting sensation caused by traversing a crest at speed - the slightly unpleasant effect one feels when going over the top of a hill on a highway - the minimum radius of curvature required at crests on track built especially for high-speed passenger services is larger than that specified for troughs in an effort to keep the accelerations within the acceptable levels.[4]

In contrast, the American Railroad Engineering Association (AREA) track standards[5] that govern the design of lines in the United States and Canada require larger-radius curves in troughs than at crests, the exact opposite of the situation for tracks built to accommodate higher-speed passenger services. This is because the concern with the design of track used for freight is control of the behavior of cars and locomotives and the inter-vehicle forces, especially in long trains. In passing over a trough, there is a tendency for rear cars to crowd on to those in front, with a consequent sudden reversal of stress in the draft gear.[6] As a result, troughs or sags are made more gradual.

Although carbody tilting reduces the amount of lateral force perceived by passengers, the ratio of lateral and vertical forces (L/V ratio) at the wheels is a critical determinant of curving safety.

Table 2.1 summarizes typical values for horizontal curves, vertical curves and gradients for freight tracks, for the mixed-use Northeast Corridor in the U.S. and for high-speed passenger-service-only tracks constructed abroad. The question of compatibility between the characteristics of existing rights-of-way with the geometric requirements of optimized high-speed passenger infrastructure is very important.

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[2] "Building the World's Fastest Railway," Andre Prud'homme, Railway Gazette International, January 1979; "The Development of a Truck for Narrow Gauge Line Limited Express Vehicles of Next Generation," Dr. S. Koyanagi, RTRI Quarterly Reports, V.26 No. 2, 1985; and "Tilt System for High-Speed Trains in Sweden," R. Persson, IMechE (Railway Division) Seminar on Tilting Body Trains, December 1989.

[3] High Cant Deficiency Testing of the LRC Train, the AEM-7 Locomotive, and the Amcoach, Boyd, P.L., Scofield, R.E., Zaiko, J.P., prepared by ENSCO, Inc. for the FRA Office of Freight & Passenger Systems, Report No. DOT-FR-81-06, (NTIS: PB 82-213018), January, 1982, p. 5-32.

[4] Prud'homme, see footnote 2.

[5] Manual for Railway Engineering, Vol 1., Ch. 5, Part 3, p. 5.3.13, AREA 1990

[6] The Design of Railway Location, Clemant C. Williams, John Wiley & Sons, Inc., 1917, p. 434.

TABLE 2.1
TYPICAL GEOMETRIC VALUES FOR FREIGHT AND DEDICATED PASSENGER TRACK ALIGNMENTS

CHARACTERISTIC	Typical North American Freight Line (No Passenger)	Northeast Corridor** 200 km/h Passenger (125 mph) 80 km/h Freight (50 mph)	JR, Shinkansen, New Tokaido 200 km/h (125 mph) Passenger Only	France TGV 300 km/h (186 mph) Passenger Only
Horizontal Curvature	198 to 1737 m (650 to 5700 ft)	436 to 1737 m (1430 to 5700 ft)	1890 to 2500 m (6200 to 8200 ft)	6096 m (20,000 ft)
Vertical Curvature	See Note *	See Note *	Crest - 14,940 m (49,000 ft) Trough - 10,060 m (33,000 ft)	(minimums) Crest - 16,000 m (52.490 ft) Trough - 14,000 m (45,930 ft)
Gradient (%)	Less than 1%	1.5 to 2.0	1.5 to 2.0	2.5 to 3.5

* AREA 1990 Manual for Railway Engineering, Vol 1, Ch.5, Part 3, page 5.3.13, expresses standard for vertical curvature in terms of allowable maximum rate of change of gradient in feet per 100 feet of curve length. The recommended limits are 0.05 for troughs and 0.10 for crests (i.e., to go from a 1° gradient to level track would require [1/.05] x 100, or 2000 feet). This defines a parabola rather than a constant-radius curve.

** Amtrak Northeast Corridor Operating Rules and Instructions, October 1989; Amtrak 1987 Track Chart.

Many existing railroad routes in the U.S. reflect the requirements of freight railway operations. The general requirement for a freight alignment is that it minimize route length while permitting operation at a relatively slow but steady speed, with the maximum (controlling) gradient limited by the ability of equipment to start a heavy train from a standing start.

The question of compatibility between the characteristics of existing rights-of-way with the geometric requirements of optimized high-speed passenger infrastructure is very important. Many existing railroad routes in the U.S. reflect the requirements of freight railway operations. The general requirement for a freight alignment is that it minimize route length while permitting operation at a relatively slow but steady speed, with the maximum (controlling) gradient limited by the ability of equipment to start a heavy train from a standing start.

To control the costs of track maintenance in curves, freight alignments seek long straight (tangent) sections connected by the shortest length of curved track, with a radius of curvature that will permit constant speed operation. Where topographic relief is a factor, freight alignments sacrifice good *horizontal* geometry to maintain acceptable gradient with a minimum of tunnelling. This means that many existing rail alignments in the U.S. have been laid out with inherent speed restrictions from the viewpoint of a passenger operator.

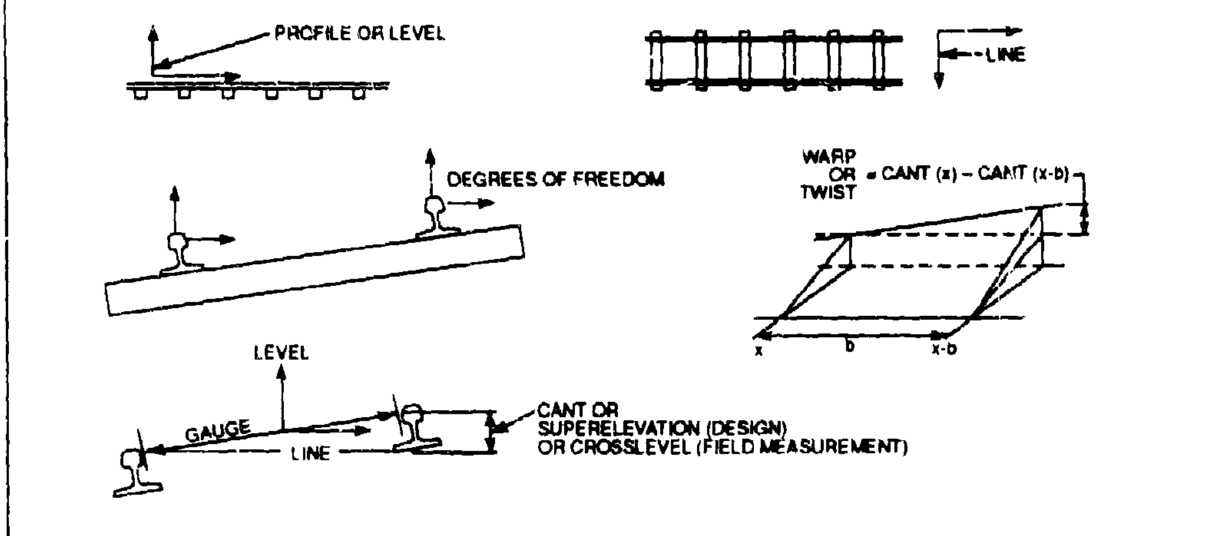
In contrast to rail freight requirements, an optimum passenger alignment minimizes achievable "trip time" through a combination of route length reduction and the elimination of geometric restrictions on speed, so that a somewhat longer route with superior geometry may be preferrable to a shorter, but slower alignment. The tradeoffs have to be made among life-cycle costs and incremental revenues arising from improved performance. Since the power-to-weight ratio, adhesion control capabilities, and safer use of momentum of high-speed passenger trains allow them to accept much steeper gradients (up to 3.5%) over longer distances, high-speed alignments tend to trade off vertical geometry where possible to maintain good horizontal geometry and avoid future speed restrictions. There may, of course, be terrain and other considerations that demand higher curvatures while restricting maximum superelevation.

This fundamental dichotomy between optimized freight and passenger alignment geometries, and the unavoidable alignment restrictions, are the driving forces behind the existence of tilt-body passenger equipment. Since only a very limited number of corridors can justify investment in new dedicated passenger track on an optimized alignment, and the competitive pressures to improve trip time, ridership and revenue are constant, there is an obvious and immediate appeal for a much less expensive, equipment-based partial solution to what is basically an infrastructure constraint.

2.2 TRACK GEOMETRY

Figure 2.3 illustrates the basic components of track geometry. In U.S. practice, these measures are *line* (the longitudinal alignment of the track in the horizontal plane, relative to a surveyed datum), *profile or level* (the longitudinal alignment of the track in the vertical plane, relative to a surveyed datum), *gauge* (or *gage* in U.S. railroad parlance: the distance between the inner faces of the running rails, by convention measured at a point 15.9mm (0.625") below the top of the rail), *superelevation or cant* (the nominal or design difference in vertical elevation between the heads of the two rails; for the actual measured difference at a given point on track, North American railroaders use the term *crosslevel*), and *warp or twist*, which is the difference in superelevation measured at two points on the track (usually over a distance of 9.5m (31 ft) for spirals or 19m (62 ft) for curves in North America, but over approximately 30m (100 ft) elsewhere). In U.S. practice, the measurements are taken from the two extremes of crosslevel found within the specified distance.

In the context of this report, superelevation and its effects on passenger comfort and operational safety are of principal concern. The other elements are important to the overall quality of the track, and ultimately to the ride perceived by a passenger in a train using that track and to the safety of train operation, but consideration of these elements lie outside the scope of this report.



(Note: Track cant in this context refers to superelevation, not "rail-head cant")

Figure 2.3: The Elements of Track Geometry

2.3 NEGOTIATING A CURVE: SOME SIMPLE PHYSICS

To get any vehicle that is moving along a straight line at constant speed to change its direction of motion and follow a curved path, there has to be some acceleration (and thus force) laterally inward toward the center of the curve, as illustrated in **Figure 2.4 (a)**. In the case of a rail vehicle, the acceleration, and thus, the force comes from contact between the wheels and the rails. However, forces occur in pairs (the *equal and opposite reaction* of Newton's third law) so that there also appears to be a force acting laterally outwards. This force, which is what passengers are aware of during curving, is termed *centrifugal* force. This force is a function of the weight of the passenger, the speed of the vehicle, and the radius of curvature. Gravity, as well, exerts a downward force on the vehicle and its contents.

At low-speed, or with gentle curves, the lateral force would not cause much discomfort, even if the curve were not banked (superelevated or canted), as in **Figure 2.4 (a)**. However, as speed increases, or curves become tighter, the force level increases, until eventually passengers no longer find the ·de acceptable. As noted above, passenger railroad designers and operators worldwide have established that this occurs once the perceived lateral force exceeds about 10% of the passenger's weight.[7]

As described above and illustrated in **Figure 2.3**, railroad track in curves is not flat; rather, it is banked (superelevated or canted) with the outside rail raised relative to the inner rail. The

[7] Railway Passenger Ride Safety, Owings, R.P., Boyd, P.L., DOT-FRA/ORD-89/06, April 1989, Table 6

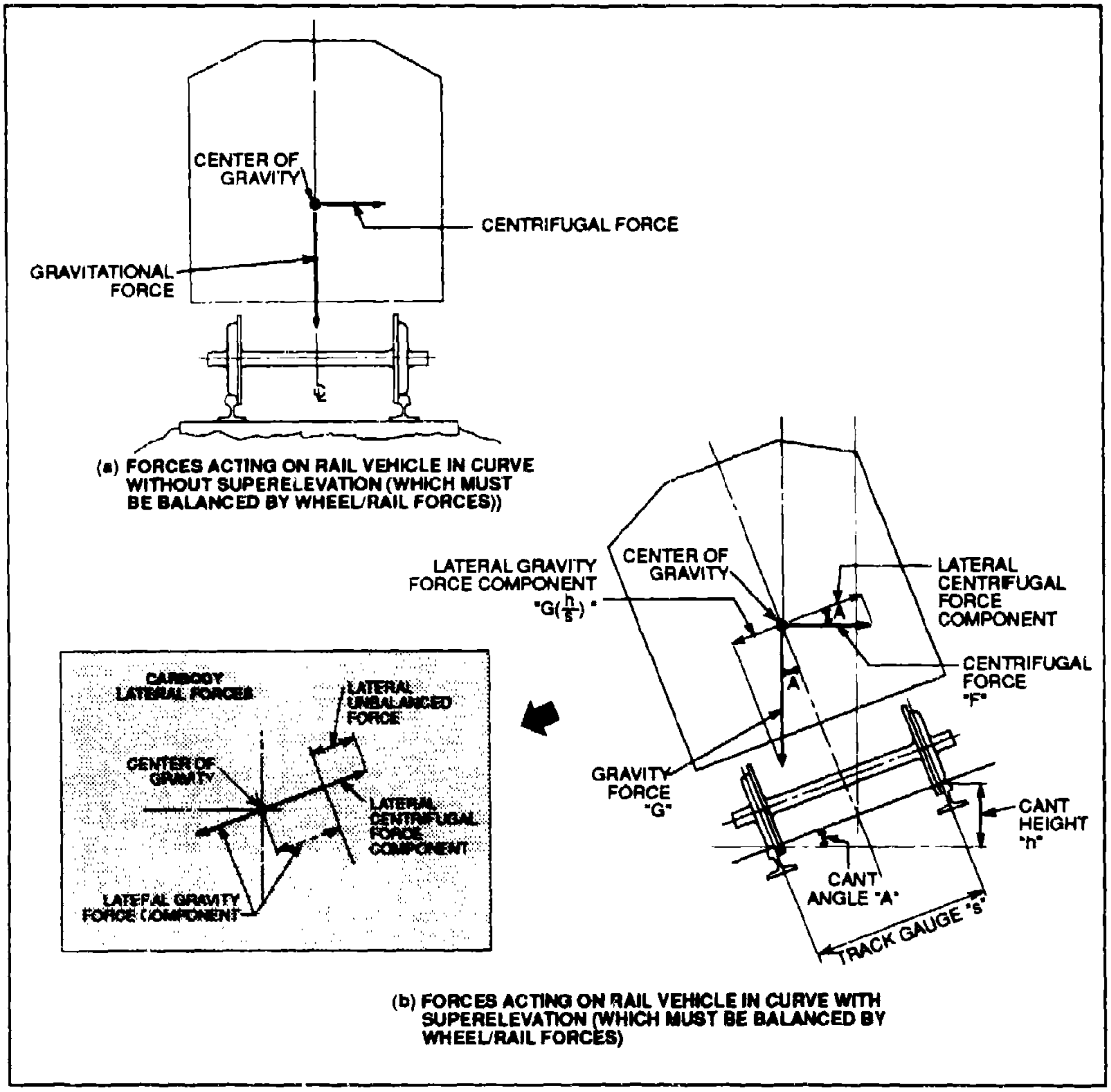


Figure 2.4: Accelerations and Forces Acting During Curving

amount of superelevation can be expressed in terms of either the difference in rail heights (in length units) or (as in a magiev guideway or the pavement of a highway) the size of the angle between the plane of the tops of the rails and the horizontal.

2.4 ACCELERATION COMPENSATION

Superelevation can reduce or eliminate the effect of centrifugal force on railway vehicle passengers by compensating this force with the lateral component of the gravitational force acting on the passenger, in the opposite direction to the perceived centrifugal force, as shown in **Figure 2.4 (b)**. By banking the track, the centrifugal force acting on the passengers is cancelled out, at least in part, by a component of the force of gravity.

Since the centrifugal force which a passenger perceives while traversing a given curve is a function of velocity, it follows from **Figure 2.4** that at some velocity, the lateral components of the centrifugal and gravitational forces acting on a passenger will exactly cancel one another.

In other words, for any given curve and track superelevation, there will be a single speed for which the lateral component of the centrifugal force will be exactly compensated by the corresponding component of the gravitational force.

This speed is referred to as the *balance* or equilibrium speed for a particular combination of curve radius, superelevation, and vehicle characteristics. For virtually all curves in railroad track, it is common practice to set the balance speed (and thus, the amount of superelevation or cant built into the curve) to accommodate the least stable freight car (in the U.S. and Canada, this might be a tri-level automobile carrier, which has a high center of gravity and large surface area susceptible to wind forces) under worst-case conditions (i.e., stopped on the curve with a strong cross-wind acting on the side of the vehicle on the outside of the curve).

The traversing of a curve at speeds either higher or lower than the balance speed results in an imbalance between the lateral component of gravity and the centrifugal force induced by operation through the curve, as shown in **Figure 2.4 (b).** It is common railroad practice to speak in terms of "cant deficiency or excess" or "inches of unbalance" when there is a difference between the actual operational speed through a curve and the balance speed of the curve.

If there were no premium on speed, the curve geometry could be set for the most demanding class of traffic and all trains would operate at that speed. Since speed is always at a premium for passenger service, and increasingly for freight as well, the curve geometry (and thus balance speed) becomes a compromise between the maximum that can be tolerated by the slowest, least stable trains and the minimum that can be accepted by the fastest trains. This means that the majority of trains may well operate at other than the balanced speed for a given curve, but always within the limits of the safety envelope for track forces and train stability.

Cant deficiency is defined as the difference between the actual superelevation (cant) in a given curve and the amount of superelevation which would be required to exactly balance the lateral (centrifugal) force acting on the train when it traverses the curve at a higher or lower speed. Cant deficiency is a particularly convenient measure of unbalanced speed operation for this context, insofar as it relates directly to the amount of vehicle carbody tilting which would be required to balance the forces acting on passengers and thus, maintain acceptable passenger comfort.

However, the geometry of the track and the speed of a vehicle or train are not the only elements affecting curving behavior and the effective angular inclination of a carbody. The situation can be complicated by the behavior of the vehicle suspension when operating above or below the balance speed. With some rail vehicle secondary suspension designs, the unbalanced lateral force acting on the vehicle at the center of gravity can tend to further tilt the vehicle in the direction of the unbalanced force by compressing the suspension springs on the outside of the curve. This would increase the magnitude of the unbalanced force, as shown in **Figure 2.5**. In this instance, the "softer" the vehicle suspension, the greater the amplification the unbalanced force would be, just as some automobiles will "roll" uncomfortably when making a turn at relatively high-speed. Many suspension systems, however, are designed to limit this effect using roll torsion bars or lateral links.

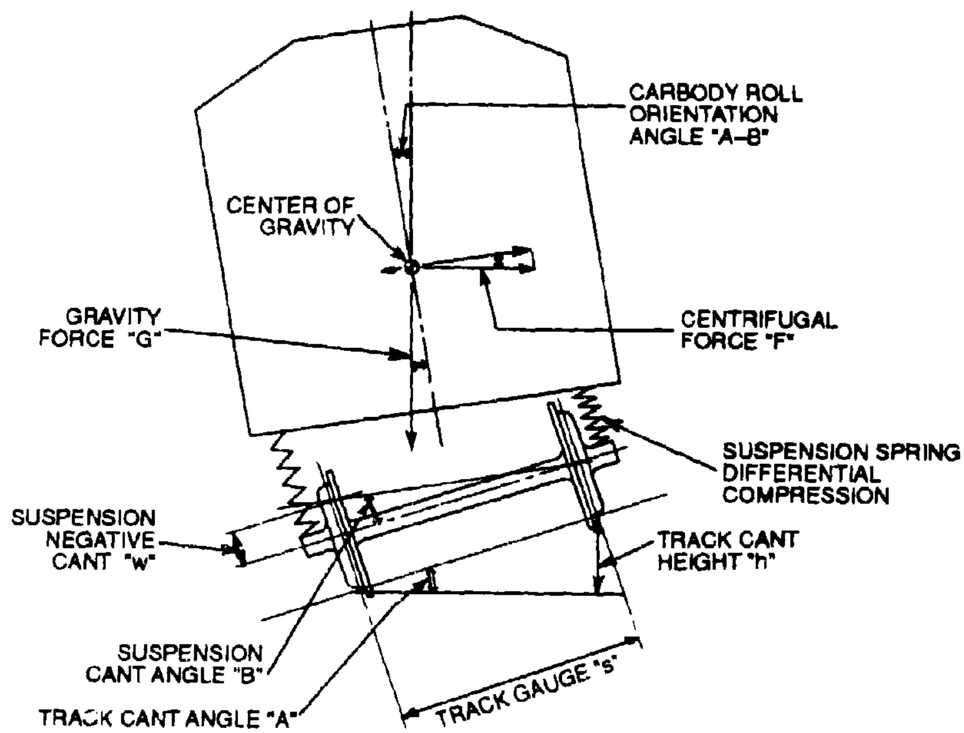


Figure 2.5: Effect of Suspension Compression on Forces Acting on Passengers During Curving

This effect is sufficiently important that it needs to be taken into account when designing or assessing the performance of vehicle tilting systems. As can be seen from **Figure 2.5**, the outward roll due to suspension compression for some suspension designs would have the same effect, with respect to passenger ride comfort, as reducing the superelevation in a curve by the amount of the differential compression (labelled "w" in **Figure 2.5**).

Finally, by intentionally tilting the body of a passenger rail vehicle, it is possible to reduce or eliminate the unbalanced lateral force acting on passengers, as shown in **Figure 2.6**. Intentional tilting affects passenger ride comfort as though the superelevation of the track in a curve was increased by the amount of deliberate banking, relative to horizontal.

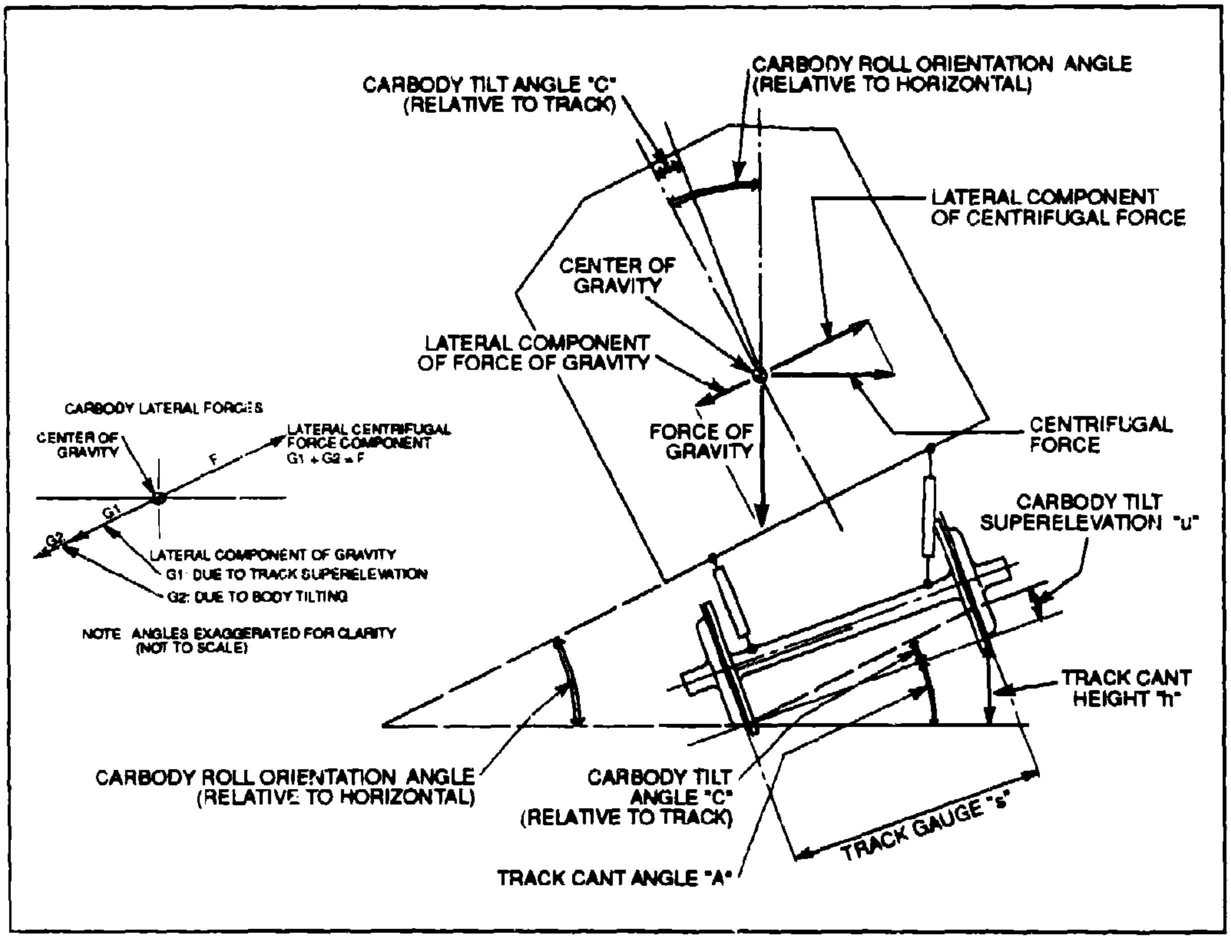


Figure 2.6: Effect of Deliberate Body Tilting on Forces Acting on Passengers

By incorporating the effects of differential suspension compression and deliberate body tilting into the expression for balance speed, a complete picture of the forces acting on rail vehicles, and, equally important, on passengers, is obtained. This allows passenger service operators to assess how tilt-body equipment would alter the time required for a particular trip. This information is essential in making an informed trade-off between the additional cost of acquiring and operating tilting equipment and the revenues to be gained from reduced trip time. Appendix A of this report provides a more detailed discussion of the physics of curve negotiation, including a step-by-step development of the complete unbalance force equation.

2.5 WHY TILT THE VEHICLE? WHY NOT CHANGE THE SUPERELEVATION?

The objective of tilting the body of a passenger rail vehicle while traversing a curve in the track at a speed above the *balance speed* (discussed above) is to achieve an acceptable ride quality with respect to the lateral force perceived by the passenger, without being forced to invest very large sums of money to build a dedicated passenger track with very large radius (very gentle) curves, or alternately to reconfigure the geometry of existing curved track to the point where safe freight operations would be compromised. By tilting the body of a passenger train vehicle, existing curves can be traversed at higher speeds without compromising passenger ride quality and without risking instability during freight operations should the track cant angle be increased.

However, it is important to distinguish that passenger ride quality is not *safety*! Tilting the carbody does not reduce forces at the level of the track; increasing speed increases the lateral centripetal force as the square of speed. Simply substituting tilting coaches (of equivalent axle load) for non-tilting coaches and increasing the curving speed without considering the effect of higher speed on the dynamic wheel/rail forces during curving, will result in a greater exposure to accident risk because the safety margin on curving forces will be reduced. This is the principal reason that tilt-body technologies with relatively high top speeds (above about 160 km/h (100mph)) also incorporate other features, such as low axle loads, low unsprung masses, steerable trucks, and/or active suspensions, to reduce track forces and improve or maintain operating safety margins, as well as body tilting to maintain passenger comfort.

Outside the Northeast Corridor (NEC), both the alignment geometry and track geometry of existing North American railway tracks have been modified over the years[8] to meet the requirements of current freight operations. This means that the balance speed and degree of superelevation in a given curve will be appropriate for relatively slow (40 to 60 mph) freight trains made up of vehicles with relatively high centers of gravity (compared to modern passenger equipment). At best, where freight and passenger operations share track, superelevation may be increased slightly above the ideal level for freight.

However, safety considerations arising from freight vehicle instability under certain conditions, and also the increased forces imposed on the lower (inside) rail in a curve by the much heavier freight cars and locomotives when traversing the curve at speeds below the balance speed, force any track geometry compromise towards the freight optimum. Imposition of heavier forces on the lower rail increases the risk of rail failure through fracture or overturning, and thus of derailment, and also causes greater rail (and wheel) wear, and thus increased maintenance costs.

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[8] Many existing railroads were originally built for (relatively) high-speed passsenger operations (160 km/h, 100 mph); these tracks were also able to accommodate the shorter, lighter-weight freight trains of the time. As freight was emphasized and train length and car weight increased, the geometry (superelevation, rate of change in superelevation) was reoptimized for freight operations at the expense of passenger operations.

Outside North America, the emphasis tends to be on railroads as passenger carriers, rather than as movers of freight. Freight cars are limited to a 22 tonne (24.2 ton) axle load, and trains are shorter, lighter, and often much faster.[9] However, in Japan and in many European countries, there are extensive mountainous areas served by secondary and even main lines with curvature that limits achievable speed below the safety limit, due to passenger comfort considerations. Even where purpose-built high-speed lines exist to remove this comfort restriction (e.g., the *Direttisma* in a mountainous region of Italy), there may be advantages to tilting technologies if train service extends through the rest of the national network or onto international routes where non-purpose-built track may be used. In addition, the emphasis on passenger operations, environmental concerns, and stringent approvals processes, especially in European countries, provide an on-going incentive to seek service improvement opportunities that are not limited to extensive new infrastructure development.

In essence, tilt-body technologies represent a potentially effective approach for improving achievable service speed for passenger equipment on existing tracks, without altering the geometry of curves and thus affecting the cost and safe operation of freight equipment, and without requiring very expensive investment in new dedicated high-speed infrastructure. For lines where passenger traffic density (and thus potential revenue) is low, this *equipment-oriented* strategy offers a cost-effective means for significant service improvement, and one that can be implemented incrementally, so as to ease the effect of financial limitations.

However, there is a fundamental conflict:

- o Body tilting can maintain *passenger comfort* through curves at higher unbalanced speed, but
- o Increasing unbalance will increase the lateral force exerted on track during curving, lessen the safety margin for curving, and could result in unsafe conditions.

The resolution of this conflict requires careful, systematic assessment and mitigation of both passenger comfort and track force effects of higher curving speeds.

2.6 THE TRADE-OFFS OF TILTING TRAINSETS

The intentional tilting of railroad passenger carbodies has the advantage of allowing a significant increase in the speeds at which existing track curves can be traversed, relative to those for non-tilting vehicles, with an equivalent level of passenger comfort. There are very substantial financial benefits that can arise from achieving higher average speeds (and thus, reduced trip

[9] German Federal Railways now operates some freight services at 160 km/h (100 mph).

times) on existing tracks, insofar as the required investment required for tilt-body vehicles is quite modest compared to that needed for infrastructure improvements.

Clearly, the magnitude of such benefits will be very much a function of the number and total degrees of curvature on any given route. Higher average speed achieved through reduction or elimination of speed restrictions on curves that have been imposed for reasons of passenger comfort may permit improved equipment utilization with correspondingly reduced requirements for capital investment in equipment, and should result in increased passenger ridership in response to the reduced travelling time.

The introduction of body tilting alone will not affect speed limits imposed for reasons of safety (i.e., to ensure that track forces and especially the lateral force exerted during curving does not exceed acceptable limits). The Swedish X2000, for example, has a maximum axle load of 17.6 tonnes (19.3 tons), with frame-hung traction motors to reduce unsprung mass, and radial-steering trucks, all of which combine to help keep track forces within acceptable limits even with a substantial increase in speed.[10]

The design of intentional-tilting passenger carbodies must address the issues of:

- o Potentially displacing the center of gravity laterally, as a result of the carbody tilting action, and decreasing the vehicle overturn safety margin,
- o With increased speed, increasing the lateral forces imposed on the track by the wheelsets with the attendant potential problems of increased rail wear, rail gauge widening, and in the extreme, derailment through rail rollover or rail overturning,
- o Potentially heightening passenger awareness of the dynamic response of cars to track irregularities through the sensitivity of the tilt control mechanisms, and
- o Increasing vehicle complexity and requirements for redundancy in design and maintenance effort to achieve reliable and safe operation.

However, as noted above, these issues can be controlled through careful design, implementation and maintenance, and so typically do not represent sufficient disincentive to offset the potentially substantial economic advantages of operating at higher speeds through curving track without major investment in or alterations to existing infrastructure.

The design considerations of tilting trains that tend to minimize the issues of tilting include:

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[10] "Tilting Train is SJ's Survival Tool," International Railway Journal, April 1990, pp 37-40.

- Use of lightweight carbody structures and truck (bogie) components (to minimize the static and dynamic loading of the track as induced by any change in the center of gravity during tilting and by the motion of the trucks and the carbody while running at speed),

- Reduction of the truck wheelbase and/or use of steerable trucks (to minimize the lateral forces imposed on the track by flange contact as the wheelsets move through a curve),

- Design of the vehicle to ensure that the roll center for the carbody during tilting remains close to or coincident with the center of gravity of the carbody (to minimize effects on the safety margin for vehicle overturn),

- Minimization of the mass of the truck components located below the primary suspension (to minimize the dynamic forces imposed on the track), and

- Use of hardware/control elements and appropriate control algorithms/ mechanisms to control the dynamic response of the tilt actuation operation.

Many of these design features have been incorporated into the various tilting technologies which are currently in revenue service or in production, or which have successfully completed development and feasibility testing. The key features of these technologies are summarized in Section 3 of this report; detailed technical information is presented in Appendices B and C.

2.7 ACHIEVING DELIBERATE BODY TILT

There are two basic approaches to deliberately tilting the body of a rail passenger vehicle. Passive-tilting designs utilize the lateral centrifugal force developed in a curve to tilt the body, while active-tilting designs employ actively-controlled components to force the body tilt.

2.7.1 Passive-Tilting

Passive-tilting is based on the pendulum effect provided by centrifugal and gravity forces when the carbody roll center is located well above the center of gravity. In effect, the carbody behaves as though suspended from pivots located at or near the top of the car, so that the body can swing laterally about its long axis, as shown in **Figure 2.7.**

Passive-tilting technologies have the advantage of technical simplicity and lower weight for the tilting components, but a high roll center means a potential reduction in the margin of safety for vehicle overturning. Technologies based on this principle include the Spanish Talgo *Pendular* the JR Series 381 Electric Multiple-Unit trainset (EMU), and the United Aircraft (UAC)

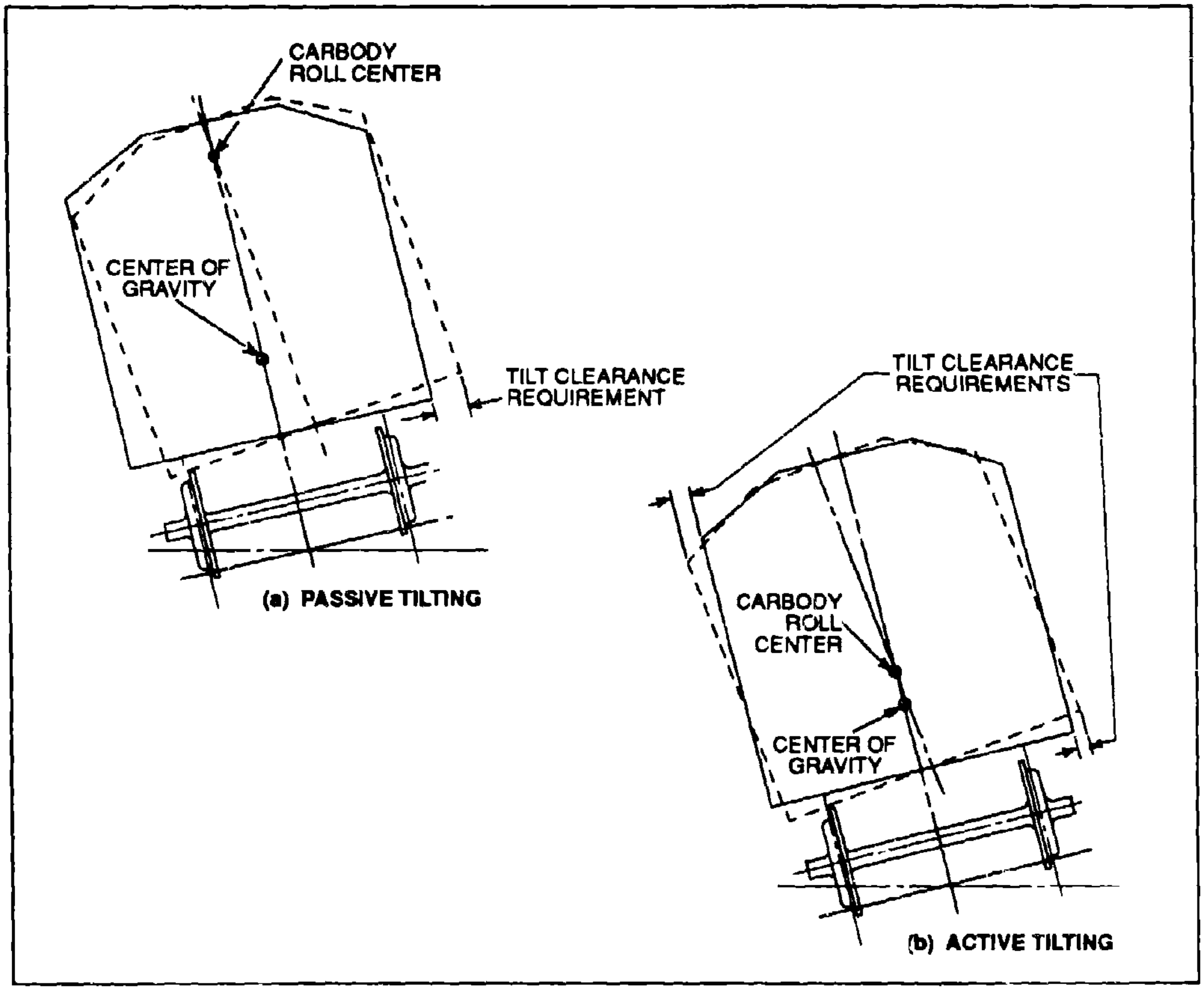

Figure 2.7: Location of Carbody Roll Center and Center of Gravity for Passive and Active Body Tilting

Turbotrain. The latter, since retired, was used by Amtrak in the early 1970s and by CN and VIA Rail Canada in the 1970s and early 1980s.

The Swiss consortium SIG has developed a truck-based passive-tilt mechanism known as *Neiko* for use with their unpowered high-speed truck; the truck can also be equipped with forced radial steering. The potential increased risk of overturning, at least in curves, can be offset by designing the vehicle to have a very low center of gravity, thereby lowering the lateral inertial overturning moment (the vehicle roll moment induced by the lateral inertial force acting at the center of gravity (c.g.) and reacted at the rail is a function of the height of tne c.g. above the rail). The Spanish Talgo passive tilting coaches (**Figure 2.8**) are notable in this regard. The lightweight carbodies are carried between the bogies, rather than on top of them (as described in Appendix C) so the c.g. is very close to the track.

Figure 2.8: The Talgo Passive-Tilt Trainset

In contrast, the other operational passive tilting technology, the Japanese Railways-Shokaku Series 381 electric multiple-unit trainset (**Figure 2.9**), has the carbody located on top of the trucks, so that the c.g. is relatively high. The effect of this high c.g. on overturn safety margin is exacerbated by the fact that this equipment operates on narrow-gauge (1 meter) track. However, these trainsets operate at relatively low-speeds (less than 120 km/h [75 mph]), and there certainly do not appear to have been any serious incidents during its 18 years of service.

2.7.2 Active-Tilting

The other technique uses hydraulic, electromechanical or pneumatic actuators in combination with a tilt control system to provide *active* body tilting. Active-tilt mechanisms incorporate mechanical linkages to keep the carbody roll center close to or below the carbody c.g., as in **Figure 2.7 (b)**. Doing so effectively eliminates any adverse effect on the safety margin for vehicle overturning, and has the additional practical advantage of minimizing the clearance envelope for the vehicle at maximum tilt, as shown in **Figure 2.7**. This approach also reduces the force exerted on passengers during tilting, in that the c.g. typically tends to be near to the passenger seat cushion level.

The principal disadvantages of active tilt mechanisms stem from the complexity and added weight of the tilt actuators and the difficulty in defining optimum (desirable) control strategies,

(Source: Japanese Railway Engineering, Dec. 1986)

Figure 2.9: The JR-Shokaku Series 381 Passive-Tilting Narrow-Gauge EMU

given the nature of the track geometry and passenger comfort. An inability to achieve reliable detection of curve onset and exit and acceptable timing of tilt actuation led to the cancellation of the British Rail Advanced Passenger Train (APT). However, some of the problem stemmed from the inadequate data processing capability available during the late 1970s and early 1980s. This would presumably no longer be a constraint, but the APT prototypes have long since been scrapped.

The MLW/Bombardier LRC coaches operated by VIA Rail Canada have also been affected by problems with curve detection and reliability of tilt operation, especially during the first half-decade of operation. Bombardier redesigned the control system during 1986-1988 and retrofitted the VIA fleet (**Figure 2.10**). The equipment tested on the NEC as part of the CONEG Task Force Program had been so modified (Section 4.4). As a consequence of this aggressive program and the extensive training of operating and maintenance personnel, VIA now employs active tilting on the Ontario-Quebec corridor.[11] Despite the problems encountered by some technologies, the successful Fiat ETR 450 EMU (Figure **2.11**), ABB X2000 (**Figure 2.12**), and LRC show that these challenges can be overcome.

[11] "Banking Performance Curves for the LRC Car Fleet," personal correspondence, R. Monette, Maintenance Operations, VIA Rail Canada Inc., 1992.

Figure 2.10: The LRC as Produced for VIA Rail Canada, Inc.

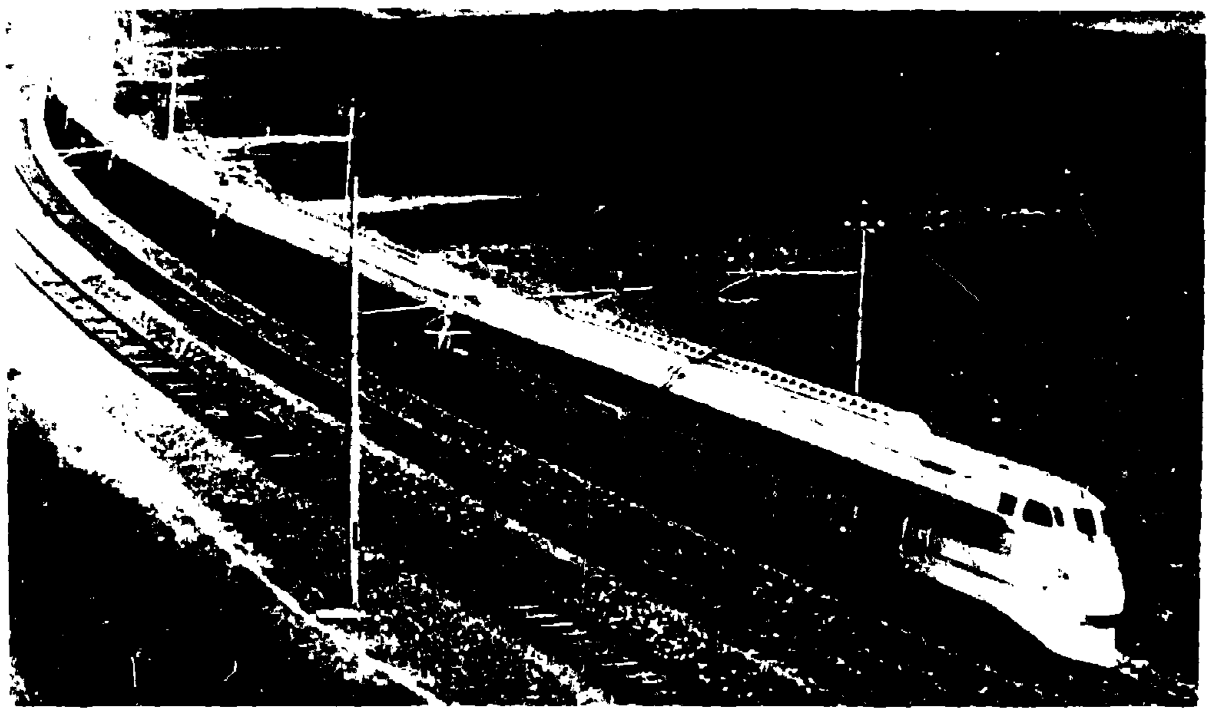

Figure 2.11: The Fiat Ferroviara ETR 450 Active-Tilt EMU Trainset

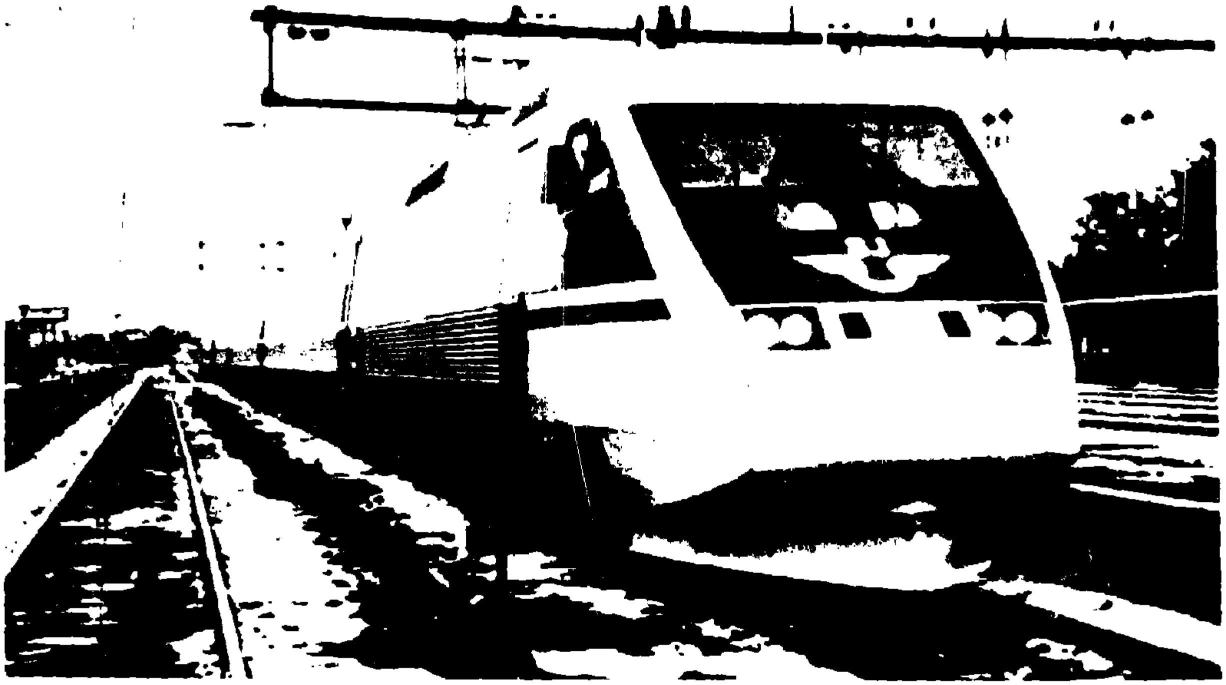

Figure 2.12: The ABB X2000 Active-Tilting Trainset

2.8 CURVE DETECTION AND TILT ACTUATION

While body tilting can maintain ride quality at higher speeds in curves, it is essential that the amount of body tilt, and the rate at which tilt is increased, closely match the increase in lateral acceleration (force) that arises as the vehicle moves from tangent track onto the run-in spiral and then onto the section of track with a uniform radius of curvature.

Similarly, the tilted carbody must be returned to its normal position as the vehicle moves over the run-out transition spiral.

This careful control of both magnitude and rate of tilting requires reliable detection of the onset of a change in track curvature. However, the mechanism must not be so sensitive as to overreact to irregularities in track geometry.

The curve detection and tilt control mechanisms incorporated in the technologies considered in this report depend on one or more of the following techniques:

- o Continuous measurement of lateral acceleration of the vehicle,

- o Continuous measurement of the carbody roll angle relative to the plane of the truck (bogie),
- o Continuous measurement of track superelevation, and
- o Continuous monitoring of vehicle location on the track relative to the known location of each transition and curve on the route.

Lateral acceleration on the vehicle is detected by accelerometers mounted on the carbody and/or the trucks. All but one of the actively-tilting technologies summarized in Section 3 and detailed in Appendix C depend on measurement of lateral acceleration. Suitable acceleration sensors are commercially available.

A number of the active-tilt technologies also measure the angle of the carbody relative to that of the truck. This measurement requires sensors that detect the difference in the position of the two sides of the carbody. Such sensors (typically differential transformers) are also commercially available.

Measurement of track superelevation forms part of the basis for curve detection and tilt control on the Fiat ETR 450 active-tilt equipment. Gyroscopes mounted on a truck of the vehicle at either end of the train provide an absolute horizontal reference against which the roll position of the truck can be measured. This information, which is instantly available, is used to supplement the lateral acceleration data, which tend to lag slightly behind the onset of curving due to the filtering of the acceleration signal to remove the effects of noise caused by random variations in track geometry.

The three curve detection techniques summarized above depend on measurement of acceleration, carbody and/or truck positions, and, as such, are generalized techniques that allow a vehicle to operate over any route. In contrast, the final technique listed above depends on access to complete information about the exact absolute location and geometry of each transition spiral and constant-radius curve on the line over which the vehicle is operating and a mechanism, such as wayside transponders, that allows detection of the exact position of the vehicle with respect to the next transition.

This technique, which was developed as part of a retrofit package for the Series 381 EMU and has since been used in the TSE-2000 DMU equipment for JR-Shikoku, is essentially a programmable control system that causes the vehicle to "follow" the lateral track geometry, banking the correct amount in the correct direction at the proper rate based on vehicle speed, just as the wheel-rail forces cause the vehicle to follow the longitudinal and vertical alignment geometry.

This approach offers several advantages in terms of overall simplicity and reliability, and in its avoidance of real-time (reactive) curve detection, the practicality of which is strongly affected by train speed. This technique could be of value to maglev, should tilt be required to adapt to the geometric constraints of existing rights-of-way.

Another important consideration in the design of active-tilt controls is the location of the sensor mechanism(s) that provide the input data to trigger the onset of tilting. Basically, there are two alternative sensor locations that are used in conjunction with the generalized techniques, as shown schematically in **Figure 2.13**:

- o On the car or vehicle immediately *ahead* of a given car, or
- o On the trucks of a given car.

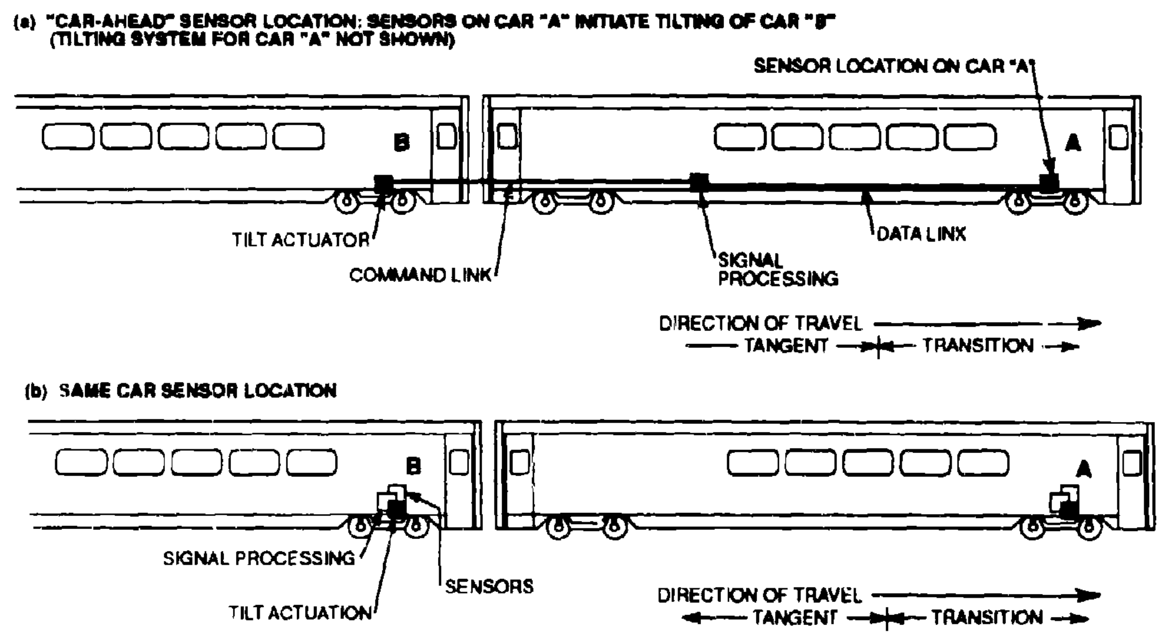

Figure 2.13: Alternative Sensor Locations for Active Body Tilt Control Systems

The "car-ahead" sensor location allows sufficient time for the control system to process the input data and "anticipate" the onset of curving, so that tilting of the vehicle can be timed to coincide with the onset of lateral acceleration. This provides superior acceleration compensation provided both rate and magnitude of tilt correspond exactly to the changes in track superelevation and the radius of curvature at each point in the transition.

This approach also permits detection of entry to and exit from the constant-radius portion of the curve, so that tilting can be halted and reversed without apparent discontinuities. However, use of "car-ahead" sensors does impose the minor requirement for transmission of sensor data and/or tilt control signals between cars or vehicles in a train.

Location of the sensor array on a given car simplifies requirements for data and/or control transmission, but imposes a lag between detection of a curve and the onset of tilting. This lag makes it more difficult to match rate and magnitude of tilt so as to exactly cancel out lateral acceleration. Mismatches between body tilt and curve geometry may result in a higher level of passenger discomfort (in the form of acceleration peaks or acceleration reversals) than would traversing the curve without body tilting.

2.9 FAIL-SAFE AND FAULT-TOLERANT DESIGN

An important consideration in the design of either active or passive tilting mechanisms is the requirement that the mechanisms be fault-tolerant and ultimately, "fail-safe." Should some component fail, the system must continue to operate safely. In the event the mechanism does not operate properly, the carbody must return to its untilted (neutral) position, be automatically or manually locked in that position, and the vehicle speed in curves be restricted to that approved for conventional (non-tilting) equipment. Each step is important, insofar as the vehicle requires minimum clearance when untilted, passenger comfort and safety would be adversely affected if the carbody were allowed to swing freely, and the ride quality would exceed acceptable limits if curves were taken at the higher speed used with a functioning tilt mechanism. It is clear from review of the technical literature that each manufacturer has considered this requirement.

2.10 RELIABILITY AND MAINTAINABILITY

As noted above, passive and especially active tilt mechanisms and the features that reduce or control track forces, add complexity to the design of passenger rail vehicles. This added complexity translates into a greater potential for failure with consequent additional requirements for maintenance, relative to a conventional (non-tilting) vehicle. Suppliers of tilt-body equipment have gone to considerable effort to ensure that their designs are as reliable as possible and also to facilitate the additional maintenance activities that are required.

In terms of reliability enhancement, tilt-train designs emphasize fault-tolerant subsystems with redundancy of critical components and sophisticated self-test and diagnostic capabilities. This strategy minimizes disruption of revenue service and facilitiates subsequent maintenance activities, but demands an aggressive and disciplined preventive maintenance program.

Fault-tolerant design for critical components and subsystems differs somewhat from, although does not obviate, the traditional "fail-safe" standards of the U.S. railroad industry; reconciliation of these two approaches is already occurring in some areas, notably train control, signaling and interlocking devices, but this process may need to be expanded to deal with the key design elements for tilt technologies.

To enhance maintainability, there is an emphasis on programmed preventive maintenance in purpose-designed facilities, and much effort has been made to ensure ease of access to important subsystems and components, and the modularization of major components and subsystems to permit rapid interchange, so that repair or servicing need not immobilize a vehicle or complete trainset.

One must also bear in mind that the entire philosophy of vehicle (and fixed facility) maintenance in Europe and Japan differs from the traditional reactive mode that has prevailed in the United States and Canada. Rigorous preventive maintenance programs that more closely resemble aviation practices are the rule elsewhere and the relatively modest maintenance increments associated with foreign tilt-body technologies reflect these systematic differences. Technologies incorporating hydraulic actuators and extensive microprocessor controls may require special training on the part of the operators in the U.S.

The same issues of maintainability and required skills pertain to steerable trucks (especially those that must balance stability at very high-speeds with superior curving performance), frame-hung or body-hung traction motors with cardan-shaft or quill drives, and indeed even to the lighter (on a kW for kW basis) and more rugged ac induction motors that are featured on some technologies and are likely to become the standard for the future.

2.11 COST VERSUS PERFORMANCE: HOW TILTING AFFECTS THIS FUNDAMENTAL TRADEOFF

Tilt-body technologies have the capability to offer improvements to trip time on routes with frequent curves of sufficiently small radius to warrant the imposition of speed restrictions for reasons of passenger comfort. (If speed restrictions are imposed because of other reasons, such as excessive wheel/rail forces, other design modifications such as the use of radial-type trucks must also be incorporated). Improvements which are achieved by raising the *average* speed on a particular route through reduction or elimination of deceleration/acceleration cycles on some curves may be significant, in terms of enabling the service operator to offer a more competitive transportation product. However, the effect on service competitiveness and ultimately on ridership and revenue is very dependent on the characteristics of each specific market.

The major potential benefit from body tilting is higher average speed without major investment in infrastructure. Tilt-body equipment may, under the right conditions, offer a much more cost-effective way to improve performance using existing rights-of-way and tracks.

Tilt-body technologies will permit speed increases in curves only to the extent that existing speed limits are imposed for reasons of passenger comfort. The use of body tilting does not alter the acceptable levels of lateral and vertical force that can be imposed on the track structure during curving, so that the effects of operating at a higher speed must be assessed for safety on a curve-by-curve basis. The most important element in controlling the magnitude of the forces imposed on the track structure at any given speed is minimization of the weight of the rail vehicle and especially what is termed its *unsprung mass* the portion of vehicle weight that is located between the track and the first set of springs (primary suspension) in the vehicle suspension, as illustrated in **Figure 2.14**.

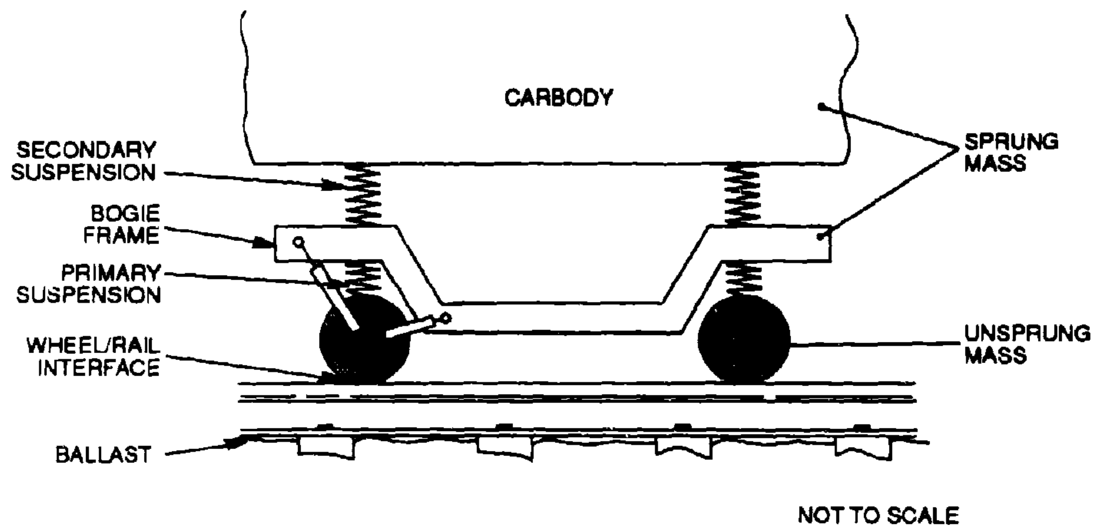

Figure 2.14: Vehicle Suspension Configuration

In North American locomotives, the unsprung mass comprises the wheelsets and axle-mounted traction motors. European and Japanese designs typically suspend the traction motor from the carbody or mount it on the truck frame above the primary suspension, with power being delivered to the wheels though a flexible driveshaft. This greatly reduces the unsprung mass (and also moves the traction motor out of a very dirty and demanding operating environment.)

Table 2.2 summarizes unsprung mass and axle load for some typical North American and foreign locomotives. As an illustration, the unsprung mass of the Bombardier LRC locomotive is over 3990 kg (8,800 lbs), similar to that of a typical four-axle freight locomotive; that of the diesel-powered HST used by British Rail is less than 2000 kg (4,400 lbs). The axle load and unsprung mass of the X2000 power car (locomotive) which draws electric power from overhead catenary is even lower, and the X2000 power car has trucks equipped with radial steering. None of these locomotives tilt.

TABLE 2.2
COMPARISON OF MAXIMUM AXLE LOAD AND UNSPRUNG MASS

Technology	Propulsion Type	Static Axle Load Tonnes (Tons)/Axle	Unsprung Mass Tonnes (Tons)/Axle
LRC	Medium-Speed Diesel	28.5 (31.4)	4.0 (4.4)
X2000	Overhead Electric	15.0 (16.5)	1.9 (2.1)
ETR 450	Overhead Electric	12.5 (13.8)	1.5 (1.6)
F40PH	Medium-Speed Diesel	29.0 (32.0)	3.6 (4.0)
HST	High-Speed Diesel	17.5 (19.3)	2.2 (2.4)

The ETR-450 electric multiple-unit vehicles, which also draw power from overhead catenary, have smaller, lighter traction motors mounted on the body structure of each car; the axle load and unsprung mass are even lower. All ETR-450 vehicles tilt.

Adherence to the U.S. standards for carbody strength (CFR Title 49 Part 229.141) instead of those specified by UIC Code 566 may affect both the axle load and unsprung mass of the vehicle[12]. The wheels and axles are sized in proportion to the mass which must be carried, so that as the static mass of the locomotive increases, as it must to provide the additional compressive strength in a cost-effective fashion, the unsprung mass must also increase (the static axle load of the LRC, which does meet U.S. standards, is 28.5 tonnes [31.4 tons]; that of the HST just 17.5 tonnes [19.3 tons]).

The bottom line from the foregoing is quite simple: an ability to maintain passenger comfort at higher speeds through curves by means of carbody tilting may be irrelevant if the track forces imposed by locomotives, compatible with U.S. standards, prevent safe operation at higher-speed.

[12] For an excellent overview of this and other differences in standards, see An Assessment of High-Speed Rail Safety Issues and Research Needs, Bing, Alan J., prepared by A.D. Little, Inc. for the FRA Office of Research and Development, Report DOT/FRA/ORD-90/04 (NTIS: PB 92-129212), December 1990.

The second consideration in the cost versus performance tradeoff is that while all the tilt technologies summarized in Section 3 are advertised as being suitable for use on existing track, they can only be applied if geometry and track structure are compatible with high-speed operation. Even on secondary lines in Europe and Japan, the basic track structure is quite different from that typically found on the (predominantly freight) railroads of the U.S.

Figure 2.15 shows the components that dominate typical railway track in the U.S. and in Europe or Japan. The major differences are in weight of rail (heavier in the U.S.), the type of fasteners (generally cut spikes in the U.S., elastic clips in Europe and Japan) and the type of ties (generally hardwood in the U.S. and on some secondary lines outside the U.S., concrete on European and Japanese mainlines and on some secondary lines). One exception is the NEC which has a high proportion of concrete ties and elastic fasteners. Outside the Northeast Corridor, even track maintained to the highest FRA Class 6 standard offers a rather different operating environment than the tracks for which foreign technologies were designed.

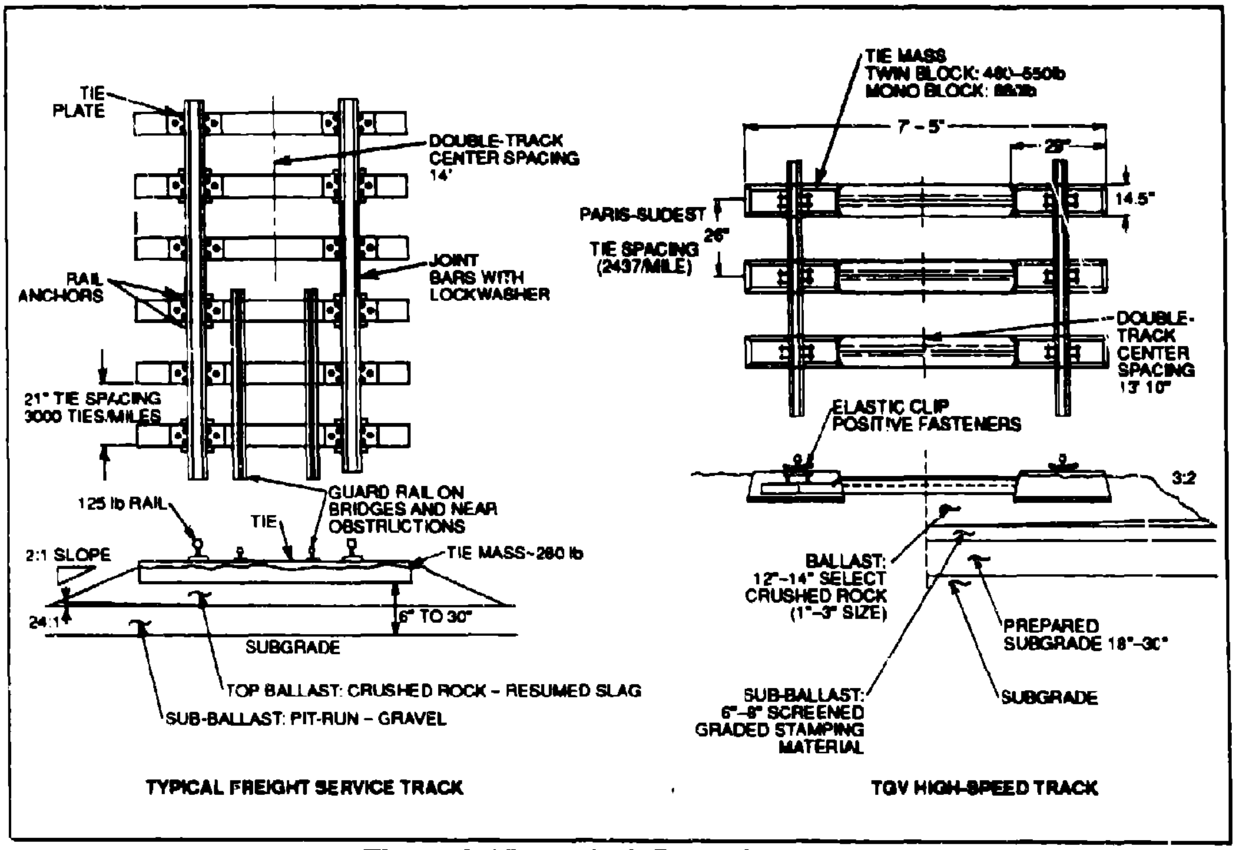

Figure 2.15: Typical Track Structures

2.12 SAFETY CONSIDERATIONS: WHAT TILTING DOES AND DOES NOT AFFECT

As noted above, body tilting is a technical solution to the problem of maintaining acceptable ride quality while increasing speed through curves, without modifying the geometry of the curve. Body tilting does not improve the safety margin for operating through a given curve! In fact, depending on how well a given tilt design positions and moves the c.g., it is possible that use of a tilting technology could reduce the margin of safety, even at the same speed. Since the objective is to increase speed, the margin of safety with respect to imposed track forces could be decreased, unless the axle load and unsprung mass of the locomotive used to propel the tilting cars is reduced, so that the track forces remain unchanged. Finally, because tilting the carbodies may increase the amount of clearance needed to ensure that the tilted vehicles will not impinge on tunnels, bridges, buildings, or trains on adjacent tracks, some investment could be needed to provide the added space.

It is important to recognize that most tilt-body technologies incorporate other design features for high-speed operation, such as lower axle loads and reduced unsprung mass, steerable trucks to reduce lateral forces during curving, and improved traction and braking control that have the potential to improve safety relative to the conventional technologies in use in the United States. These features are noted in the detailed assessments of each technology reported in Appendix C of this report. The systematic application of preventative maintenance practices for both vehicles and infrastructure also contributes to enhancement of safe operations; emphasis on event avoidance, rather than on event survivability, has much to recommend.

Most operational tilt-body technologies are not aimed at very high-speed operation. The ETR 450, with a 250 km/h (156 mph) service maximum, is the fastest revenue-tilting train. This is at the upper limit of what might be termed the *intermediate-speed* in the context of proposed technologies. The X2000 has a 200 km/h (125 mph) design speed, and has begun operating at that speed on portions of selected routes in Sweden; it reached 250 km/h (156 mph) during running trials on German high-speed track. The production LRC has been limited to 155 km/h (95 mph) or less during its service with VIA Rail Canada, primarily because that is the maximum speed Canadian federal regulations permit on track with at-grade road crossings. A much lighter prototype locomotive and coach was operated in test at 200 km/h (125 mph) at Pueblo, and two trainsets leased to Amtrak operated at lower speed on segments of the Northeast Corridor between New York and Boston (this equipment was returned to Bombardier in July 1981 at the expiration of the lease period since Amtrak's limited budget would not allow purchase of the trains[13]).

[13] "Canada's LRC: Low cost, high speed," Railway Age, August 9, 1982.

In part, these relatively modest speeds reflect an inherent conflict between the characteristics of trucks capable of stable (safe) operation at very high speed on purpose-built track, and the characteristics of trucks designed to run on existing tracks. Simply put, high-speed trucks are very rigid to resist hunting; trucks for existing track must be quite flexible, even if not steerable. The advanced truck designs proposed for high-speed tilt trains like the Fiat "AVRIL" incorporate independent wheels, active lateral and vertical suspensions, and a variety of unusual propulsion configurations to help address the challenge posed by this divergence.

3. OVERVIEW OF TILT-BODY TECHNOLOGIES

Tables 3.1, 3.2, and 3.3 provide an overview of the tilt-body technologies examined in this report. **Table 3.1** summarizes technologies employing active body tilting that are either operational or under construction. For completeness, the ABB X2000 is included, although that technology was reviewed in some detail in the report recently completed for the FRA.[14]

Table 3.2 summarizes advanced active-tilt technologies at the conceptual design stage.

Table 3.3 summarizes passive-tilt equipment in service or under current development.

This report does not address two tilt-body technologies that are primarily of historical interest:

- The United Aircraft Turbotrain, equipped with passive tilting, which was operated with varying degrees of technical success and market acceptance both by Amtrak in the U.S. and by Canadian National Railways and VIA Rail Canada, in the late 1960's, 1970's, and early 1980's, but has since been replaced (the passive-tilt aspects of the Turbotrain, with low c.g., were favorable, but lack of information on any current development precluded further review); and

- The British Rail Advanced Passenger Train (APT), an electrified, 156 mph active-tilt articulated trainset which was developed and tested in prototype in the 1970's and early 1980's, but which failed to perform reliably and was subsequently scrapped.

[14] Safety Relevant Observations on the X2000 Tilting Train, prepared for the FRA Office of Research and Development, DOT/FRA/ORD-90/14 (NTIS: PB 91-129668), December 1990.

TABLE 3.1
ACTIVE TILT TECHNOLOGIES

STATUS: *In Service*	TECHNOLOGY TYPE	MAXIMUM SPEED	TILT CONTROL	TILT ACTUATION	DESIGN STANDARD	OPERATOR/ FLEET SIZE	SERVICE EXPERIENCE	FIRST USED	COMMENTS
LRC	Tilting Coaches; Non-tilting diesel locomotive	155 km/h (95 mph) service; 200 km/h (125 mph) design	Same-car accelerometer	Hydraulic; 10° maximum tilt	FRA/ AAR/ TC	VIA Rail Canada; 100 coaches, 32 locomotives	Tilting coaches in service; 10 locomotives in service	1981	Tilt extensively used, Ontario-Quebec corridor
ETR 450	Electric MU; all cars tilt	250 km/h (156 mph)	Lead car gyro and accelerometers	Electro-hydraulic; 10° max but limited to 8° for passenger comfort	UIC	Italian State Railways; 82 2-car powered units [illegible]	Quite successful; additional order placed 1991	1987	Forms backbone of FS high-speed services
X2000	Tilting coaches and driving trailer; non-tilting electric locomotive	200 km/h (125 mph) service; has reached 250 km/h (156 mph) in test	Lead vehicle (locomotive or DT) accelerometer; differential transformer on each tilt truck	Hydraulic; 8° maximum tilt, 6.5° effective tilt angle	UIC	Swedish State Railway; 20 1-5-DT sets in service or construction	Successful but limited by small (as yet) fleet	Sept 1990	Being tested in Germany and Switzerland; a candidate for U.S. testing
TSE-2000	Narrow-gauge 3-car diesel MU	120 km/h (75 mph)	Look-ahead controller based on geometric data file, wayside transponders	Pneumatic; 5° maximum tilt	UIC	JR-Shikoku; 38 3-car sets (Nov '91)	Successful; may see other applications	1989	Tilt controller could have potential for HSR, maglev in U.S., elsewhere
E7 Coach	Coach	200 km/h (125 mph) design; 160 km/h (100 mph) service	Locomotive-mounted gyro and accelerometers	Hydraulic; 7° maximum tilt	UIC	Norwegian State Railway; NSB standard coach	Successful, but with tilt locked out	1985	Coaches built post '86 have tilting trucks, but feature not used
STATUS: ***In Production***									
VT-610	Two-car diesel MU	160 km/h (100 mph)	Same as ETR-450, but without sequenced tilting	Same as ETR 450; 8° maximum tilt	UIC	German Federal Railway; 20 2-car units	Delivery of first unit due December 1991	1992?	An interesting export success for Fiat
Series 4012	3-car electric MU	200 km/h (125 mph)	Same as ETR-450	Same as above	UIC	Austrian Federal Railways; 3 6-car sets (2x3)	First delivery 1994	1994/ 1995?	AC induction motors and EM rail brakes

TABLE 3.2
ACTIVE TILT TECHNOLOGIES - CONCEPTUAL DESIGNS

TECHNOLOGY	TECHNOLOGY TYPE	MAXIMUM SPEED	TILT CONTROL	TILT ACTUATION	DESIGN STANDARD	PROGRAM SCHEDULE	OTHER FEATURES	COMMENTS
Fiat "AVRIL"	Electric MU, 4-car traction unit	320 km/h (200 mph)	Based on ETR-450 system	Not stated	UIC	Announced Nov 1990	Independent-wheel truck	Could be affected by FS decision to proceed with ETR-500 procurement
RTRI 250X	Narrow-gauge Articulated EMU	250 km/h (156 mph)	Based on TSE-2000 system ?	Not stated	JR/UIC	Announced April 1991	Independent-wheel single-axle truck with ac hub motors and active suspension	Will require use of advanced materials and very sophisticated on-board control system (an "intelligent train"); will have to be able to draw power from third rail and catenary

TABLE 3.3
PASSIVE TILT TECHNOLOGIES

STATUS: *In Service*	TECHNOLOGY TYPE	MAXIMUM SPEED	TILT LIMITS	DESIGN STANDARD	OPERATOR/ FLEET SIZE	SERVICE EXPERIENCE	FIRST USED	OTHER FEATURES/ COMMENTS
Talgo Pendular	Coach	200 km/h (125 mph) design, 180 km/h (112.5 mph) service	3.5°	UIC	RENFE; 428 cars as of 1989; additional cars under con-struction	Excellent; market acceptance and performance has led to expansion of services	1980	The backbone of RENFE international services; automated gauge change between Spain/ France; has been tested on NEC, in Germany, Austria; ran at 288 km/h (180 mph) on DB
JR Series 381	Narrow-gauge EMU	130 km/h (81 mph) design, 120 km/h (75 mph) service	5°	JR/UIC	JNR to 1987; JR-Shikoku thereafter	Still in service, but many problems with tilt nausea, braking	1973	Retrofit active-tilt package developed due to tilt nausea problem; did not achieve objectives for higher-speed; a dead end.

STATUS: *Under Development*	TECHNOLOGY TYPE	MAXIMUM SPEED	TILT LIMITS	DESIGN STANDARD	PROGRAM SCHEDULE	OTHER FEATURES	COMMENTS
Talgo 250	Coach	250 to 300 km/h (156-187.5 mph)	Unstated	UIC	Announced 1989; Prototype December 1991	Improved brakes, doors, windows, pressure sealing	Originally targetted 250 km/h (156 mph), but success of trial on DB high-speed line has upped objective; as of July 1990 working with Siemens and Krauss-Maffei (power cars) to produce complete train
SIG Truck with "Neiko"	Truck only	230 km/h (144 mph) design	about 3°	UIC	Truck design 1986; "Neiko" tilt mechanism 1990	Truck can also be equipped with "Navigator" forced radial steering	Tilt mechanism depends on inclined links between bolster and truck frame to create virtual tilt center above C_g, augmented by central airspring to reduce tilt inertia; most of effective tilt angle comes from elimination of differential suspension compression.

4. U.S. EXPERIENCE WITH TILT-BODY TECHNOLOGIES

Foreign designed and built tilt trains have been considered for possible application in the United States. Two such technologies, the Spanish Talgo *Pendular* and the Canadian LRC, have been tested in the U.S. with equipment provided by the developers, although they have not subsequently been used in revenue service.

4.1 EARLY EXPERIENCE WITH LRC

The original LRC technology, developed between 1967 and 1970, led to a first prototype train, which consisted of a 12-cylinder diesel-electric locomotive and one banking coach, in July 1971. This train was built to verify the feasibility of providing a safe high cant deficiency operation over existing infrastructure in North America with passenger comfort. Extensive testing was performed on the prototype train between 1971 and 1976 in Canada, at the U.S. Department of Transportation, Transportation Test Center near Pueblo, Colorado, and in the Northeast Corridor. These series of high-speed tests, which were performed at speeds up to 210 km/h (130 mph), verified many aspects of the train such as ride quality, the effectiveness of the tilting mechanism, vehicle stability and curving, and track loading. The tests demonstrated that a low center of gravity, low profile train such as the LRC, could safely negotiate curves at much higher speeds than were presently permitted in the U.S. and provide a reasonable ride comfort.

Two LRC trainsets were leased by Amtrak in 1980. This equipment consisted of two 16-cylinder diesel-electric locomotives and 10 banking coaches. In a joint FRA/Amtrak project, high-speed curving tests were carried out in the summer and fall of 1980 on the LRC locomotive, the LRC banking coach, the standard Amcoach, and the AEM-7 locomotive.[15] The vehicles were equipped with instrumented wheels, carbody accelerometers, and displacement transducers. In repetitive runs in the Northeast Corridor, the Amcoach was tested at up to 229 mm (9 in) of cant deficiency, and the LRC train was tested at up to 381 mm (15 in) of cant deficiency. Similar runs, up to a cant deficiency of 279 mm (11 in), were also performed on the AEM-7 locomotive at a test site on the Philadelphia-Harrisburg line equipped with the required electrification. In addition, the vehicles were run on a large sample of curves at high cant deficiency to investigate the transient performance of the vehicles over a wide range of typical perturbations.

4.2 SAFETY LIMITS

Safety considerations which were examined relating to operation at higher cant deficiencies included vehicle overturning, wheel climb, rail rollover, and track panel shift (see Section

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[15] High Cant Deficiency Testing of the LRC Train, the AEM-7 Locomotive, and the Amcoach, Report No. DOT-FR-81-06, (NTIS: PB 82-213018), January 1982.

5.1.1). It was found that the maximum safe cant deficiency limit of each train tested was set by the vehicle overturning safety criteria for the coach, and in particular by the steady state side-to-side weight transfer. The safety limit was set by the coaches rather than by the locomotives after making allowance for 90 km/h (56 mph) crosswind loading which is more restrictive for coaches. Results showed that, except for a few unusually harsh curves, the LRC train could run safely at up to 229 mm (9 in) of cant deficiency, while maintaining less than the recommended AAR comfort limit of 0.1g steady-state lateral acceleration by tilting the coaches, and that a conventional train consisting of the AEM-7 locomotive and Amcoaches would run safely at 203 mm (8 in) of cant deficiency at the expense of "strongly noticeable" (about 0.15g) steady-state lateral acceleration.

4.3 BANKING AMCOACH

In a second joint FRA/Amtrak project in 1982, tests were performed on the F40PH diesel-electric locomotive and an Amcoach modified for banking, with and without the banking system in operation.[16] The modified coach used a truck frame with softer primary suspension and an air-actuated torsion bar device, supported by bearings secured to the carbody, to tilt the body by overcoming the secondary airsprings. An electronic controller initiated the full four-degree available tilt when the damped lateral acceleration of the truck frame reached a threshold level of 0.04g. Safety at high cant deficiency was evaluated by comparing direct wheel/rail force measurements to safety criteria. Again, a general cant deficiency limit, imposed by the steady state overturning criterion, was found to be 203 mm (8 in) for both the banking Amcoach and the standard Amcoach. The general cant deficiency limit of the F40PH locomotive was found to be 229 mm (9 in), although several exceptions were identified by the transient overturning criteria. The banking system of the modified Amcoach was successful in maintaining a low level of steady state carbody lateral acceleration at high cant deficiency, although a recommendation was noted that fail-safe devices should be required to prevent one truck of a banking coach from operating while the other is disabled.

4.4 CONEG TESTS

During the spring and fall of 1988, Amtrak/FRA, working closely with the Coalition of Northeastern Governors (CONEG), conducted high-speed tests of tilt and turbo equipment in the Northeast Corridor between Boston and New York City.[17] These tests were performed to evaluate the feasibility of utilizing existing and proven technologies to achieve the CONEG objectives of reduced trip time and enhanced passenger comfort. These tests were also required to validate the train performance models used to predict running times for various equipment

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[16] High Cant Deficiency Test of the F40PH Locomotive and the Prototype Banking Amcoach, Report No. DOT-FR-83-03, (NTIS: PB 83-219139), January 1983.

[17] CONEG (Coalition of Northeastern Governors), Tilt and Turbo Train Test and Evaluation, January 1989.

options and configurations, as well as to assess the benefits of proposed fixed plant improvements. The route, particularly suited to a tilt technology assessment, was 367 km (228 miles) long, in which there were 238 curves; typically the percentage of track which had more than one degree of curvature was about 40%. The total length of these curves was more than 121 km (75 miles).

The equipment technologies tested were the Amfleet cars (currently in Northeast Corridor operation), the RTL and RTG turboliner trainsets, the Spanish Talgo *Pendular* passive tilting coaches and an LRC active tilt trainset. The Amfleet cars were tested to provide a baseline for comparison of the candidate equipment. All equipment types were instrumented to measure speed and carbody lateral acceleration, and were operated at higher speeds around curves than were currently permitted. The FRA required that sufficient instrumentation be installed on each trainset in order to relate test behavior to previously tested equipment known to be safe. The cant deficiency was limited by sensible considerations of passenger comfort and safe passenger mobility. Because of the frequent proximity of many curves, high cant deficiency speeds could not always be achieved due to the low acceleration capabilities of the locomotives. "The performance of all tests was verified in accordance with Congressional intent by the FRA and by consultants to the CONEG Policy Research Center Inc."[18]

Measurements were analyzed into steady state lateral acceleration, peak lateral acceleration, jolt (the maximum difference in trainset lateral acceleration within any one second interval) and absolute peak-to-peak lateral accelerations. Tests were conducted incrementally to attain maximum curve speed, permitting analysis of applied forces and dynamic responses during and at the conclusion of each test run, before proceeding to the next incremental level of cant deficiency.

Review of all test data disclosed that passenger trains could operate at higher cant deficiency speeds without compromising passenger comfort and derailment safety limits. The running time from Boston to New Haven could be reduced to 1 hour and 56 minutes, 24 minutes faster than trains operating at conventional 76 mm (3 in) cant deficiency speeds.

The trends of steady state and peak lateral accelerations and jolt averaged from 33 curves provided an overall comparison of the test vehicles; a comparison of the trend lines with increasing cant deficiency up to eight inches is shown in **Figure 4.1**.[19] The steady state lateral acceleration of the LRC was sustained near zero "g" by its active-tilt system. The Talgo showed a large reduction in steady state acceleration but its passive-tilt system did not completely cancel

[18] Amtrak Evaluation of Tilt and Turbo Train Technologies, Amtrak Report, 1989.

[19] Ibid., Amtrak Report.

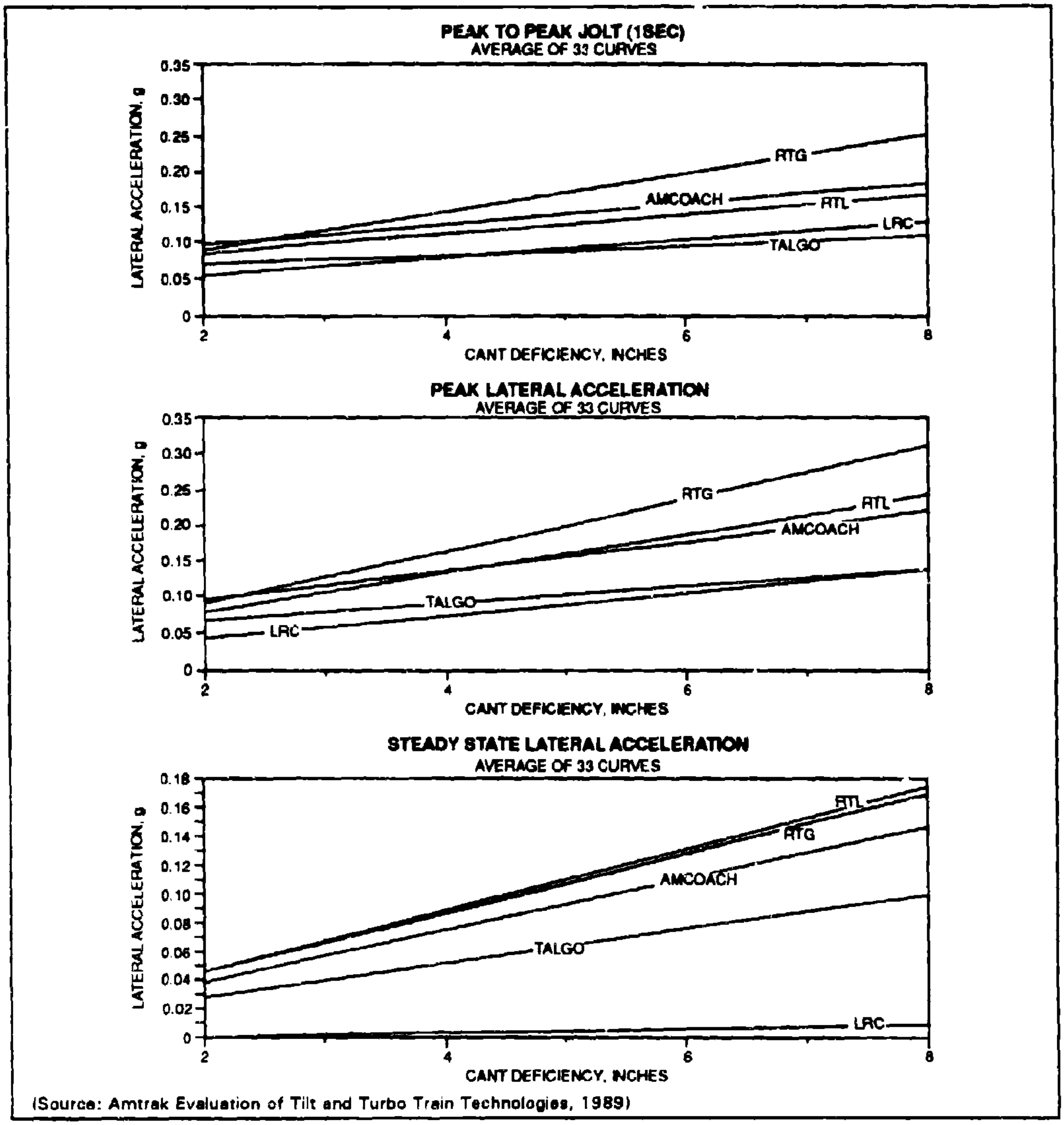

Figure 4.1: Comparison of Carbody Lateral Acceleration Trend Lines

these accelerations. Both the LRC and Talgo offered significant improvements over the baseline vehicle in dynamic performance and lower steady state accelerations.

The LRC exhibited a somewhat smaller peak lateral acceleration at low cant deficiencies but very little difference as cant deficiency increased. The peak-to-peak jolt of the Talgo was less sensitive to cant deficiency than the LRC. Both the LRC and Talgo handled jolts extremely well, although the LRC was superior only on long smooth curves. The LRC coach, with an active suspension tilting mechanism, exhibited a lateral acceleration, increasing in the entry spiral of a curve until the control system tilted the body to cancel the steady state lateral acceleration. However, as the car left the curve, the body remained tilted until the control system responded to remove it. This system lag produced a significant negative lateral acceleration at curve exit. During curve entry transition, the LRC was vulnerable to track perturbations which cause jolts. The Talgo kept the steady state lateral accelerations below 0.1g and its negative lateral acceleration at curve exits was usually insignificant. It was superior on short curves and rough-entry curves.

The steady state and peak lateral acceleration measurements were also used to monitor derailment safety during the test runs. The most restrictive of the derailment safety criteria is the vehicle overturning criterion which is formulated to prevent excessive side-to-side weight transfer in curving. The steady state and peak lateral accelerations were used to estimate the respective wheel load transfer using calculations based on known vehicle suspension characteristics and previous measurements of some vehicles with force sensing wheels. Truck, rather than body accelerations, were used to estimate wheel load transfer of the active suspension LRC coach because the tilt action eliminated the means of estimating steady state load transfer. All test vehicles were deemed to be within the safety limits up to a cant deficiency of 203 mm (8 in).

In parallel with the measurements, passenger evaluations of ride quality were obtained from a survey of volunteers recruited by CONEG to ride each of five train trips made to simulate revenue service.[20] The passengers riding at these higher curve speeds reported the occurrence and severity of each instance of discomfort, and provided subjective ratings of the entire trip. Generally, the results indicated passengers' acceptance of higher than normal curve speeds. Over 84% of the passengers in the test rated the ride quality of their runs at these higher curve speeds as either good or excellent. The average number of reports of curve-related discomfort per passenger over the course of the 251 km (156 mile) distance from Boston to New Haven was only 8.2.

From an analysis of the discomfort reports, the increase in discomfort was attributed to the increase in lateral acceleration forces (steady state and jolt) felt by passengers as curve speeds increased. The reports also showed that steady state and jolt acted together to exacerbate passenger discomfort. The reports on individual curves and ratings of overall ride quality indicated that tilt trains can make a difference. The tilt trains provided the most comfortable ride of the demonstration trains and produced quite acceptable levels of comfort even at the highest curve speeds tested (only about 7% of passengers expressed discomfort).

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[20] Passenger Evaluation of Tilt and Turbo Train Rides, Report to the FRA, April 3, 1989.

A larger percentage of passengers on the tilt trains rated the quality of their ride as "excellent" than did so on either the Amfleet baseline or Turbo trains. One significant finding of the survey was that most passengers accepted the practice of higher unbalance levels in train travel.

4.5 CONCLUSIONS FROM U.S. TILT-BODY TRAIN EXPERIENCE

The general conclusions of the U.S. experience can be summarized as follows:

- Speeds of passenger trainsets can be increased through curves to reduce trip time on existing guideways and still operate safely,
- Tilt-body vehicles offer the potential to maintain good passenger ride comfort in curves at the higher cant deficiencies, and still remain safe, and
- The practical limits to speed in curves will be safety-related, not passenger-comfort related, if tilt-body technology is used.

5. TILT OPERATIONS IN THE U.S.: WHAT ARE THE ISSUES?

There are several important issues - and potentially important opportunities - that would arise if U.S. railroads were to make use of vehicles equipped with tilt-body capabilities. These issues encompass a range of safety, technical compatibility, and regulatory compatibility considerations. Some of the issues and most of the opportunities arise from the consequences of body tilting itself, and would pertain even for designed-for-America equipment. An obvious example of an issue in this class is the effect of body tilting on compatibility with the clearance envelope for a given (existing) railroad or route. In terms of opportunities, active body tilting may permit co-location of high-speed rail or Maglev in some existing rights-of-way without unacceptable degradation of ride quality.

There are also important issues that exist because all but one (the LRC) of the existing tilting technologies have been designed and built for different sets of technical and safety standards and operating conditions than exist in the U.S. The issues in this class are the same in principle as those that affect non-tilting foreign technologies like the TGV or the ICE.

The most obvious example of this category is the difference between FRA structural strength standards and those of UIC Code 566. Treatment of these *generic* issues affecting technologies originating outside the U.S. is beyond the scope of this investigation, and the reader is directed to the recent FRA report[21] for a comprehensive overview. Buff strength, as an example, is a measure of occupant compartment structural integrity. This measure is adequate for a particular type of car construction (body-on-underframe) and for low-speeds, when train buckling is not a great concern. Different vehicle structural designs may allow increased occupant compartment structural integrity and decreased vehicle weight. The FRA currently is examining the issue of crashworthiness in a major study on *Collision Avoidance and Accident Survivability* scheduled for completion in 1992. Some of the generic issues - notably the example cited above - do bear directly on the tilt-specific issues, and are discussed below.

5.1 BODY TILTING ISSUES

There are five issues that must be addressed prior to the use of tilting rolling stock in the U.S., even if all the generic issues related to use of equipment built to non-U.S. standards are resolved. These issues are:

[21] An Assessment of High-Speed Rail Safety Issues and Research Needs, A.J. Bing, prepared by A.D. Little, Inc. for the FRA Office of Research and Development, Report No. DOT/FRA/ORD-90/04 (NTIS: PB 92-129212), December 1990.

- o Effects of increased curving speed on operating safety, including "worst case" scenarios;
- o Compatibility with clearance envelopes for existing tracks and equipment types;
- o Maintainability and reliability, including availability of appropriate facilities and labor;
- o Effects of U.S. alignment geometry and track maintenance standards on effectiveness of foreign tilt mechanisms in maintaining passenger comfort; and
- o The incremental costs and benefits of tilting.

5.1.1 Increased Curving Speed and Operating Safety[22]

The fundamental basis for safe curving at higher speed is satisfactory control of forces acting at and across the wheel-rail interface. Existing FRA regulations (49 CFR Part 213) specify track geometry deviations for various speed regimes and a maximum allowable cant deficiency of 76 mm (3 inch). The FRA regulations do not directly address track-train forces, lateral/vertical force ratios, or allowable maximum lateral and vertical static or dynamic loads. Industry standards and practices also do not address these areas.

The criteria applied to determine whether a rail vehicle can safely negotiate a curve at a given speed differ from jurisdiction to jurisdiction internationally. All are concerned with assessing the risk of vehicle derailment through four basic mechanisms:

- o Vehicle overturning,
- o Wheel climb,
- o Gage widening (rail rollover, rail lateral deflection), and
- o Lateral track panel shift.

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[22] Much of the material in this section is drawn from Chapter 5 and Appendix B of the 1983 Report DOT-FR-83-03, (NTIS: PB 83-219139) entitled High Cant Deficiency Test of the F40PH Locomotive and the Prototype Banking Amcoach, prepared by ENSCO, Inc., for Amtrak and the FRA Office of Freight and Passenger Systems. Appendix B of this reference is based on the work of Battelle Columbus Laboratory carried out as part of the IPEEP Program, and reported by Dean and Ahlbeck "Criteria for the Qualification of Rail Vehicles for High-Speed Curving," IPEEP Working Paper, Oct. 1977; and "Criteria for High-Speed Curving of Rail Vehicles", ASME Paper No. 79-WA/RT-12, December 1979.

These criteria are basically reference standards against which experimentally-measured wheel force values are assessed, taking into account the effects of wind loading as well as the forces generated by curve negotiation. **Table 5.1** summarizes the safety criteria limits that were applied in the cant deficiency tests on the Northeast Corridor (NEC).[23]

TABLE 5.1
SUMMARY OF SAFETY CRITERIA LIMITS FOR SPECIFIC TEST VEHICLES
1980-1982 NEC CANT DEFICIENCY TESTS

Maximum Permissible Test Measurement

Derailment Mechanism	Measurement	F40PH Locomotive	Banking Amcoach	Standard Amcoach	AEM-7 Locomotive	LRC Locomotive	LRC Coach
Vehicle Overturning	Steady State Weight Vector Intercept	399 mm 15.7 in	325 mm 12.8 in	325 mm 12.8 in	411 mm 16.2 in	414 mm 16.3 in	318 mm 12.5 in
	Transient Weight Vector Intercept	551 mm 21.7 in	478 mm 18.8 in	478 mm 18.8 in	564 mm 22.2 in	566 mm 22.3 in	470 mm 18.5 in
Wheel Climb	Transient Wheel (L/V)	0.9	0.9	0.9	0.9	0.9	0.9
Rail Rollover	Transient Truck Side (L/V)	0.57	0.65	0.65	0.59	0.57	0.65
Track Panel Shift	Transient Lateral Axle Force	186.4 kN 41,900 lb	83.2 kN 18,700 lb	83.2 kN 18,700 lb	151.2 kN 34,000 lb	183.7 kN 41,300 lb	81.0 kN 18,200 lb
	Transient Lateral Truck Force	266.0 kN 59,800 lb	121.4 kN 27,300 lb	121.4 kN 27,300 lb	215.3 kN 48,400 lb	262.0 kN 58,900 lb	119.7 kN 26,900 lb

In the context of this assessment, it is important to distinguish between aspects of body tilting, if any, which might impact the potential for derailment, and the more general safety concerns related to traversing curves at higher unbalanced speeds.

5.1.1.1 Vehicle Overturning

For tilt body operation, the issue in vehicle overturning is the likelihood that the combination of lateral inertial force acting at the c.g. of the vehicle in *higher cant deficiency operation*, coupled with the loading due to cross wind acting at its center of pressure (C_p) will be sufficient to remove any vertical load from the inside wheels in the curve. (It is the intended higher cant deficiency operation which is the fundamental issue; vehicle overturning is a design concern for any vehicle in worst case situations, such as travelling at underbalance speed (or stationary) through a superelevated curve with a crosswind inward to the curve).

[23] Ibid., FRA report DOT-FR-83-03, p. 1-4.

The lateral inertial force may be considered as comprising the steady-state force as developed in the body of the curve, and transient or dynamic forces resulting from transition spirals and alignment perturbations. Transient phenomena involve time duration which may or may not be sufficiently long for actual overturning to take place. Most overturning criteria deal with the forces acting through the c.g., with the wind-loading force used as a modifying factor which can be computed separately and applied additively.

The concept of weight vector intercept (WVI) has been traditionally used to quantify both steady state and transient criteria for vehicle overturning. The WVI is the distance from the centerline of the track to the point where the resulting force vector acting on the vehicle (from the lateral centrifugal and wind forces, and vertical gravitational force) intersects the plane of the railheads. A WVI of zero indicates symmetrical loading, while a WVI approaching 760 mm (30 in) for standard gauge track (one-half the track gauge) signals impending overturn.

Three criteria were identified and discussed in the primary sources cited above. The first criterion, the AAR's so-called "One-Third Rule," states that the WVI, neglecting wind loading, must lie within the center third of the track (no more than 254 mm [10 in] each side of the track centerline for standard gauge track). While in common use, it must be regarded as overly conservative.

The second criterion identified is the "Overturning Moment Safety Factor" developed by the Association of German Locomotive Manufacturers. This criterion is based on the ratio of the restoring moment based on the vehicle weight acting through the laterally-shifted c.g. to the sum of the overturning moments, including an allowance for a 110 km/h (68 mph) crosswind, being greater than 1.2. No distinction is made between steady state and transient loads.

The third criterion identified is the Vertical Wheel Load Reduction Ratio used at that time by JNR and later by its successor companies. This criterion measures lateral weight transfer in terms of the percent reduction in the vertical load on the inside (low-rail) wheels, and explicitly deals with both steady-state and transient load transfer effects. A reduction in wheel load by 60% of the nominal value is permitted for steady state curving, while an 80% reduction is allowed for transient peak wheel unloading (in terms of WVI for standard gauge track, this translates to 457 mm [18 in] steady state and 610 mm [24 in] peak). In establishing these limits, the transient overturning computations included the effects of transition spirals but not of track alignment perturbations causing short duration transients, and comparison of measured data through irregular track to the criteria may be somewhat conservative. The effect of wind loading is quantified by estimating the force generated by a wind velocity acting perpendicular to a surface area of the whole vehicle with a drag coefficient of 1. The overall equations used to establish the limiting WVI for the JNR criterion are given in **Figure 5.1**.

	Steady State Vector Intercept	$\leq 18 - (.0153V^2Sh_{cp}/W)$ inches
and		
	Transient Vector Intercept	$\leq 24 - (.0153V^2Sh_{cp}/W)$ inches
where:		
	V = the lateral wind speed in miles per hour	
	S = the lateral surface area of the vehicle in square feet	
	h_{cp} = the height of the center of wind pressure in feet	
	W = one-half of the unloaded weight of the vehicle in pounds	

Figure 5.1: JNR Vehicle Overturning Criteria

This criterion was used as a basis for assessment in the 1980-82, 1983, and 1988 cant deficiency tests of, variously, LRC, F40PH, Amfleet, Talgo, RTG Turbo I and RTG Turbo II equipment on the NEC. As discussed in Section 4, it was the vehicle overturning criteria which was the most restrictive derailment safety limit on the passenger equipment operating at high cant deficiency. **Figure 5.2** compares the three criteria as applied to the LRC coach as a function of wind speed. A wind speed of 90 km/h (56 mph) was used as the limiting value in the assessment because it is the greatest expected in the NEC within 4.5 meters (15 ft) of the ground for a 10-year mean recurrence interval. In this case, the crosswind allowance by itself could equal a wheel unloading of almost 20%.

A question remains as to which of the two criteria, steady-state or transient, might limit the cant deficiency allowable for safe operation. The maximum cant deficiency satisfying the steady-state overturning criteria for a particular vehicle with a maximum crosswind can be determined analytically from a knowledge of the suspension characteristics, mass distribution and surface area, and correlates well with tests. The estimation of limiting cant deficiency based on transient criteria is more difficult to validate, both analytically and by test. For the JNR criterion, use of a cant deficiency limit based on steady-state weight transfer implicitly assumes that there are no track perturbations capable of causing additional transient wheel unloading greater than 20%.[24] Few exceptions to the limits based on the steady-state criterion were found in the NEC tests, and all exceptions were associated with switches, undergrade bridges or grade crossings.

[24] Railroad Passenger Ride Safety, Owings, R.P., Boyd, P.L., prepared by RHOMICRON, Inc. and ENSCO, Inc. for the FRA Office of Research and Development, Report No. DOT-FRA/ORD-89/06, April 1989.

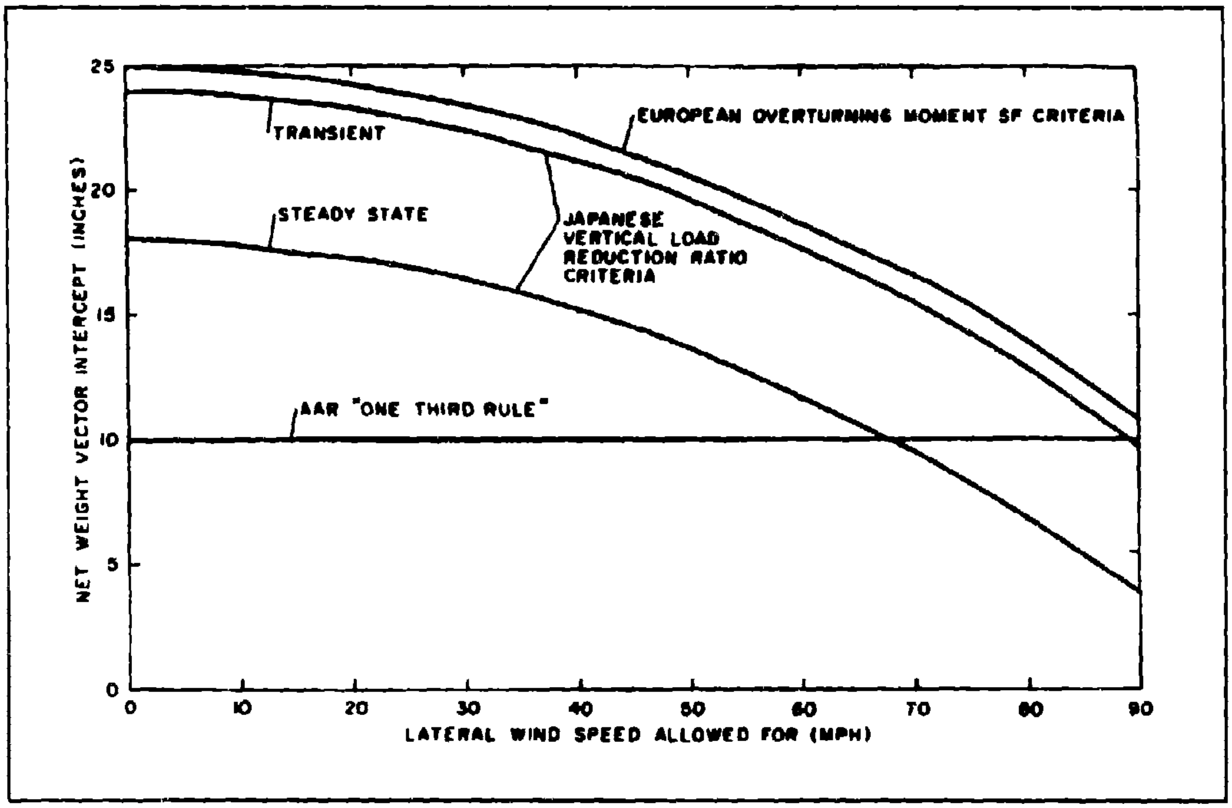

Figure 5.2: Comparison of Three Different Overturning Criteria Applied to the LRC Coach

The risk of derailment from vehicle overturning is of particular concern with passive-tilting technologies. Since passive-tilting is based on pendular motion, even a relatively modest tilt angle will result in additional outward lateral displacement of the c.g. and the weight vector intercept. The only effective countermeasure is to make the center of gravity of the vehicle as low as possible, and to restrict the maximum tilt angle, so that the consequent overturning moment is minimized.

This concern with the risk of overturning, as well as passenger comfort considerations related to tilt rate, have effectively limited the passive-tilt angles to 5° or less, in contrast to the 7° to 10° that are commonly achieved with active-tilt systems. The successful Talgo *Pendular* coaches combine a limited tilt angle with a low c.g. achieved by supporting the body structure *between* the articulating trucks, rather than on top of the trucks as is the case in most conventional passenger equipment. The UAC Turbotrain also adopted this strategy, and this was one of the features of that equipment that performed consistently well.

The inclusion of wind-induced lateral force in the assessment of vehicle overturning risk results in a more stringent cant deficiency limits for coaches, which are typically much lighter, and have a larger lateral area (due to their greater length) than for (shorter, heavier) locomotives, since the vertical gravitational force is limited by the vehicle mass. There is a clear incentive to minimize the area of the vehicle side exposed to crosswinds, and to optimize vehicle aerodynamics to address this as well as more conventional concerns. Again, the Talgo coaches do very well in that regard, being about half the length of the LRC coach.

5.1.1.2 Wheel Climb

"Wheel climb" refers to a phenomenon in which the forward motion of the axle combines with the wheel and rail profiles, surface conditions and interactive forces, to permit the wheel flange to roll, with creepage or slip, up onto the head of a rail.[25] This derailment condition may be temporary or it may result in wheel drop. Wheel climb has been known to occur in steady-state curving, spiral negotiation, and dynamic curving which is often exacerbated by braking and traction forces in curves, and is almost always accompanied by some wheel unloading.

The maximum ratio of lateral force (L) to vertical force (V), or maximum L/V ratio on any individual wheel, continues to be used in assessing proximity to wheel climb derailment. As the ratio between lateral and vertical forces increases, the risk of derailment due to wheel climb rises. It has been shown that the risk of wheel climb derailment has no explicit relationship to the time duration of the applied forces,[26] although some empirical relations, developed from specific vehicle and track-operating condition tests, have been used as criteria in the past.[27]

On curved rail, with conventional trucks and pairs of wheels that are fixed to a single axle, the flanges of the outside wheels may be forced into contact with the inside face of the rail at some angle of attack (the angle between the direction of the velocity of the axle center and the normal to the axle center of rotation). As the wheel rotates, for positive angles of attack, the force of friction between flange and rail face attempts to lift the wheel upwards. This climbing force is resisted by the downward vertical wheel load, P_w. As speed through a given curve increases, the (lateral) contact force between the wheel flange and the rail will also increase, causing greater adhesion between wheel and rail, and thus a greater "climbing" force, while the vertical downward force remains unchanged. For negative angles of attack, the friction forces act to inhibit derailment and larger L/V ratios can be sustained.

A comprehensive review of wheel climb derailment and the criteria used to estimate the critical values of L/V is given in the AAR Report No. R-717 cited above. The criterion as applied

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[25] A Review of Literature and Methodologies in the Study of Derailments Caused by Excessive Forces at the Wheel/Rail Interface, Blader, F.B., AAR Report No. R-717, December 1990.

[26] "Wheel Climb Derailment Criteria for Evaluation of Rail Vehicle Safety," H. Weinstock, presented to the Rail Transportation Division, ASME Winter Annual Meeting, Dec. 1984.

[27] "Theory of the Derailment of Wheelset," K. Yokose, Quarterly Report, RTRI, V.7, No. 3, 1966. "Dynamics of High-Speed Rolling Stock," T.M. Matsudaira, JNR Quarterly Report (Special Issue), 1962.

during the NEC tests was to limit the L/V ratio to 0.90, except for short duration transients.[28] Testing revealed that L/V ratios remained below about 0.5 during high cant deficiency curving, except at switches in high speed, low cant deficiency curves.

The L/V ratio is very much a function of the angle of attack. As such, the propensity to derail through wheel climb will be primarily a function of the truck performance and only secondarily by carbody tilting. In fact, wheel climb derailments are less likely at speeds over balance, for otherwise similar circumstances.

Technologies equipped with steerable trucks (the X2000, the SIG truck with "Navigator") will clearly have an advantage in this regard, with suspension elements interlocked with the tilt mechanism to reduce the angle of attack in curves, and the ETR-450 which has an active lateral secondary suspension and a longitudinally-flexible primary suspension. The advanced-concept tilt trains, such as the Fiat "AVRIL" and the RTRI 250X concept with independent-wheel trucks and active suspensions may well offer the best control of wheel climb, albeit at a price in terms of added complexity and sophistication.

With passive-tilting technologies, in which the roll stiffness of the carbody may be softer, an important design criteria is to ensure that no harmonic roll effects lead to dynamic wheel unloading (lower V) which might enhance the potential for wheel climb (and vehicle overturn).

5.1.1.3 Gage Widening

Under the influence of static-wide gage track and large lateral forces between wheel and rail, sufficient lateral rail deflection can occur to allow a wheel to drop between the rails. This "gage widening" derailment process may involve rail rollover and/or lateral translation of the rail cross-section, and will be influenced by the rail-tie fasteners which restrain the rail from translation, rollover and longitudinal creep. The restraining force can vary substantially, from about 3.6 tonnes (4 tons) for elastic fasteners such as are used in the NEC and generally on concrete ties, to about 1.6 tonnes (1.8 tons) for new wood ties with cut spikes. Lateral rail deflection without roll occurs when the lateral spreading force reaches the limit of adhesion (between the rail base and tie surface) for the vertical load carried.[29] Lateral rail deflection typically occurs on lower-speed track and is usually a result of the loss of adequate cross-tie and rail-fastener strength.

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[28] High Cant Deficiency Testing of the LRC Train, the AEM-7 Locomotive, and the Amcoach, Report No. DOT-FR-81-06, P. 5-20, (NTIS: PB 82-213018), January 1982.

[29] Development of an Improved Vehicle Loading Characterization, associated with the Gage Strength of the Track, Manos, W.P.; Scott, J.F.; Choros, J.; and Zarembski, A.M., AAR Report No. R-493, August 1981.

Gage-spreading forces between the wheels and rails arise from an angle of attack of the wheel to the rail, and the resulting forces may be large in curving, again dependent on the performance of the truck. Long and rigid trucks which prevent the axle from steering adequately induce large forces. Transmission of loads from heavy bodies, such as locomotives, when excited by track perturbations has also been a concern in gage-widening derailments. Gage-widening can be self-sustaining, in that, as the rail-tie fastening becomes degraded, track geometry irregularities become more pronounced which, in turn, lead to higher wheel/rail loads and gage-spreading forces. Accordingly, regular track inspections are required to minimize the risk.

For the NEC high cant deficiency tests, the instantaneous ratio of the sum of lateral forces to the sum of vertical forces of the wheels on the high rail side of a truck (known as the truck L/V ratio) was used to quantify the likelihood of rail rollover, based on AAR studies. Truck L/V ratios measured at the high rail side of the vehicles tested remained low relative to the critical levels, for cant deficiencies up to 280 mm (11 in).

Recent contributions made to the prediction of gage widening are presented and theories discussed in the above cited AAR Report No. R-717. From a vehicle standpoint, improved truck technology will be instrumental in minimizing the risk of gage widening in high cant deficiency operation.

5.1.1.4 Lateral Track Shift

This final curving safety criterion addresses the likelihood of derailment as a consequence of lateral movement of the entire track superstructure (rails, fasteners, ties) through the ballast. Any shift of noticeable magnitude (of the order of one inch) is regarded as an incipient derailment. Track panel shift has become increasingly important as the speed of vehicles increases and more continuous welded rail (CWR) is used.

Vehicle induced forces which have increased in magnitude with speed are generally large inertial loads arising from high cant deficiency operation and from heavy dynamic response to poorly aligned track.

Track lateral stability is dependent on the characteristics and condition of the ballast, the width of the ballast section outside the end of the ties, the degree of compaction due to traffic, the shape, weight, material and spacing of the ties, the stiffness of the rail and fasteners as well as changes in ambient temperature. Results from tests on one type of track construction and condition may not be applicable to another when establishing safety limits for allowable forces. As an example, compaction due to traffic appears to have a large effect: the lateral resistance of loaded ties is reported to double after 100,000 gross tonnes (110,000 G Tons) of traffic, and to stablize at around three times the value for uncompacted ballast after 1.5 MGT (1.65 MG Tons). The tie-related factors, including material (concrete) add up to 60% to lateral resistance.

On the other hand, repeated passes over irregular track may reduce buckling strength, and ground-borne vibrations may cause loss of lateral ballast resistance. The situation is further complicated by thermally-induced forces in CWR.

The criterion used in the NEC tests was based on measurements on French track using the "Wagon Derailleur" car,[30] modified to account for internal forces in CWR due to temperature changes and lateral carbody forces due to unfavorable crosswinds for wood-tie track on compacted ballast. It was assumed that a single axle bears half the lateral wind load.

Criterion was established both for maximum axle lateral force and maximum truck force. The results were quite conservative insofar as the only quantitative track shift data are much out of date, being the product of SNCF tests on 45 kg/m (92 lb/yd) rail.

Figure 5.3 shows the lateral track shift criterion as applied to different vehicles under different ballast conditions.[31] Measurements, little more than half the critical levels, indicated that safety against lateral track shift did not limit the cant deficiency for the trains under test.

As well as the curving criteria discussed above, U.S. standards and practices do not consider vertical impact loads beyond definition of the maximum axle load acceptable under AAR interchange rules - 30 tonnes (33 tons). These dynamic forces adversely affect rail life and pose a risk of derailment through fracturing of the rail. Railways in Europe and Japan have developed a number of criteria for vertical impact load.[32] The consequence of these criteria is to limit the static axle load to 20 tonnes (22 tons) or less and the unsprung mass to about 2 tonnes (2.2 tons). It would be informative to explore how the equipment tested in the NEC would fare in terms of this criterion.

5.1.2 Compatibility With Clearance Envelopes for Existing Lines and Equipment

Tilt-body operation could very well require greater right-of-way clearances than rolling stock in current operation. Compatibility with clearance envelopes for existing tracks and equipment types must be carefully examined on routes over which the tilting equipment may be employed. If tilt capability is procured to increase speed in curves and reduce travel time on existing tracks, the purpose is somewhat defeated should new or extensively rebuilt tracks be required to accommodate tilt.

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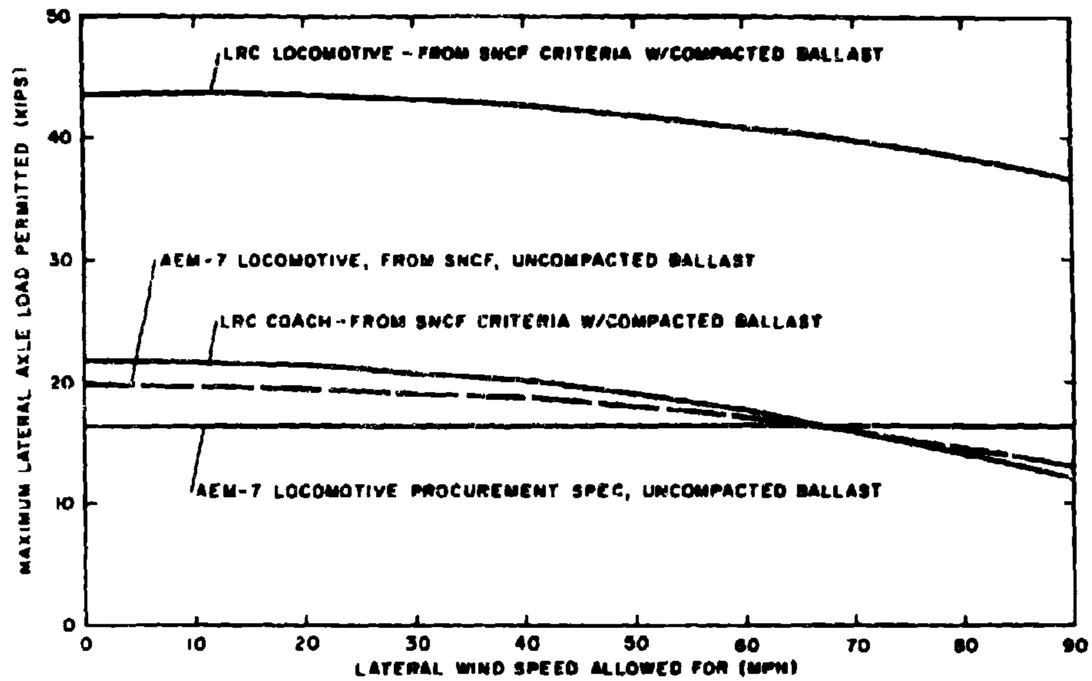
[30] Elastic and Lateral Strength of the Permanent Way, Sonneville, R. and Bentot, A., Bulletin of the International Railway Congress Association, November 1969, pp. 685-716.

[31] High Cant Deficiency Testing of the LRC Train, the AEM-7 Locomotive, and the Amcoach, Report No. DOT-FR-81-06, p. 5-29, (NTIS: PB 82-213018), January 1982.

[32] See, for example, "The Effect of Track and Vehicle Parameters on Wheel/Rail Vertical Dynamic Forces," H.H. Jenkins et. al., Railway Engineering Journal, January 1974.



(Source: FRA/Amtrak Report DOT-FR-83-03)

Figure 5.3: Comparison of Lateral Track Shift Criteria for Different Vehicles

Particular clearance considerations include:

- o Interference between tilted vehicles and wayside obstacles in curves, both side-to-side and overhead,

- o Interference between tilted vehicles and all equipment-type vehicles (tilted or stationary) on adjacent track in curves, and

- o Interference between tilted vehicles in a failed condition anywhere in the system and either wayside obstacles or other failed tilt vehicles on adjacent track, including, in the worst case, vehicles tilted at the opposite extremes.

For the tilt-body vehicle, this requires calculations or measurements of the maximum tilt and the lateral offset of the c.g. that would be expected in normal operation at the maximum cant deficiency for a safe comfortable ride. In fact, a more conservative "worst case" approach would be to consider the vehicle's maximum tilt throughout the system as an indication of potential trouble should a tilt system fail in its maximum tilt position. However, a "fail-safe" tilt system design should obviate this requirement somewhat.

Track centerline spacing is a major clearance factor. The Amtrak Specification for Construction and Maintenance of Track (MW-1000) give standards for new construction as: tangent track, 4.27 m (14 ft, 0 in) track centers; curved track, increase track center spacing 25 mm (1 in) for each 0.5 degree of curvature and add 89 mm (3.5 in) for every 25 mm (1 in) difference in superelevation between the two tracks. This standard for new construction provides a 152 mm (6 in) minimum clearance for various curvatures and superelevations for conventional domestic equipment. Amtrak's Standard Minimum Roadway Clearances (Drawing No. 70050-A) describe wayside clearances which must be observed as new construction standards. However, caution must be exercised since much of existing track is not new construction and existing track centers are frequently 3.66 m (12 ft) and sometimes less in the Northeast Corridor.

A clearance evaluation for tilt-body vehicles in Northeast Corridor operation was included as part of the IPEEP in 1978.[33] The existing dimensions of the Northeast Corridor were accommodated by ensuring that procured equipment would stay within the clearance envelope described by Amtrak Drawing No. 70050-G titled "Maximum Dimensions for Passenger Equipment Moved in Penn Central Electrified Territory In-between New Haven and New York; New York and Washington; New York and Harrisburg; and Washington and Harrisburg." These dimensions provided sufficient clearance at the mid-point and ends of cars with 18.14 m (59 ft, 6 in) truck centers and conventional (inactive) suspension systems for curves up to 13 degrees. Examples from the clearance evaluation for the prototype active-tilt LRC passenger coach in the Northeast Corridor are shown in **Figures 5.4** and **Figure 5.5**.

In **Figure 5.4**, a comparison of the LRC car is made against the Northeast Corridor Construction Limit Outline, both "at rest" and at the "full-tilt" condition. The Construction Limit Outline allowed for body roll and lateral offset of 3^0 and 51 mm (2 in) respectively of <u>conventional equipment</u> as well as limits for normal service conditions such as wheel wear, maximum spring travel, and faulty springs without fouling wayside obstacles. It can be seen that the LRC car was a borderline case, slightly exceeding the limit outline and requiring a more comprehensive examination. The most restrictive conditions were determined to be a moving train passing a stopped train on a 152 mm (6 in) superelevated curve, and a moving train passing a stopped train in the B & P tunnel. Calculated clearances at specific locations for normal operations of the LRC are shown in **Figure 5.5** where a potential problem in the tunnel was identified for tilt operation.

[33] <u>Clearance Considerations of Tilting Body Vehicles on the Northeast Corridor</u>, Working Paper for IPEEP, L.T. Klauder and Associates, July 25, 1978.

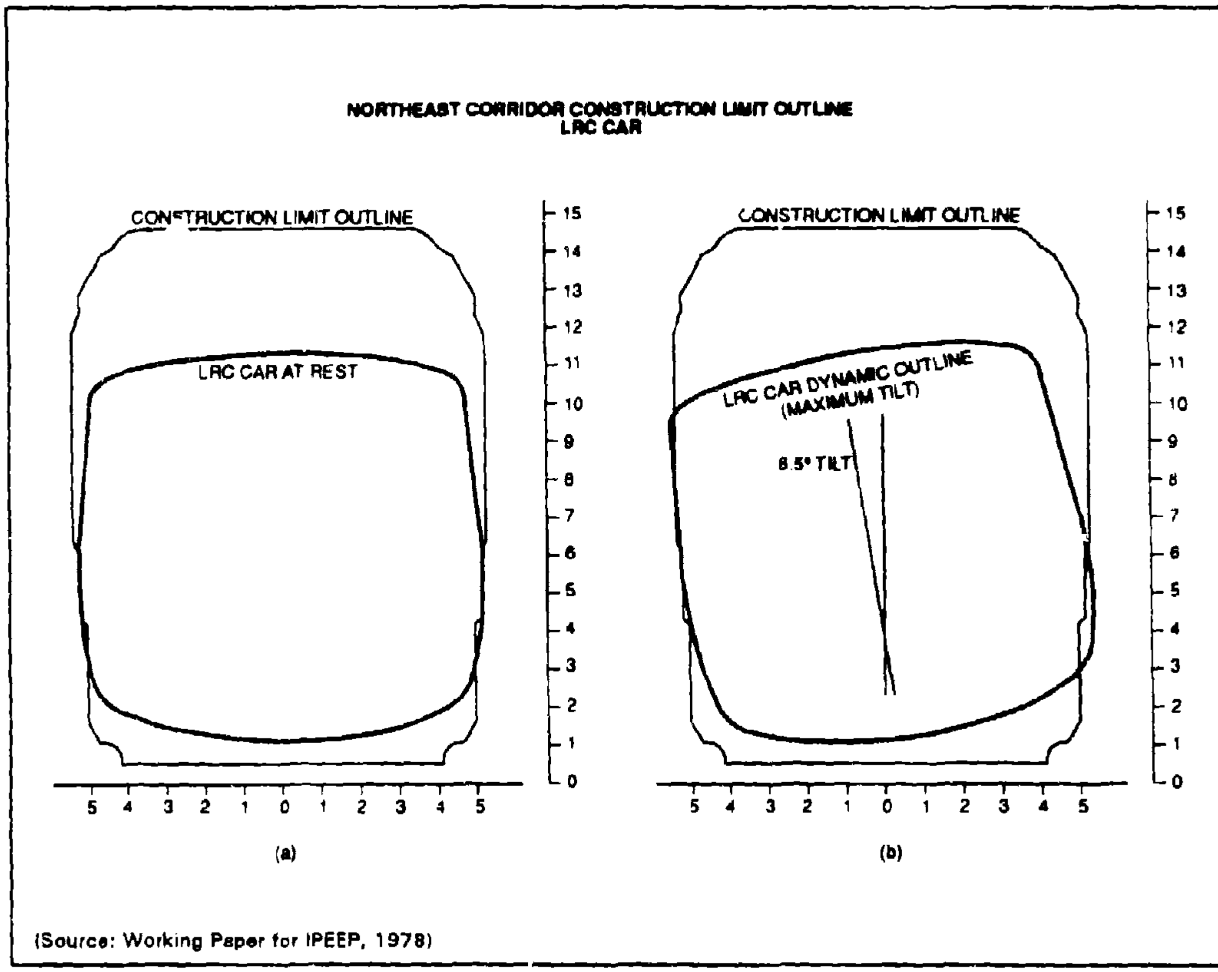


Figure 5.4: Northeast Corridor Construction Limit Outline, LRC Prototype Car, Maximum Tilt

5.1.3 Maintainability and Reliability

The keystones of conventional North American railroad equipment design have been historically, and to a considerable extent remain, the rugged simplicity and interchange compatibility. Perhaps the most outstanding example of this emphasis is the three-piece truck, which literally supports rail freight movements and is the basis for most trucks on existing Amtrak passenger vehicles, albeit with a more sophisticated secondary suspension. With AAR interchange compatibility, a freight car can be operated anywhere from southern Mexico to northern Canada, and may spend much of its service life off the tracks of the owning railroad. Passenger equipment has traditionally remained on its owning railroad, but the vehicles were built to meet AAR interchange requirements, and so reflected a similar simplicity. Robust mechanical, electrical and/or pneumatic designs were and are standard, with more sophisticated electronics just starting to have a real impact on the national locomotive fleet. Complex subsystems, especially those with hydraulic components such as dampers, have been regarded with distrust

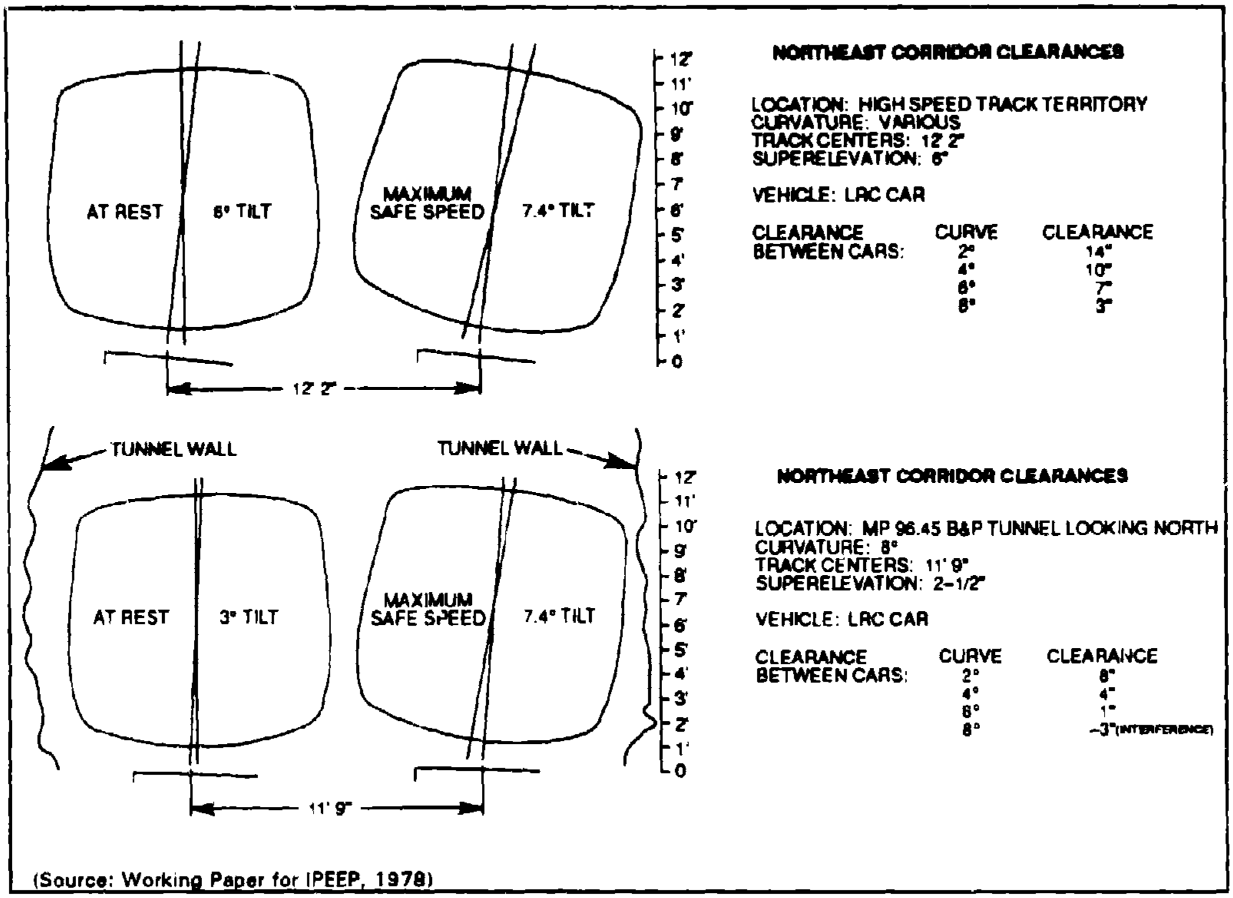


Figure 5.5: Northeast Corridor Clearances, LRC Prototype Car, Curves and Tunnel

by the North American railroad industry, and with some reason, given the way equipment was operated and maintained.

One of the consequences of these factors was the necessity to be able to maintain vehicles virtually anywhere, with limited facilities and often under very primitive conditions (outside, in winter, for example). Another consequence was that such maintenance as was done was virtually always reactive; that is, performed because something had failed and the car could not be used until repairs were made. This approach to maintenance was largely dictated by an inability to monitor either utilization or elapsed time at the level of the individual car - a situation that is rapidly changing for the better on the freight side and has largely been abandoned by Amtrak. However, a significant proportion of railroad managers and maintenance employees grew up with the "wait 'til it breaks then fix it" philosophy, and this background certainly affects their attitudes toward complex vehicles and programmed preventative maintenance.

Put simply, active-tilt passenger vehicles are all sophisticated and complex, incorporate unfamilar and often quite delicate components in critical subsystems, and must be maintained in a purpose-built, or at least purpose-renovated facility, in accordance with an aviation-style maintenance schedule linked to utilization and/or elapsed time, by skilled workers familar with the full range of advanced components in the equipment. Suppliers and railroads in Europe and Japan have decades of experience designing, managing, and executing this type of programmed maintenance activities, both for vehicles and track. For these railroads, the advent of tilt-body trains represented an increase in complexity, but a rather modest one, a change of degree rather than of nature in the skills, facilities, and procedures required.

As part of the implementation and commissioning process for the X2000, SJ, in concert with ABB Traction, have developed a comprehensive program of scheduled maintenance, based on a detailed analysis of possible failure modes and a comprehensive component database with MTBF and MTTR data for each component, and detailed information on labor qualifications and standard unit inputs for labor and materials.

Table 5.2 summarizes the programmed maintenance procedures developed by ABB and SJ for the X2000 equipment, including the nature and interval for each class of scheduled maintenance planned for the X2000. Note that this table does not include any estimate of the level of effort involved in refurbishment of components that are changed out during any of these scheduled activities.

From the joint experience of ABB and SJ with the experimental trainset and with similar components in revenue service with SJ and elsewhere, SJ anticipates that the ratio of scheduled maintenance to corrective or emergency maintenance will be between 4:1 and 5:1 (i.e, about 16% to 20% of maintenance effort will be corrective; the rest will be scheduled). This is typical for European passenger equipment, and for some types of (non-tilting) technology, such as TGV, the ratio is even higher, approaching 9:1.

Table 5.3 summarizes the scheduled maintenance activities for Amfleet cars. No level-of-effort estimates were available for these activities.[34] Although the level of detail in the respective source materials varies, it is clear that the ABB/SJ program is more comprehensive in scope and deals with more critical subsystems and components in an aggressive fashion (i.e., changing all truck dampers after nine months of operations, and changing out all vital components after three years, rather than depending on inspections and judgement to determine the timing of component changeouts).

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[34] Amtrak Maintenance and Parts Manual - Locomotive-Hauled Passenger Cars, Vol. 4.

TABLE 5.2
X2000 PREVENTATIVE MAINTENANCE SCHEDULE AND REQUIREMENTS

FREQUENCY	TYPE OF ACTIVITY	LEVEL OF EFFORT
Each Trip	General Visual Check	0.9 hours, 0.9 person-hours
6280 km (3900 miles) or Weekly	Safety Check: general inspection of bogies and brakes, check of brake function, external check of hydraulics for leaks, inspection of pantograph	1.0 hours, 3.0 person-hours
25,100 km (15,625 miles) or Monthly	First-Stage Preventative Maintenance: brake and brake control tests, internal check of hydraulic system for leaks, door function and controls, pantograph contacts, other general inspections	3 hours, 13 person-hours
100,600 km (62,500 miles) or Quarterly	Second-Stage Preventative Maintenance: All work specified above, plus test of magnetic rail brakes, pressurized air system, dampers and tie-rods, wires and cables, measurement/correction of wheel profiles, inspection of cooling system and filter change, inspection of fire and other safety equipment and batteries	8 hours, 36 person-hours
301,500 km (187,500 miles) or once every 9 months	Third-Stage Preventative Maintenance: All work specified above, plus total brake function and control validation, HVAC inspection, test/validation of on-board computer system, inspection of electrical joints and cooling pipes, check of solid-state electronics, and high-voltage equipment, oil change on compressors, change of brake shoes/pads, change of primary dampers and yaw dampers, oil change in gear box and transmissions	24 hours, 115 person-hours
603,500 km (375,000 miles) or 18 months	Fourth-Stage Preventative Maintenance: All work specified above, plus check of set limit values in control system, change of hydraulic oil, air spring inspection, coupling lubrication	24 hours, 170 person-hours
1,207,000 km (750,000 miles) or 36 months	First Major Overhaul: Exchange of vital components (motors, fans, compressors, gear boxes, trucks, hydraulic cylinders, valves and compressor units, active components in brakes and pressurized air system, vacuum pumps), cleaning of all electrical cabinets, oil exchange in converter system, selective renewal of interior components	21 days, 1400 person-hours
3,621,000 km (2,250,000 miles) or 9 years	Second Major Overhaul: All of above work, plus additional component exchange on brakes, electrical contacts and compressed-air system; complete renewal of train interior and exterior	42 days, 5000 person-hours

TABLE 5.3
SCHEDULED MAINTENANCE ACTIVITIES - AMFLEET CARS

FREQUENCY	TYPE OF ACTIVITY
DAILY	General Inspection; check brake linings, drain toilet-holding tanks
MONTHLY	Check battery system; check A/C system; check/clean food service car condensors
QUARTERLY	Inspect/clean HVAC; Inspect journals; check trucks, brakes and electrical system
SEMI-ANNUALLY	Lubricate journals; inspect/clean water coolers and toilets; inspect/service shock absorbers and bolster center pivot.
YEARLY	Clean/check couplers; check/service HVAC; inspect/service door system and handbrakes.
2-YEAR	Check/test/service brakes
3-YEAR	Service brakes in accordance with AAR PC Rule 2
4-YEAR	Remove, clean, test door mechanism; overhaul brake cylinder, tie rod, bearing; replace air hose
5-YEAR	Overhaul airbrakes; overhaul truck; Remove, inspect and repair journal bearings

This difference has a direct bearing on the operational reliability of the equipment. Even though it is complex, the maintenance program will largely ensure that faults capable of disrupting service or posing a risk to passengers and crew will be detected before a failure occurs. It should be noted that a significant number of the reliability problems that plagued the VIA Rail Canada LRC fleet were eliminated or much reduced after VIA opened its purpose-built maintenance facilities in Montreal and Toronto and took over the contracts of its maintenance employees from CN and CP. The latter change allowed VIA to institute effective training and trouble-shooting programs.

The more complex tilt-body equipment has been designed to facilitate inspection and maintenance activities, typically in conjunction with design of the facilities and tools required to best do those activities. Coupled with skilled and well-educated labor and effective training programs, the results at FS, for example, have been very good.

The bottom line with respect to maintainability and reliability is that there will have to be a major shift in the philosophy of vehicle maintenance towards aviation practices, together with an expansion of labor and management skills to deal with complex hydraulics, sensors, and microelectronics. U.S. operators will also have to deal with the skills and knowledge base needed to cope with ac traction motors, steerable trucks and other elements of tilt-train design that are not directly linked to the tilt mechanism, but form an essential part of the equipment. If this is done, and the specified maintenance activities performed as programmed, there is no reason why these technologies should not perform as well in the U.S. as anywhere in the world.

5.1.4 Effects of U.S. Alignment Geometry and Track Maintenance Standards

The principal issue here stems from the fact that to date, essentially all tests on tilt-body equipment in the U.S. have been conducted in the NEC. While there are very good and practical reasons why this should have been the case, one must be cautious in extrapolating these results to other rail lines.

There are several reasons why this caution is justified and why additional investigations are needed to establish the general applicability of tilt-body equipment.

First, the track quality in the NEC is arguably some of the best in the country. While the alignment geometry of the line north of New York is certainly not exceptional, the track geometry is very good and the track structure is excellent. While categorized as FRA Class 6 track - the best track classification available under current U.S. regulations - there is no question that the quality is much closer to that of the "conventional" (160-200 km/h; 100-125 mph) tracks of Europe and Japan, certainly well above the Class 5/Class 6 boundary.

At present, there are no data to demonstrate how tilt-body equipment will respond to the alignment and track geometry conditions on routes which are still Class 6 but marginally so. The implications of operation on rougher track must be assessed not only in terms of the ability of the tilt and suspension systems to deliver acceptable ride quality at higher curving speeds, but also in terms of the effects on component life, required maintenance cycles, and the life-cycle costs of alternative technologies.

5.1.5 The Incremental Costs and Benefits of Tilting

While assessment of technology-specific costs and benefits is beyond the scope of this report there are several underlying principles that need to be borne in mind when considering supplier claims with respect to cost, or examining the cost experience of a foreign operator.

First among these is that, with the possible exception of the LRC, any other tilt technology imported to the U.S. will be operating on a technological "island," with little or no opportunity to benefit from economies of scope or scale, and with the prospect of being at the end of a rather long supply line in terms of parts and expertise. This in itself will raise the level of effort required for many activities, at least in the early stages of deployment. For a foreign operator whose work force and facilities are already attuned to the technological complexities and maintenance requirements of equipment of this class, whether tilting or non-tilting, the addition of tilting trainsets to its fleet will represent an increment to an already established national network. The first U.S. operator could be faced with what amounts to a *state change* in process as well as skills and facilities. The nature of the cost base for a cost assessment in the U.S. will be fundamentally different than would be the case in Europe or Japan. Estimated cost increments should not be extrapolated to U.S. situations.

Second, there should be a clear understanding of cost causality and of the input factors (materials, labor by skill class, tools and equipment, facilities, etc.) required for all aspects of the life cycle of the tilt equipment, including all associated processes and procedures. These data will allow development of a realistic model of activities reflecting differences in utilization, procedures and factor quantities, and ultimately of the life-cycle costs.

With respect to an assessment, at the level of an operator, of the benefits of tilting, the key issue is to make sure that the trip time gains from body tilting are based only on speed improvements on curves which are constrained by passenger comfort considerations. There may be other features of a given technology that will improve the curving characteristics of the vehicles. Specificity and attention to the details of a given route are essential for credibility.

5.2 ISSUES FOR EQUIPMENT NOT DESIGNED TO U.S. STANDARDS

Equipment and technology developed outside the U.S. may be built to a variety of technical standards which may differ from those applicable to conventional railroad equipment and infrastructure in the U.S. The issues arising from the potential application of offshore tilt technology are the same in principle as those affecting non-tilting technologies like the TGV or the ICE.

5.2.1 Overview of Safety Issues

A comprehensive and thorough assessment of the safety issues and concerns associated with the types of high-speed rail systems like the tilt body has been recently prepared for the FRA.[35] That report lists individual safety issues for further study. **Tables 5.4 to 5.15** address major safety issues and list the sub-issues which are typically the subject of a set of regulations, standards and practices, and the types of accident affected by the issue. The reader is directed to the cited report for an in-depth review of the "generic" safety issues affecting all high-speed rail systems.

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[35] An Assessment of High-Speed Rail Safety Issues and Research Needs, A.J. Bing, prepared by A.D. Little, Inc. for the FRA Office of Research and Development, Report No. DOT/FRA/ORD-90/04, (NTIS: PB 92-129212), December 1990.

TABLE 5.4: PRIMARY STRUCTURAL CRASHWORTHINESS

HSR SAFETY ISSUES—1—PRIMARY STRUCTURAL CRASHWORTHINESS

Issue	Sub-Issues	Information Needed Regarding Regulations, Standards and Practices	Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)
Car and Locomotive Structural Integrity and Crashworthiness: Ability of car and locomotive structures to withstand normal service and emergency (collision, derailment) loadings, and to provide adequate protection for occupants	• Buff strength • Collisions posts • Couplers • Anti-climb features • Truck body connection • Structural integrity of engineer's cab	• Buff strength criteria • Collision post strength • Coupler strength and energy absorption • Truck-body connection strength • Engineer's cab protective structure - locomotives - cab-cars • Specific rules for cab-car operation with cab leading • New equipment qualification tests (e.g., squeeze test)	• Reduces risk that occupants of vehicles involved in a high-energy collision or derailment will become casualties

TABLE 5.5: CONSTRUCTION OF TRUCKS AND BRAKES

HSR SAFETY ISSUES—2A—CONSTRUCTION OF TRUCKS AND BRAKES

Issue	Sub-Issues	Information Needed Regarding Regulations, Standards and Practices	Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)
Truck and Braking System Integrity Ensuring that trucks, especially wheel sets, can withstand the normal operating environment, and that the brake system operates in a proper fail-safe fashion	• Wheel/axle/bearing integrity • Truck structure integrity • Wheel load variation • Ensuring acceptable stopping distance under all operating conditions, relative to signal standards • Avoidance of damage to wheels, brake discs, etc. • Potential failure modes • Adequacy of parking brake • Adequacy of non-conventional brake systems, e.g., hydraulic activation, eddy-current brake)	• Wheel/axle/bearings - dimensions - materials - manufacturing and assembly requirements • Truck structural design criteria • General description of braking system • Brake performance - normal service - emergency - failure modes (e.g., reverting from electric to pneumatic control) - spin/slide protection system • Parking brake design and performance • New design test and qualification procedures	• Reduces risk of derailment caused by equipment defects, such as brake, truck or wheelset failures

TABLE 5.6: TRACK TRAIN INTERACTION

HSR SAFETY ISSUES—2B—TRACK TRAIN INTERACTION			
Issue	**Sub-Issues**	**Information Needed Regarding Regulations, Standards and Practices**	**Relationship to FaultTree (Types of Accidents/ Incidents or Situations Affected)**
Track Train Interaction Avoiding unsafe wheel-rail forces or force ratios which could damage track or cause derailment or overturning	• Flange-climbing derailments • Track panel shift • Rail rollover • Overturning of car • Safety acceptability of high cant deficiency curving, with or without tilt • Standing and walking passenger safety • Active/passive tilt system integrity	• Acceptability criteria used for - Lateral force at individual wheel, wheelset or truck - L/V force ratios - Max. wheel unloading on warped track - Max. acceleration in passenger space, including those applicable to standing & walking passengers • Maximum cant & cant deficiency permitted as a function of speed & curvature • Precautions against truck hunting • New design qualification test procedures • Tilt system safety features	• Reduces risk of derailments - Overturning - Flange-climbing • Track failures due to excessive wheel-rail forces • Tilt system malfunctions

TABLE 5.7: ROLLING STOCK MAINTENANCE AND INSPECTION

HSR SAFETY ISSUES—2C—ROLLING STOCK MAINTENANCE AND INSPECTION			
Issue	**Sub-Issues**	**Information Needed Regarding Regulations, Standards and Practices**	**Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)**
Maintenance and inspection procedures needed to keep vehicles in safe operating condition	• Wheels/axles/bearings	• Frequency/nature of inspection - Acceptability criteria - Wheel wear limits - Use of line-side detectors	• Reduces risk of derailments or collisions due to brake or truck malfunctions arising out of defective components or systems
	• Dynamic stability	• Monitoring devices used - Acceptability criteria	
	• Brakes	• Frequency/nature of inspection - Acceptability criteria	
	• Tilt systems (if fitted)	• Flequency/nature of inspection - Acceptability criteria	
	• Maintenance staff training, qualification procedures	• Staff training and qualification procedures • Quality control in maintenance work	

TABLE 5.8: NON-STRUCTURAL VEHICLE SAFETY AND CRASHWORTHINESS

HSR SAFETY ISSUES—3—NON-STRUCTURAL VEHICLE SAFETY AND CRASHWORTHINESS			
Issue	**Sub-issues**	**Information Needed Regarding Regulations, Standards and Practices**	**Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected**
Non-structural rolling stock safety and crashworthiness Adequacy of non-structural rail vehicle features to protect passenger and train crew from hazards	• Fire precautions • Doors • Vehicle interior crashworthiness • Baggage restraint • Glazing and windows • Emergency access and escape • Air pressure changes (e.g., on tunnel entry) • FRA safety appliances • FRA flammability and smoke-emission standards	• Fire precautions - Warning devices - Firefighting equipment - Flammability standards • Doors - Step heights, etc. - Locking • Glazing standards • Interior crashworthiness - Seat/structure fastening strength - Protection of hard surfaces - Baggage restraint • Air pressure change limits • Emergency access and escape provision • Emergency lighting • Emergency response plans	• Reduces risk of casualties to occupants of colliding or derailed trains • Reduces risk of casualties to passengers boarding cr alighting from trains • Reduces risk of casualties to railroad employees working around moving vehicles (coupling switching, etc.) • Reduces casualties due to on-board fires

TABLE 5.9: TRACK STRUCTURE INTEGRITY

HSR SAFETY ISSUES—4—TRACK STRUCTURE INTEGRITY			
Issue	**Sub-issues**	**Information Needed Regarding Regulations, Standards and Practices**	**Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)**
Track structure integrity Construction standards for track structure to insure safety under normal operating conditions	• Track strength - Roadbed stability - Panel shift - Rail roll-over safety under normal service loads • Track quality - Geometry - Rail and weld metallurgical quality Includes: - Curves and tangent - "Plain" and "special" trackwork	• Dimensions, materials, specifications, components - Ballast section - Ties - Rail - Welds - Rail-tie fasteners - Special trackwork - Spirals and curves • Critical design criteria - minimum acceptable strengths and material properties	• Reduces risk of accidents due to failure of track structures or components

TABLE 5.10: TRACK INSPECTION AND MAINTENANCE

HSR SAFETY ISSUES—5—TRACK INSPECTION AND MAINTENANCE			
Issue	**Sub-Issues**	**Information Needed Regarding Regulations, Standards and Practices**	**Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)**
Track inspection and maintenance Maintenance and inspection needed to insure continuing safety in service	• Track strength - Panel shift - Buckling - Rail rollover • Track quality - Geometry - Rail flaw - Fastening security (not vibrate loose/out) • Strength of subgrade, fills, etc.	• Acceptable standards at different speeds for - Ballast - Ties and fasteners - Track geometry (alignment, warp, crosslevel, profile, gauge) - Rail flaw • Inspection methods and frequencies for geometry & rail flaw • Monitoring of fills, subgrade, etc. against failure • Maintenance practices & equipment • Post maintenance inspection and practices (especially speed restrictions after machine surfacing), weld inspection • Staff qualifications, training and experience	• Reduces risk of accidents due to the degradation of track structures or components

TABLE 5.11: SECURITY OF RIGHT-OF-WAY

HSR SAFETY ISSUES—6—SECURITY OF RIGHT-OF-WAY			
Issue	**Sub-Issues**	**Information Needed Regarding Regulations, Standards and Practices**	**Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)**
Security of right-of-way Physical protection of the right-of-way against hazards from the "external environment" including physical intrusion, vandalism and weather events	• Vandalism and trespassing • Grade crossing safety • Weather hazards - High winds - Snow - Temperature extremes • Shared right-of-way with conventional rail operations • Earthquakes • Protection against obstacles on track	• Grade crossing practice - Max. speed permitted - Protection system used at different speed levels • Fencing right-of-way • Shared right-of-way - Max. speeds in mixed traffic operation - Special precautions taken - Precautions against encroachment from adjacent tracks • Warning systems for intrusion, or foreign objects on track • Weather hazards and earthquakes - Warning devices used - Critical values & actions to be taken	• Reduces risks of - collisions with foreign objects on track or intruding on right-of-way - Grade crossing collisions, whether these cause a train derailment or not - Weather related track/ right-of-way accidents - Hitting person in right-of-way

TABLE 5.12: SIGNALS AND TRAIN CONTROL

HSR SAFETY ISSUES—7—SIGNALS AND TRAIN CONTROL

Issue	Sub-Issues	Information Needed Regarding Regulations, Standards and Practices	Relationship to Fault Tree (Types of Accidents/Incidents or Situations Affected)
Signals and Train Control			
A. Design and manufacture of signal and train control systems	• Use of cab signalling and automatic train control systems • Interlocking of signals and track circuit • Fail-safe verification • Maximum speeds for use of lineside signals • Manual override potential	• General description of train control system control features (vital and supervisory) • Headways • Train-track control center communication systems • Policy regarding speed thresholds at which cab signalling/ATC is required • Requirements for vehicle location detection (e.g., shunting resistance) • Inspection methods and frequencies	• Reduces risk of collisions or derailments due to signal malfunction or faulty design or installation • To the extent that automatic train control or operating features are present, the signal system reduces the risk of human error-caused collisions and derailments
B. Maintenance and inspection of signals and train control	• Inspection procedures • Staff qualifications and training • Quality control	• Staff training and qualification requirement • Quality control methods	

TABLE 5.13: WAYSIDE ELECTRIC TRACTION POWER SUPPLY

HSR SAFETY ISSUES—8—WAYSIDE ELECTRIC TRACTION POWER SUPPLY

Issue	Sub-Issues	Information Needed Regarding Regulations, Standards and Practices	Relationship to Fault Tree (Types of Accidents/Incidents or Situations Affected)
Electric Power Supply			
A. Design and construction to ensure safe operation	• Electrical clearance between catenary and structures • Grounding • EMI protection • Circuit-breaker performance • Insulation • Electric shock injuries	Specification details for each issue	•Reduces risk of: - Electric shock casualty to either employees or to the public - Significant EMI hazards Fires due to wayside power supply failure
B. Maintenance and inspection to insure continuing safety	• Deterioration of insulation, etc.	• Testing and inspection practices • Training or staff working on or near high voltage lines	

TABLE 5.14: OPERATING PRACTICES

HSR SAFETY ISSUES—9—OPERATING PRACTICES

Issue	Sub-Issues	Information Needed Regarding Regulations, Standards and Practices	Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)
Operating Practices Operational practices, required to assure safe operation	• Dispatching procedures • Brake test procedures • Train crew requirements • Prevention of unsafe actions by passengers • Emergency response procedures • Avoidance of alcohol and drug abuse • Special precaution for operations in tunnels • Use of engineer vigilance devices on locomotives and cab cars	• Applicable rules and practices for each issue - Normal operating rules - Permitted hours of service - Mandatory rest periods during and between shifts - Number of train crew - Procedures to avoid alcohol/drug abuse	• Reduces risk of collisions or derailments due to errors of train crew or signal and dispatching employees • Reduces risk of occupants of trains involved in derailments and collisions becoming casualties, through use of good emergency response procedures

TABLE 5.15: EMPLOYEE QUALIFICATIONS AND TRAINING

HSR SAFETY ISSUES—10—EMPLOYEE QUALIFICATIONS AND TRAINING

Issue	Sub-Issues	Information Needed Regarding Regulations, Standards and Practices	Relationship to Fault Tree (Types of Accidents/ Incidents or Situations Affected)
Employee Qualifications and Training Qualifications and training requirements for operating employees (train crew, signal operator and dispatcher) to minimize the risk of "human factors" caused accidents	• Engineers and train crew - Qualifications - Experience - Training - Route knowledge - Certification • Signal operators and dispatchers - Qualifications - Experience - Training	Details of the qualifications and training requirements for each group of employees, including any aptitude tests used and repeat training to maintain skills Also details of training to avoid personal casualties (hit by train, electric shock)	• Reduces risk of collisions or derailments due to operating employee errors • Reduces risk of employee casualties

5.2.2 Crashworthiness

The issue of crashworthiness and adherence to U.S. standards is of direct relevance to foreign tilt-technology vehicles which are typically light-weight in design. The primary structural standard used by all European railroads is UIC Code 566, in which structural strengths are generally much lower than FRA/AAR strengths, and the scope of definition is quite different. For instance, there is no UIC requirement for minimum vertical coupler or anti-climber force since buffers and screw-tensioned chain couplers (which cannot sustain vertical loads) are commonly used.

A review and comparison of FRA/AAR standards and UIC Code 566 is included in the previously cited FRA Report[36]. Highlights of the comparison are shown in **Table 5.16**, indicating the difference in requirements and in the scope of definition.

TABLE 5.16
COMPARISON OF CRASHWORTHINESS STANDARDS

Location/Load Type	FRA/AAR (Train Empty Weight > 272,200 kg (600,000 lb))	UIC Code 566 (OR)
Buff strength in line with coupler	355.9 kN (800,000 lb)	2,000 kN (449,000 lb)
Diagonal load at buffer level	Not specified	498 kN (112,000 lb)
Tensile force at coupler	-	1,500 kN (337,000 lb)
Collision posts, Number	2	Not specified
Shear strength	1,334 kN (300,000 lb) each (457 mm [18"] above coupler level)	400 kN (90,000 lb) (356 mm [14"] above coupler level)
At "center-rail" level (just below windows)	Not specified	298 kN (67,000 lb)
At "cant-rail" level (side to roof joint)	Not specified	298 kN (67,000 lb)
Truck to Body Shear Strength	1,112 kN (250,000 lb)	Typically 222 kN (50,000 lb) (function of car and truck mass)
Anti-climbing vertical Strength	445 kN (100,000 lb)	None
Vertical coupler strength	445 kN (100,000 lb)	None

[36] Ibid., FRA Report DOT/FRA/ORD-90/04

The AAR requirements, identical to the FRA regulations applicable to Multiple Unit locomotives, apply to all passenger cars operated in trains exceeding 272,200 kg (600,000 lb) lightweight. These standards have been adopted by Amtrak and all other providers of rail passenger service in the U.S. and Canada, although the AAR does not now formally issue passenger car standards. The loads indicated above must be sustained without permanent deformation of the car structure, except for collision post and truck to body shear loads, which must be sustained without total failure. Car specifications issued by operators of commuter and intercity rail service customarily require compliance with these standards; a structural test is normally required for any new design to confirm the buff strength standard, and design calculations must be submitted as evidence of compliance with other strength requirements.

The UIC Code 566 differs in definition and requires that car end walls strengthened by anti-collision pillars must be joined to the headstock (buffer beam) center rails and cant rails in such a way as to absorb collision energy and retain a high resistance to "override" shear forces. Specific strength or energy absorption requirements are not set for these. Truck-to-body shear strength force is a function of car and truck mass, typically in the range of 222 kN (50,000 lb).

The substantial differences between U.S. structural standards and those followed in the design of foreign tilt-body technology trains raises the question:

> "Under what circumstances, if any, can cars built to the UIC structural standards be operated in the United States?"

In the U.S. operating environment, decisions will depend critically on the degree of segregation of the high-speed tilt-train rail service from conventional U.S. passenger and freight operations. If not fully segregated, the high-speed trains will be sharing tracks with trains built to FRA/AAR standards over at least part of a route. If grade crossings are present on the route, there will be a significant risk of collision with a highway vehicle. There is also significant risk of intrusion onto the right-of-way, or presence of foreign objects on the tracks, leading to a collision. These risks are very real and have historically been a problem. Moreover, higher speeds lead to a need for greater energy absoption in structural deformation in the event of a collision. Research, test, and analysis will be required to determine the relationship between train weight, speed, strength, energy dissipation, and structural damage in accidents.

The question remains as to whether light-weight tilt-body trainsets such as the ETR-450, Talgo, or X2000 should be required to meet the buff strength standards of conventional U.S. railroading or should a different approach from current practice be undertaken. The choice of structural strength standards obviously has a direct impact on train weight (both sprung and unsprung mass), particularly the locomotive, and thus on high-speed tilt-train performance, cost and project viability. Major design changes may be necessary to meet the standard, not to mention the issue of higher vertical forces and wheel loads imposed by the increased sprung and unsprung mass. Increased vertical track forces could very well lead to unacceptable track degradation. The issue is whether foreign manufacturers of tilt-body trains will compromise their overall system performance and design for the higher loads and possibly accept a decrease in technical performance.

5.2.3 Regulatory Compliance

The existing FRA regulations, developed over decades in response to safety problems not solved by industry standards and practices, do not consider railroad operations in excess of 176 km/h (110 mph) or at more than 76 mm (3 in) of cant deficiency. Accordingly, they address specific issues discretely and do not treat whole railroads as integrated systems. That approach which has proven satisfactory thus far for conventional railroads, as evidenced by the remarkable safety record of the railroad industry in the last decade, appears to be in need of some modification for application to new tilt-body technologies such as the ETR-450 that are designed and operated as part of an integrated system having a significantly higher order of interdependent subsystems than conventional railroads.

Integrated, highly interactive, fault-tolerant systems invite regulatory treatment as a system. For example, the curve sensors, the on-board microprocessor network, the speed control system, and the braking systems for tilt-body technologies are so interdependent and interactive that the safety of any component of those subsystems can be fully understood only in context of the whole system. This may be difficult to achieve in a set of rules of general applicability, each of which governs one of those subsystems.

There is now no standard for fault-tolerant systems. How many components of such systems and what kinds of them may fail before a train is prohibited from leaving the terminal? How many components of such a system and what kinds of them may fail en route before a train is required to stop or proceed only at restricted speed to the nearest repair point?

Similarly, there is now no standard for the reliability of the computer hardware and software on which these systems rely. Moreover, many safety issues pertaining solely to passenger service have not been addressed by regulation. Instead, because Amtrak is the sole provider of intercity rail passenger service, those issues have been dealt with separately in the context of the special relationship between Amtrak and the FRA. (The Secretary of Transportation is a member of Amtrak's Board of Directors, appoints two of them, recommends candidates for the other positions to the President, holds all of Amtrak's preferred stock, and holds security interests in virtually all of Amtrak's equipment and real property). With new providers of intercity rail passenger service entering the market, it is highly desirable that passenger safety issues now be handled through rules of general applicability. It is clear that some additions to and modifications of some of the existing rules are needed.

Athough the FRA and Amtrak have worked out practical solutions pertaining to seat securement, luggage securement, equipment securement in dining cars, fire detection and suppression, and emergency training for passenger crews, no regulations currently exist. The FRA should not rely on attaining and maintaining the same sort of relationship with the management of each technology operator as FRA has with Amtrak.

The FRA track safety standards offer a somewhat different case in point. They now do not permit rail passenger operations at speeds above 176 km/h (110 mph) or at cant deficiencies excess of 76 mm (3 in). Amtrak operates at speeds up to 200 km/h (125 mph) on the Northeast Corridor under a waiver and will have to seek a similar waiver to operate tilt-body equipment at more cant deficiency than 76 mm (3 in). It seems undesirable to entertain a waiver petition every time a new high-speed or high cant deficiency service is contemplated. Amendments to the regulations setting standards for high-speed, high-cant deficiency passenger service seems to be in order, and a review of the power brake rule also seems appropriate. There is now no standard for the types of vital braking systems on which high-speed tilt-body technology systems typically rely.

Crashworthiness also merits new attention. Should light-weight tilt-body trainsets such as the ETR-450, Talgo or X2000 be required to meet the buff strength standards of conventional American railroading? Should there instead be some standard requiring controlled crushing to protect occupants of these trainsets? Should collision posts be required? Should there be an applicable anti-climb standard?

Buff strength is a measure of occupant compartment structural integrity. This measure is adequate for a particular type of car construction (body-on-underframe) and for low-speeds, when train buckling is not a great concern. Different vehicle structural designs may allow increased occupant compartment structural integrity and decreased vehicle weight.

The FRA currently is examining the issue of crashworthiness in a major study on *Collision Avoidance and Accident Survivability* scheduled for completion in 1992.

These subjects and the potential regulatory issues (in areas such as emergency preparedness, fire safety and equipment, and track inspection standards), many of which are quite complex, are underway and will take considerable time to address.

In addition, items not addressed in this technology-oriented report, such as environmental issues and personnel qualifications and training, will be the subject of other potential regulatory issues to be investigated in the future.

6. SUMMARY

The prospect of deploying tilt-train technology in the U.S. presents a number of challenges that must be met as a condition of success. Perhaps the most important of these challenges will be alteration of the attitudes toward complex vehicles and programmed preventative maintenance that have traditionally prevailed in the U.S. "railroad culture."

Active-tilt passenger vehicles are sophisticated and complex, and incorporate unfamiliar and often delicate components in critical subsystems. If these vehicles are to perform safely and reliably in commercial operation, there will have to be a major shift in the philosophy of vehicle and infrastructure maintenance, away from the traditional reactive practices of railroads and towards the aggressive programmed preventative maintenance followed by commercial aviation.

As well, there will have to be a significant expansion of management and labor skills, by operators and by regulators, to deal with the complex hydraulic components, sensors, and microelectronics essential to effective and reliable active body tilting. U.S. operators and regulators will also have to acquire the knowledge base required to deal effectively with ac traction motors, steerable trucks, active lateral suspensions and other elements of tilt-train design that are not part of the tilt mechanism, but that are essential components of the equipment.

Deployment of tilt-body equipment originally designed for conditions outside the U.S. may also require alteration of infrastructure maintenance practices. While alterations to alignment geometry may not be required, it is not clear whether changes to the measurement and maintenance of track geometry parameters will be needed. There are significant differences in the geometric standards adhered to by U.S. and foreign railroads, and indeed to the nature of the measurements upon which assessments of geometric conditions are based. Investigation of the behavior of key subsystems on U.S. track will be required to determine the extent to which either equipment design and/or track maintenance practices may need to be altered to replicate foreign performance, especially outside the Northeast Corridor.

U.S. application of tilt-body technologies will also pose a challenge to recognize the limits of what tilting can accomplish and to carefully avoid overstatement of the benefits to be gained, both within the management structure of operators and regulators, and among the travelling public at large. Body tilting is not a universal solution to the constraints on higher-speed operation on existing track. Its effectiveness will vary significantly from route to route. Creating unreasonable expectations which cannot be fulfilled can only work against the long-term success of the passenger rail mode.

The challenges noted above should not prevent selective application of tilt-body technologies. On some routes, active body tilting will offer a cost-effective mechanism to exploit market opportunities contingent on reduced trip time and improved ride quality. The other features incorporated in tilt technologies may also contribute significantly to overall improvement in the commercial performance of passenger rail.

However, the opportunity with the greatest potential is likely to be found beyond the scope of existing tilt-body technologies. Incorporation of active tilt mechanisms in high-speed (300 km/h+ [186 mph+]) wheel-on-rail or maglev systems could allow co-location of these technologies in some existing highway, rail and/or utility rights-of-way without unacceptable degradation of ride quality. In this regard, the active-tilt control strategy based on a digitized representation of the running surface alignment geometry and accurate and precise knowledge of vehicle location, appears to be especially promising. This technique effectively decouples tilt actuation from the (speed-sensitive) real-time detection of curve location and geometry. Application of this class of control system to maglev could enhance ride quality through transition curves and minimize the effect of constraints on alignment geometry on overall system performance.

APPENDIX A THE PHYSICS OF RAILROAD OPERATION ON CURVED TRACK

A.1 OBJECTIVES

The rationale for incorporating carbody tilting capability in a railroad passenger vehicle is quite straightforward: this technique permits maintenance of acceptable passenger ride quality with respect to lateral acceleration (and the consequent lateral force) received by riders when a vehicle traverses curved track at a speed in excess of the balance speed built into the curve geometry. (The balance speed of a specific track curve is the speed at which the centrifugal force induced by the track curvature is exactly balanced by the lateral component of gravitational force resulting from the superelevation or cant built into the track structure.) By tilting the body of a rail passenger vehicle relative to the plane of the track running surface, it is possible to operate at a higher speed than the balance speed without reducing the ride quality (in terms of lateral acceleration/force) perceived by passengers.

A.1.1 THE GEOMETRY OF RAILROAD TRACK

Before addressing the physics of curve negotiation and the principles of acceleration compensation, it is essential to establish the basic elements of railroad track geometry. There are two aspects to track geometry:

- o macroscopic elements that define the limiting conditions for the alignment as a whole (vertical and horizontal curvature, gradient), as shown in **Figure A.1** and **Figure A.2**; and

- o microscopic elements that define the orientation of the track at a specific location or for a short segment of an alignment, relative to a set of orthogonal axes (line, profile or level, gauge, superelevation or cant, warp or twist), as shown in **Figure A.3.**

Figure A.1 illustrates the elements of *route or alignment* geometry. A straight section is referred to as *tangent* track. Curves may be described either in terms of the number of degrees (the smaller the value, the shallower the curve), or in terms of the length of the radius (the larger the value, the shallower the curve). In North American practice, the degree of curvature is defined as the central angle subtended by a chord of 100 feet between two points on the centerline of the curve.

To reduce the rate of change in lateral acceleration (and thus force) between tangent and curved sections of track, all railroad tracks incorporate what is known as a *transition spiral* or *spiral*

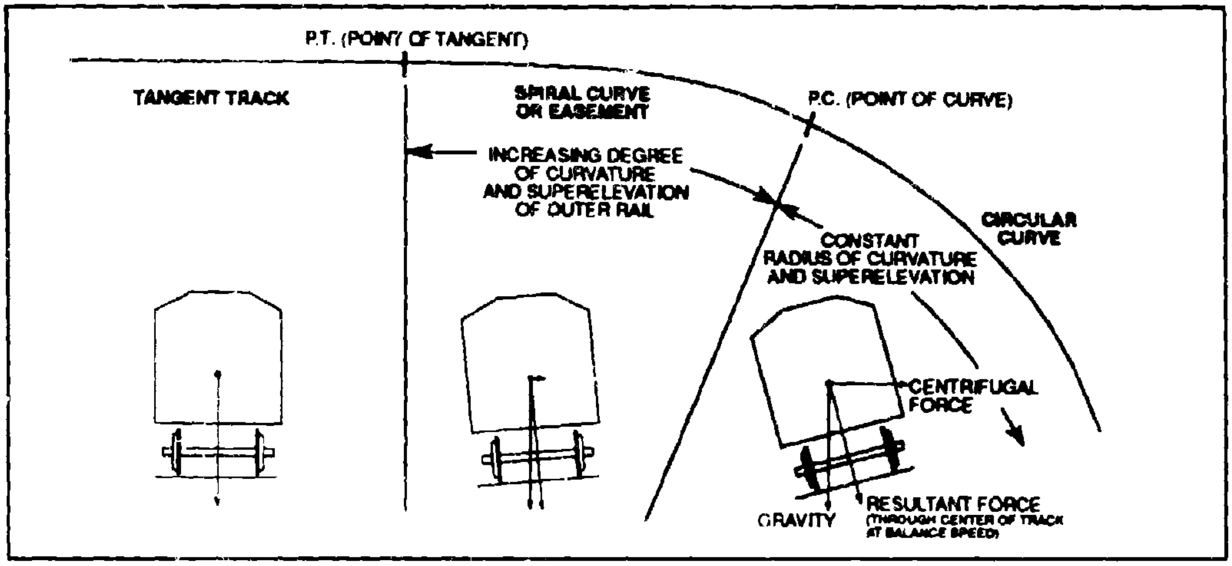

Figure A.1: Elements of Horizontal Alignment Geometry

easement, also as shown in **Figure A.1**. The transition spiral permits a gradual and controlled increase in curvature and superelevation (Section A.1.2), which serves to make the rate of change in accelerat[illegible] (and force) less noticeable, both by riders and in terms of the forces imposed on the track. In conventional (i.e., freight) railroad practice, the length of the transition spiral is driven primarily by the allowable rate of change in superelevation rather than curvature. The objective is to match superelevation with curvature throughout the transition.

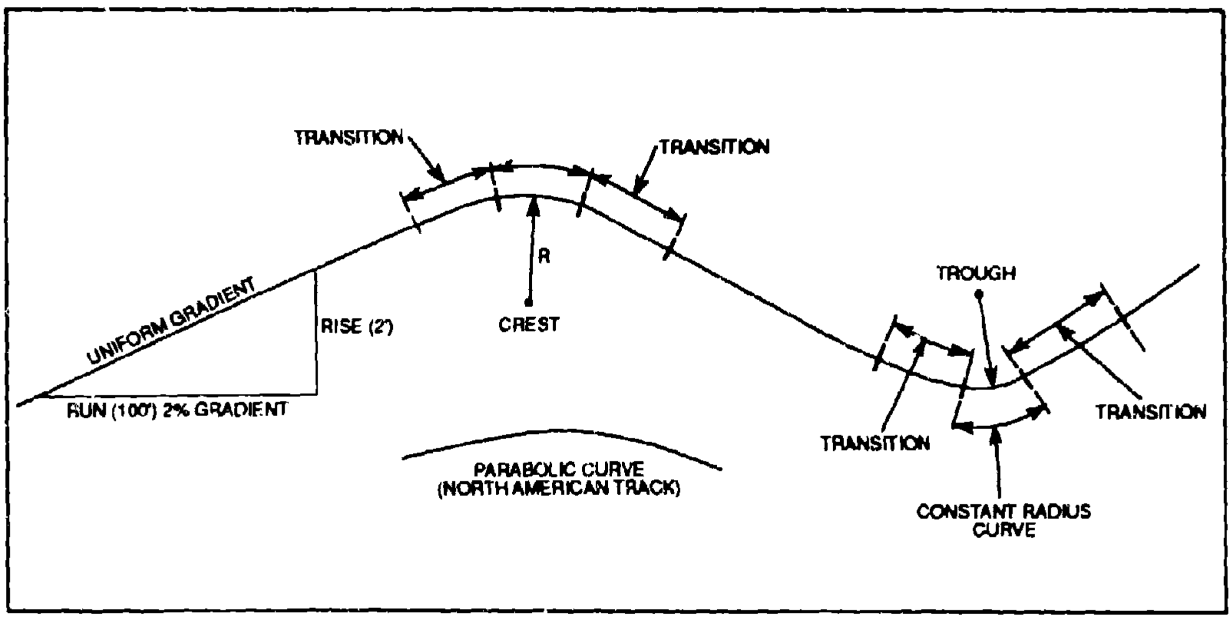

Figure A.2: The Elements of Vertical Alignment Geometry

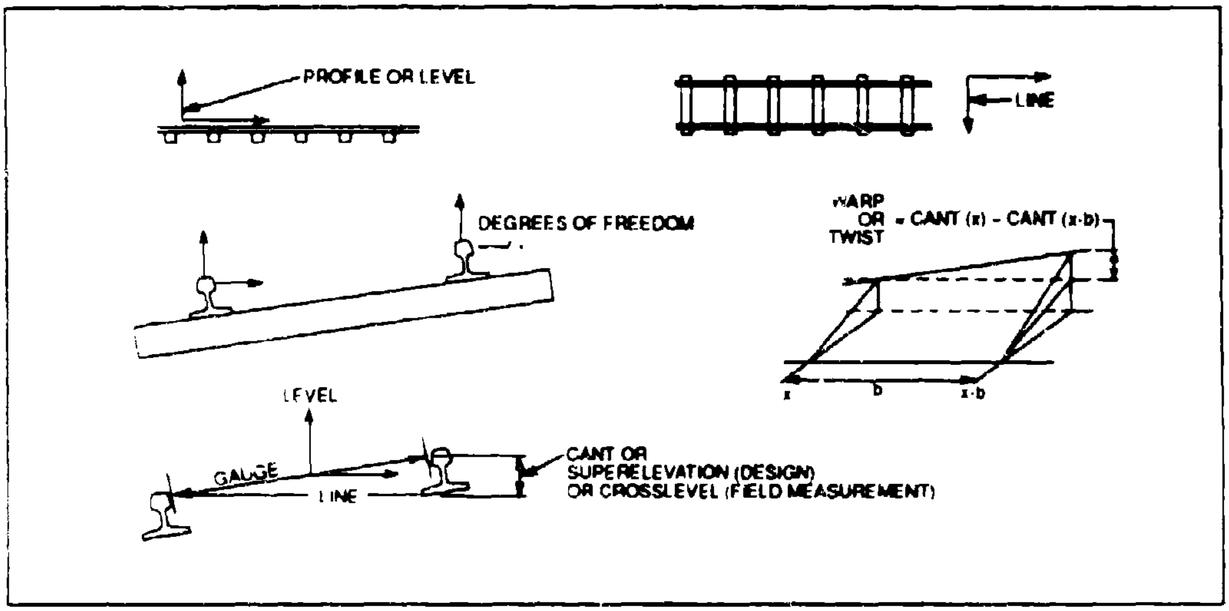

Figure A.3: The Elements of Local Track Geometry

In terms of vertical geometry (**Figure A.2**), the key measures are slope or gradient the rate at which the elevation of a track changes, and the radius of curvature at crest or trough. The gradient is usually measured as a percent (a 1.5% grade means a change in elevation of 1.5m for every 100m horizontally). Vertical curves may be described by degrees or radius of curvature, just as for horizontal curves. (Note that vertical curves, whether at crests or in troughs, also require transition spirals. Conventional railroad practice in North America is to design vertical curves as parabolas, rather than circular curves; parabolas provide an inherent transition from the uniform gradient line.)

The expectations of travellers with respect to comfort are the basis for most geometric limits. These limits are the levels of lateral and vertical acceleration, expressed as a proportion of gravitational acceleration (g), that have been shown to be acceptable to the majority of passengers - 0.08g to 0.10g for lateral and downward vertical accelerations, 0.05g for upward accelerations. Most passengers cannot detect accelerations of less than 0.04g.[1]

Because passengers are typically more sensitive to the unweighting sensation caused by traversing a crest at speed - the slightly unpleasant effect one feels when going over the top of a hill on a highway - the acceptable level of acceleration, and thus the minimum radius of curvature required at crests on track built especially for passenger service is larger than that

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[1] "Building the World's Fastest Railway," Andre Prud'homme, Railway Gazette International, January 1979; "The Development of a Truck for Narrow Gauge Line Limited Express Vehicles of Next Generation," Dr. S. Koyanagi, RTRI Quarterly Reports, V.26 No. 2, 1985; and "Tilt System for High-Speed Trains in Sweden," R. Persson, IMechE (Railway Division) Seminar on Tilting Body Trains, December 1989.

specified for troughs.[2] This design principle provides some additional margin of safety to ensure that the wheels of the trainset do not lose contact with the rail and so risk derailment.

In contrast, the AREA track standards that govern the design of lines intended for freight operations - virtually all track in the United States and Canada - require larger-radius curves in troughs than at crests, the exact opposite of the situation for tracks built to accomodate higher-speed passenger services. This is because the concern with the design of freight track is control of the behavior of cars and locomotives and the inter-vehicle forces, especially in long trains. In passing over a trough, the rear cars tend to crowd on to those in front, with a consequent sudden reversal in stress in the draft gear. As a result, troughs or sags are made more gradual.

Table A.1 summarizes typical values for horizontal curves, vertical curves and gradients for freight tracks, for the mixed-use Northeast Corridor in the U.S. and for high-speed passenger-service-only tracks constructed abroad.

The question of compatibility between the characteristics of existing rights-of-way with the geometric requirements of optimized HSGT infrastructure is very important. Most existing railroad routes in the U.S. reflect the requirements of freight railway operations. The general requirement for a freight alignment is that it minimize route length while permitting operation at a relatively slow but steady speed, with the maximum (controlling) gradient limited by the ability of equipment to start a heavy train from a standing start.

Figure A.3 illustrates the elements of local geometry of railroad track. In U.S. practice, these measures are:

- *line* (the longitudinal alignment of the track in the horizontal plane, relative to a surveyed datum),

- *profile or level* (the longitudinal alignment of the track in the vertical plane, relative to a surveyed datum),

- *gauge* (the distance between the inner faces of the running rails, by convention measured at a point 15.9mm [0.625"] below the top of the rail),

- *superelevation or cant* (the **nominal** or **design** difference in vertical elevation between the heads of the two rails; for the actual measured difference at a given point on track, North American railroaders use the term *crosslevel*), and

[2] Ibid, Prud'homme.

TABLE A.1
TYPICAL GEOMETRIC VALUES FOR FREIGHT AND DEDICATED PASSENGER TRACK ALIGNMENTS

CHARACTERISTIC	Typical North American Freight Line (No Passenger)	North-East Corridor** 200 km/h Passenger (125 mph) 80 km/h Freight (50 mph)	JR, Shinkansen, New Tokaido 200 km/h (125 mph) Passenger Only	France TGV 300 km/h (186 mph) Passenger Only
Horizontal Curvature	198 to 1737 m (650 to 5700 ft)	436 to 1737 m (1430 to 5700 ft)	1890 to 2500 m (6200 to 8200 ft)	6096 m (20 000 ft)
Vertical Curvature	See Note *	See Note *	Crest 14,940 m 49,000 ft) Trough 10,060 m 33,000 ft)	(minimums) Crest - 16,000 m (52,490 ft) Trough - 14,000 m (45,930 ft)
Gradient (%)	Less than 1%	1.5 to 2.0	1.5 to 2.0	2.5 to 3.5

* AREA 1990 Manual for Railway Engineering, Vol 1, Ch.5, Part 3, page 5 3.13, expresses standard for vertical curvature in terms of allowable maximum rate of change of gradient in feet per 100 feet of curve length. The recommended limits are 0.05 for troughs and 0.10 for crests (i.e., to go from a 1 gradient to level track would require [1/.05] x 100, or 2000 feet). This defines a parabola rather than a constant-radius curve.

** Amtrak Northeast Corridor Operating Rules and Instructions, October 1989; Amtrak 1987 Track Chart.

- *warp or twist*, which is the difference in superelevation measured at two points on the track (usually over a distance of 9.45m or 18.9m [31 or 62 feet] in North America, but ove. approximately 30m [100 feet] elsewhere). In U.S. practice, the actual measurements are taken between the two extremes of crosslevel found within the specified distance.

In the context of this report, the amount of superelevation, the rate of change of superelevation, and the effects of these parameters on passenger comfort and operational safety are of principal concern. The other elements are important to the overall quality of the track, and ultimately to the ride perceived by a passenger in a train using that track and to the safety of train operation, but consideration of these elements lie outside the scope of this report.

A.1.2 TRANSITION CURVE GEOMETRY AND VEHICLE RIDE QUALITY

Passive tilting systems respond exclusively to the force induced when the vehicle moves through the transition curve. The geometry of the transition curve directly affects both the rate of change in the tilt angle (and higher-order derivatives) and the magnitude of the tilt angle while in the transition curve. Accordingly, the vehicle ride quality with respect to lateral motion is directly dependent on the geometry of the transition curve.

For active tilting systems, the situation is slightly different. These systems typically sense lateral acceleration at the vehicle truck, and/or the track superelevation, as input signals to the tilting system controller. Accordingly, the vehicle ride quality will be influenced by the transition curve geometry to the extent that these signals affect the rate and magnitude of tilting motion through the system controller.

Early railroad track alignments, laid out when very modest train speeds prevailed, connected straight track to fixed radius curved track without the use of a transition curve or incorporation of superelevation. The corresponding variation in track curvature, in lateral acceleration and in rate of change of lateral acceleration or lateral *jerk* are shown in **Figure A.4**. In practice, the lateral jerk impulse of infinite amplitude shown in **Figure A.4** as a consequence of the lack of any transition curving would be limited to some finite but still very large amplitude by the compliance of the lateral suspension system of the vehicle. This spike of lateral jerk at the interface between straight and fixed radius curving track resulted in very high passenger discomfort levels as train speeds increased.

Initially, this transition alignment problem was addressed by introducing multiple-arc or compound transition curves for run-in and run-out between fixed radius curves and straight track. Such compound curves were composed of a series of interconnecting circular arcs of decreasing radii. Although the amplitude of the jerk impulse was significantly reduced, there were now a number of jerk impulses through the transition curves, which still seriously degraded passenger ride quality.[3]

This situation led to the introduction and the eventual widespread application by most of the world's railways of track transition curves having continuously changing curvature and, if applicable, superelevation between zero at the straight track end of the transition curve and that of the fixed radius of curvature at the other end. A linear variation of curvature and superelevation over the transition curve (i.e., a constant rate of change of curvature and superelevation), as shown in **Figure A.5**, with the corresponding lateral acceleration and lateral jerk variations, was widely adopted. This transition curve alignment geometry is known as a *clothoid spiral*.

It is evident from **Figure A.5** that the jerk impulse(s) of the earlier curve alignments is eliminated with the clothoid transition curve. However, a sudden discontinuity in lateral jerk is still present at the beginning and end of the transition curve. That is, as the transition is entered, the lateral acceleration instantly begins increasing at a linear rate, continuing to increase until the constant radius curve is reached, where it instantly stops increasing; the lateral jerk (the rate of change of lateral acceleration) instantly changes from zero to this linear value at

[3] "American Railway Engineering Association: Length of Railway Transition Spiral - Analysis and Running Tests," Proceedings of A.R.E.A., Vol. 65, 1964.

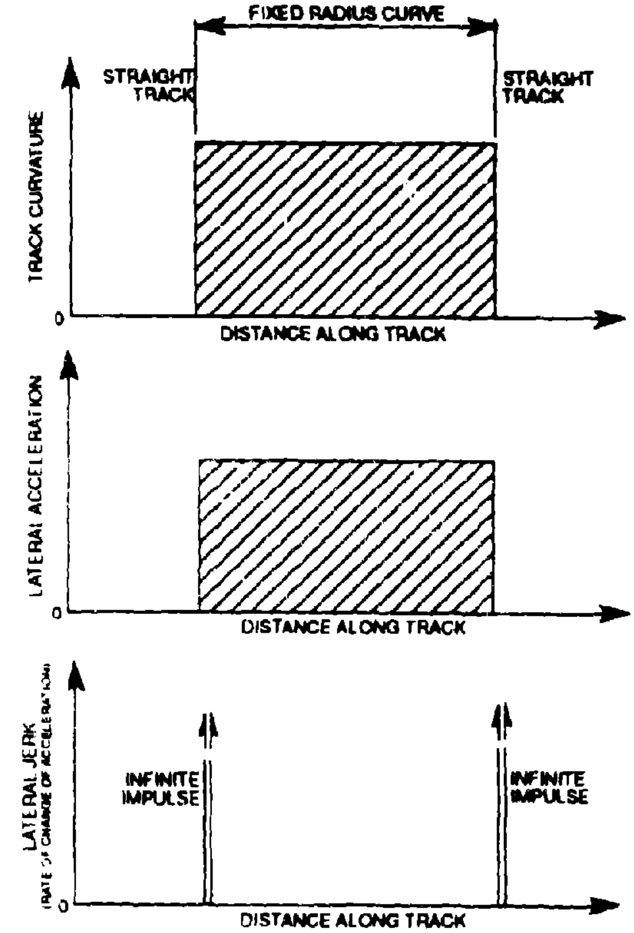

Figure A.4: Lateral Acceleration In Absence of Transition Curve

transition entry and instantly returns to zero when the lateral acceleration no longer changes. (In mathematical terms, the first derivative or rate of change of the track curvature with respect to distance along the track is discontinuous at the transition endpoints). These lateral jerk discontinuities are the major cause of degradation of passenger ride comfort, particularly at high speed.

The clothoid transition curve is frequently approximated in railway practice by a "cubic parabola" curve alignment for purposes of convenience of track transition curve layout in the field. This curve provides a very close approximation of the clothoid transition curve. A very large proportion of North American mainline railroad tracks incorporates transition curves laid out using the cubic parabola formula, and with linearly-increasing superelevation for all but the gentlest of curves. The length of these "clothoid" transition curves and accordingly, the amplitude of the lateral jerk impulses for any given connecting fixed radius curve are dictated by the practice at the time of construction and the subsequent operational history of the line.

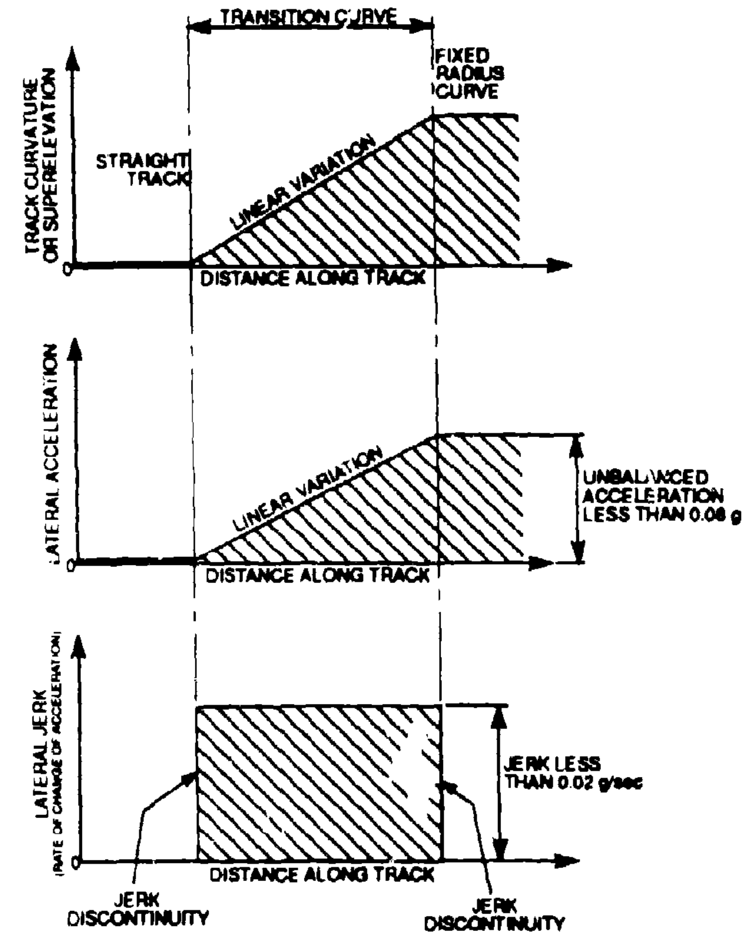

Figure A.5: Lateral Acceleration with Clothoid Spiral Transition Curve

Transition curve alignments which eliminate the start and end lateral jerk discontinuities of the clothoid transition curve are specified for track dedicated to very high-speed passenger services. In these alignments, both the change in curvature and the rate of change of curvature are made to be continuous at both the transition entry and exit locations. Such dedicated high speed track alignments already exist in Japan and France and are being extended in both countries as well as into other parts of Europe. Dedicated high-speed railway lines are also being considered for a number of North American corridors, most notably in Texas.

One proven high-speed transition curve is characterized by curvature variation along the curve length defined by a trigonometric half-cosine function, as shown in **Figure A.6** along with the corresponding lateral acceleration and lateral jerk variation.

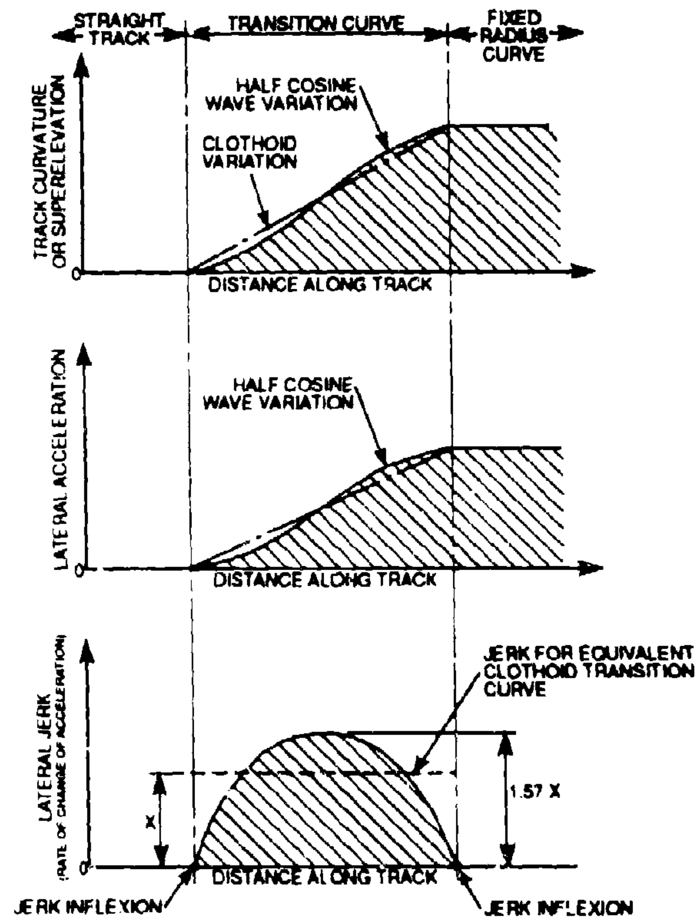

Figure A.5: Lateral Acceleration with Shinkansen High-Speed Transition Curve

This curve geometry eliminates the lateral jerk discontinuities at the start and the end of the clothoid transition curve. This type of transition curve is used for the track alignment of the Shinkansen in Japan, where it is referred to as a "half sine wave" transition curve.[4] However, this transition curve geometry increases the maximum lateral jerk at the midpoint of the curve length by a factor of about 1.6, relative to the magnitude of the constant lateral jerk for an equivalent clothoid transition curve. This increase in maximum lateral jerk for the half-cosine transition curve is generally considered to be much less significant in terms of its effect on perceived ride quality than the lateral jerk discontinuities of an equivalent clothoid transition curve.

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[4] "Analysis of Relationship Between Transition Curve Profile and Railway Vehicle Vibration," S. Hashimoto, Quarterly Reports of the Railway Technical Research Institute, Japanese National Railways, Vol. 30, No. 4, November 1989.

Additional improvement in the lateral jerk characteristics of transition curves could be realized by the appropriate superimposing of a trigonometric full sine wave onto the linear curvature variation of the clothoid transition curve, as shown in **Figure A.7.**[5]

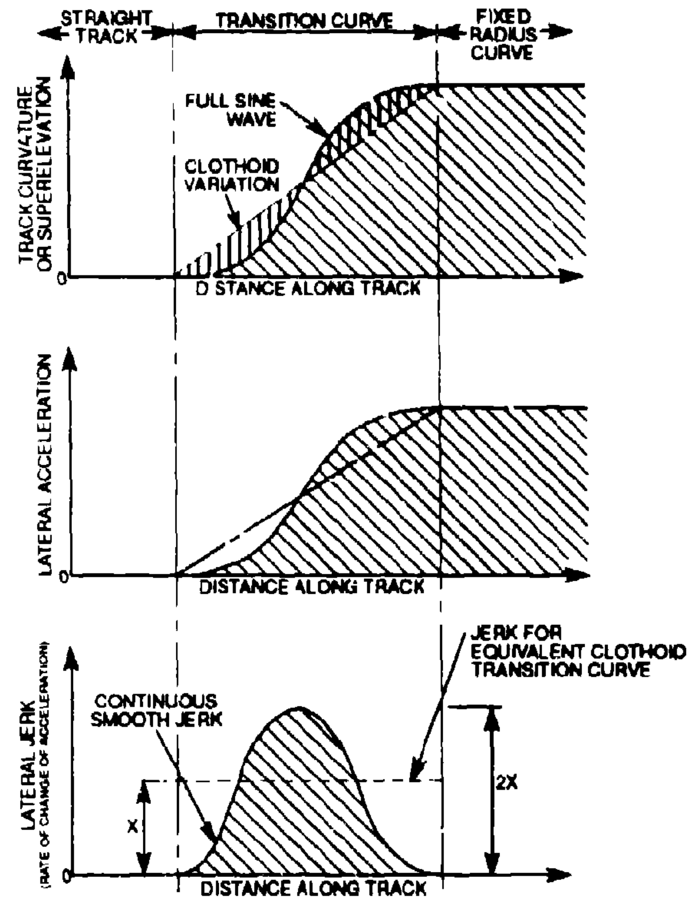

Figure A.7: Lateral Acceleration With Full Sine Wave/Clothoid Hybrid Geometry

This combination of sine wave/clothoid alignment geometry results in the lateral jerk not only being continuous over the length of the transition curve but also having zero slope at the start and end of the curve length. (That is, the curvature and the first and second derivatives of track curvature with respect to length along the track are continuous at the transition end points). However, this type of transition curve increases the maximum lateral jerk half way along the curve length by a factor of two relative to the magnitude of the constant lateral jerk for an

[5] "Railway Transition Curve Planning Methods," J. Gubar, Rail International, pp 31-43, April 1990.

equivalent clothoid transition curve, although this again is considered less damaging to ride quality than the sudden change in jerk for an equivalent clothoid transition curve.

Techniques for optimizing railway vehicle passenger ride comfort through transition curves by means of the active carbody tilting, not surprisingly, attempt to duplicate the lateral acceleration and lateral jerk variations of the full sine wave/clothoid transition curve geometry. Strategies for achieving this goal are examined in Appendix B of this report.

A.2 THE PHYSICS OF CURVE NEGOTIATION

Consider a railroad vehicle that is moving along a straight, level track at a constant speed, as shown in **Figure A.8**. The vehicle has considerable inertia, and is subject to the downward force of gravity (g) and the equal but opposite supporting force exerted by the track structure and subgrade, but is not subject to any lateral forces.

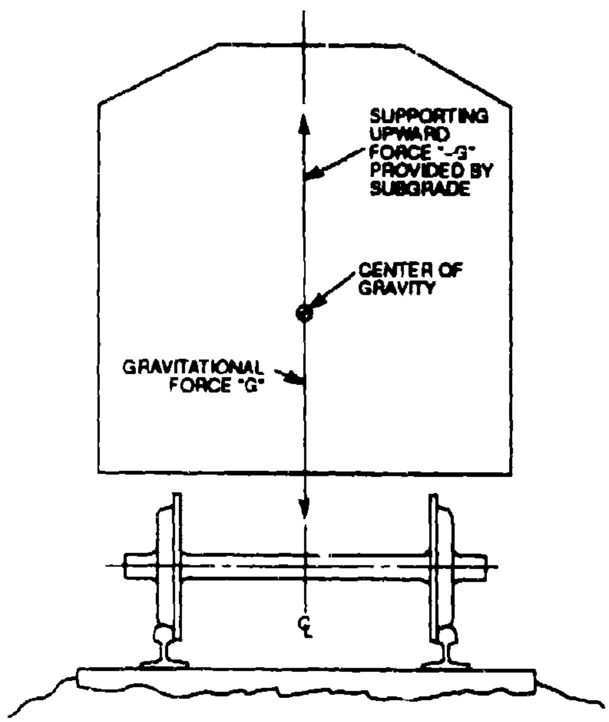

Figure A.8: Forces Acting on a Rail Vehicle on Level Tangent Track

As the vehicle enters a curved section of track (ignoring for the moment the question of transition between straight [tangent] track and curved track), deflection of the vehicle from its previous path requires application of a lateral force acting inward toward the center of the curve, as shown in **Figure A.9**.

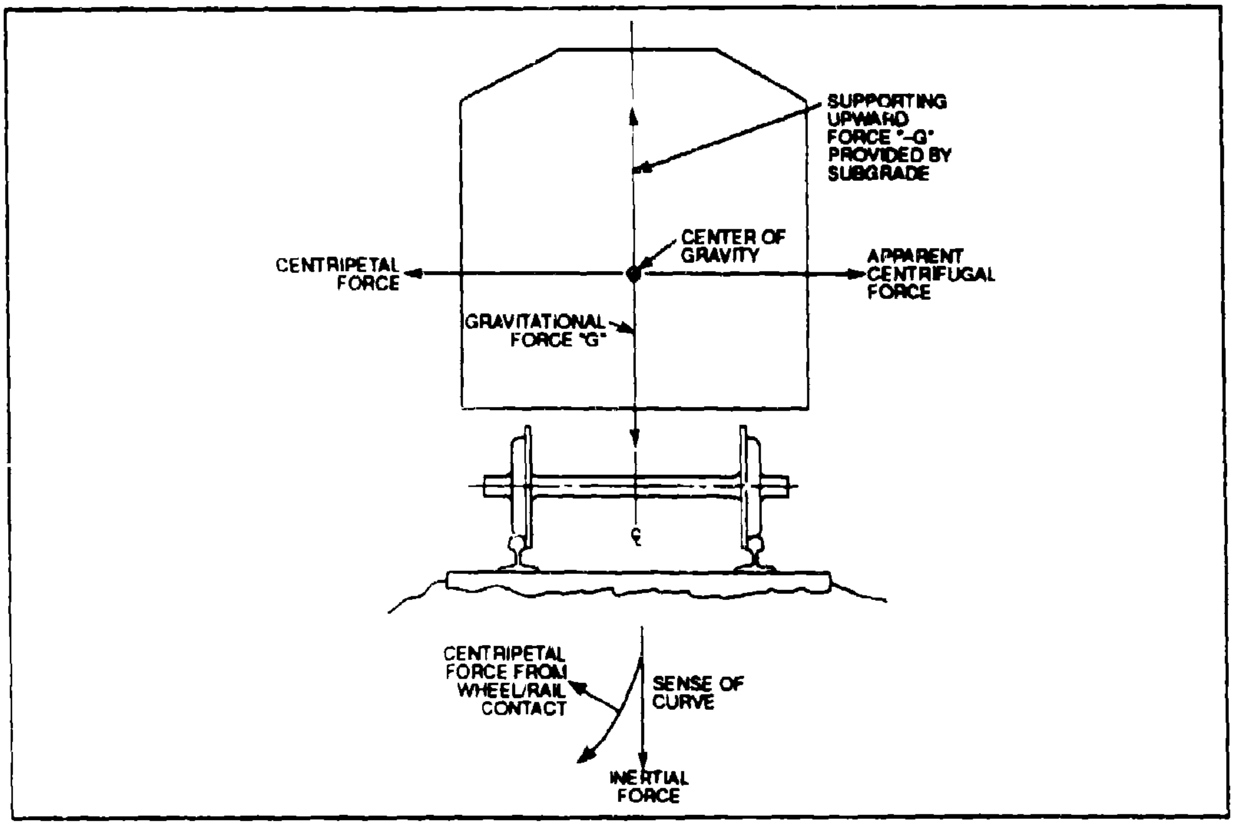

Figure A.9: Forces Acting on Rail Vehicle During Curve Negotiation

This real *centripetal force*, must, under Newton's Second law, be balanced by an "equal but opposite" force. This is termed the *centrifugal force*, which appears to act outward from the center of the curve.

For a passenger riding in a rail vehicle traversing a curve of radius "**R**" at a constant speed "**v**", there will be an apparent *centrifugal acceleration* "**a**", acting laterally outwards, and a corresponding lateral force, **F = ma**, where **m** is the mass of the passenger. The magnitude of the apparent centrifugal acceleration is given by:

$$\mathbf{a} = \mathbf{v}^2/\mathbf{R} \qquad \text{(A.1)}$$

where **v** is the vehicle velocity along the curve, and **R** is radius of curvature at the point where the acceleration is being measured (this is termed the "local" radius of curvature, and is important because the radius of curvature varies constantly in the transition sections between tangent and circular-curve track sections). The corresponding *centrifugal force* acting radially outwards is given by:

$$\mathbf{F} = \mathbf{m}\mathbf{v}^2/\mathbf{R} \qquad \text{(A.2)}$$

As an example, a passenger weighing 125 kg (276 lb) riding in a vehicle travelling at 100 km/h (62.5 mph) around a track curve with a radius of 500 m (1,640 ft) would be subjected to a centrifugal force of just under 193 Newtons (43.4 pounds force) acting radially outwards from the curve center.

As noted above, it is generally accepted in passenger railway practice in Europe and Japan that passengers will tolerate a sustained lateral acceleration of about 0.08 g, which corresponds to an applied lateral force of 8% of the passenger's mass, without noticeable discomfort. This steady-state lateral force comfort limit is generally consistent with the low frequency vibration comfort limit from the ISO Standards for Human Exposure to Mechanical Shock and Vibration.[6]

This standard indicates a whole-body vibration exposure lateral acceleration limit of 0.12 g at a frequency of 1-2 cycles per second for up to six minutes duration.

Extrapolating the lateral acceleration limits given by this standard to the sustained lateral acceleration limit of 0.08 g as generally used in passenger railway practice, the corresponding frequency of vibration would be about 0.2 cycles per second, which is close to a steady-state condition, and thus consistent with the 0.08 g sustained lateral acceleration limit used in practice.

The angular orientation of the rails relative to the horizontal plane, **A**, is the angle of superelevation or "track cant angle" and is given by[7] (see **Figure A.3**):

$$\sin \mathbf{A} = \mathbf{h/s} \qquad (A.3)$$

where s = track gauge, and
h = track superelevation

Inclusion of superelevation in curved track reduces or (at the balance speed of the curve) eliminates the effect of centrifugal force on passengers by compensating the curving force with a component of the gravitational force acting on the passenger. This principle of *acceleration compensation* is demonstrated in **Figure A.10**, showing the carbody forces.

With reference to **Figure A.10**:

> if $G = \mathbf{mg}$ is the gravitational force acting vertically downward on a passenger of mass **m**, then $\mathbf{F} = \mathbf{mv^2/R} = \mathbf{Gv^2/gR}$ is the centrifugal force acting radially outwards from the curve center, where:

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[6] International Standard 2631, Guide for the Evaluation of Human Exposure to Whole-Body Vibration

[7] or A=h/s (radians), when the angle of superelevation is relatively small

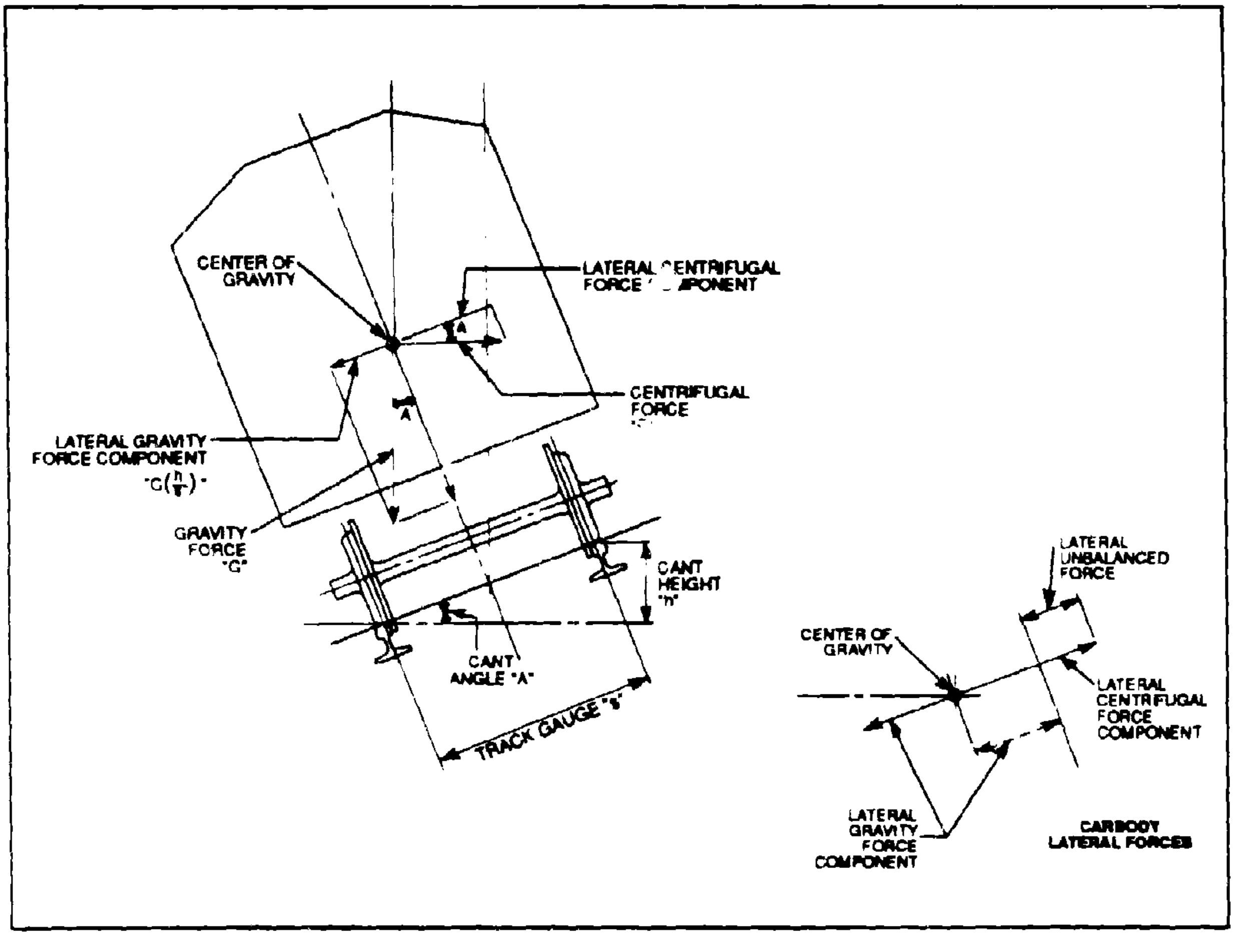

Figure A.10: Acceleration Compensation from Superelevation in Curves

g = acceleration due to gravity;
G = passenger weight. **m** = passenger mass;
v = railway vehicle velocity;
R = local radius of curvature of track; and
h and s are as defined above.

It is evident from the vector diagram in **Figure A.10**, which represents the components of the centrifugal and gravitational forces **F** and **G**, that when these forces act on a passenger in a rail vehicle traversing a curve with superelevation, the lateral force acting on a passenger relative to the tilt orientation of the vehicle will be completely compensated (cancelled out) when[8]:

$$\mathbf{G \sin A = F \cos A} \qquad (A.4)$$

[8] or G(h/s) = F if the angle of superelevation is small

This balanced lateral force condition can be rewritten in terms of the vehicle velocity at this condition, $\mathbf{V}^*$ and the local radius of curvature, **R**:

$$\mathbf{G(h/s) = mV^{*2}/R} \qquad \text{(for small angles } \mathbf{A}\text{)} \qquad \text{(A.5)}$$

Accordingly, for any given track curve of radius **R**, track gauge s and track superelevation **h**, there will be a single unique vehicle speed, $\mathbf{V}^*$, for which the lateral component of the centrifugal force (relative to the vehicle tilt orientation) will be exactly compensated by the corresponding component of the gravitational force.

This vehicle speed is referred to as the *balance or equilibrium speed* for a curve with a given set of geometric characteristics, and is given by:

$$\mathbf{V^* = \sqrt{Rg(h/s)}} \qquad \text{(A.6)}$$

The traversing of a given curve at a speed either higher or lower than the balanced speed results in an unbalanced lateral force being exerted on a vehicle and its occupants, as shown in **Figure A.11**. This resultant unbalanced force, $\mathbf{F_y}$, is given by:

$$\mathbf{F_y = (V^2/R) - (gh/s)} \qquad \text{(A.7)}$$

When discussing operation on curved track at speeds in excess of the balance speed, it is common railway practice to refer to the difference between the amount of superelevation actually in the curve and the amount that would be required to increase the balance speed to the actual operating speed. This difference in superelevation is termed *cant deficiency*, or uncompensated superelevation. *Cant deficiency* is defined as the difference between the track superelevation for which the vehicle traversing a given curve at a given speed would experience no unbalanced lateral force, and the actual amount of superelevation incorporated in the geometry of the curve. *Cant deficiency* is a convenient way to measure the degree of unbalance speed operation, in that it relates directly to the amount of vehicle carbody tilting which would be required to exactly compensate for the unbalanced force so as to maintain a uniform level of passenger comfort.

Determination of the amount of angular inclination of the carbody required to compensate for lateral force while traversing a superelevated curve at other than the balance speed is complicated by the action of springs in the vehicle suspension.

When a curve is traversed at some speed above or below the balance speed, the resultant unbalanced lateral force acting on the centre of gravity of the vehicle will tend to tilt the vehicle further in the direction of the unbalanced force, effectively amplifying that force. This is caused by application of a roll torque to the carbody which compresses the springs in the vehicle suspension, as shown in **Figure A.12**.

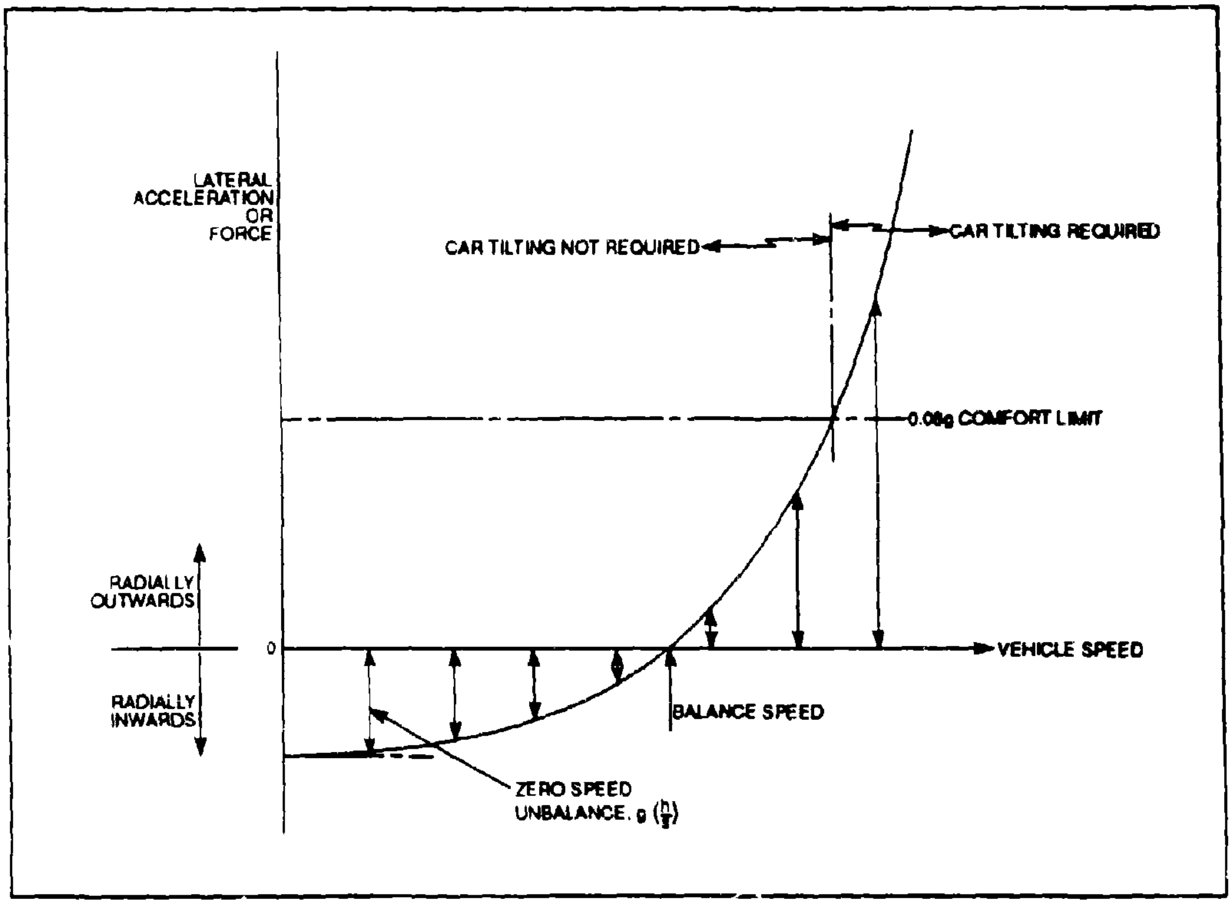

Figure A.11: Effect of Speed on Acceleration Compensation

The softer the springs in the vehicle suspension, the greater the amplification of the original unbalanced force. This effect on carbody inclination angle typically is sufficiently large that it must be taken into account in determining the performance of vehicle tilting systems. It is evident from **Figure A.12** that tilting due to differential compression of the vehicle suspension has the same effect on passenger ride comfort as changing the amount of superelevation in a given curve by the amount of angular rotation of the vehicle body relative to the angular rotation of the plane of the rail running surface.

It is convenient to express this vehicle carbody tilt angle, **B**, in terms of a negative amount of superelevation[9]:

$$\tan B = -w/s \qquad (A.8)$$

where s = track gauge, and **w** = negative superelevation tilt equivalent.

[9] or B = -w/s (radians) where the superelevation is small

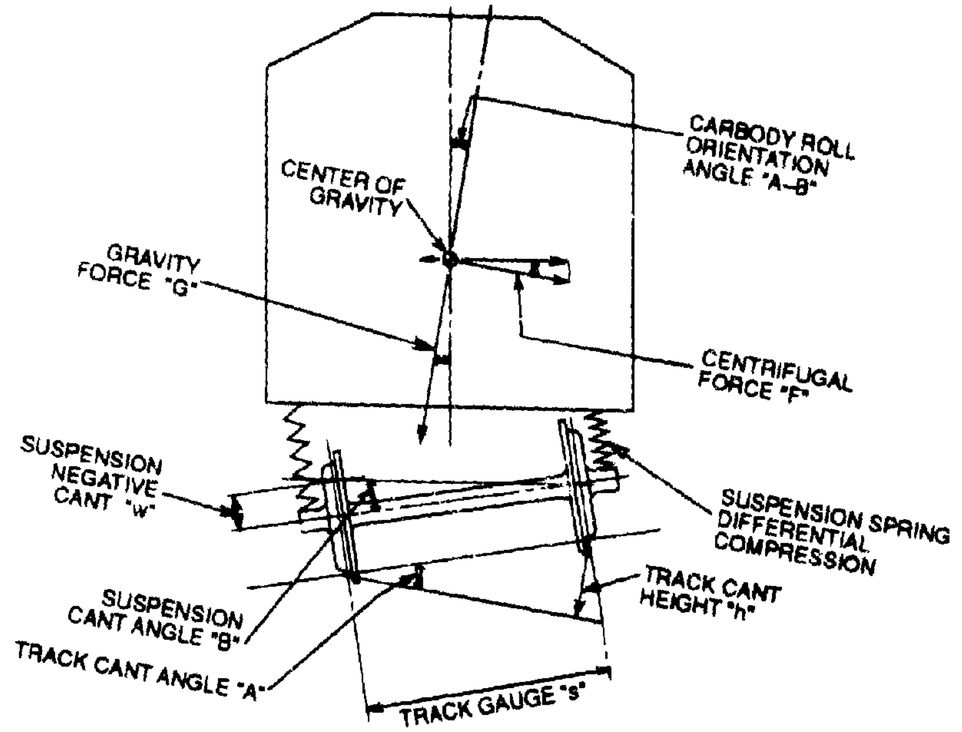

Figure A.12: Effect of Suspension Compression

By adding intentional carbody tilt to that produced by superelevation and differential compression of the vehicle suspension, it is possible to reduce or eliminate an unbalanced lateral force acting on passengers, as shown in **Figure A.13**. Intentional tilting of the carbody has the same effect, with respect to passenger ride comfort, as changing the amount of superelevation in a given curve by the amount of angular rotation of the vehicle body relative to the plane of the rail running surface. As with differential suspension compression, it is convenient to express the vehicle car body tilt angle, **C**, in terms of a superelevation equivalent. In this instance[10],

$$\tan C = u/s \qquad (A.9)$$

where s = track gauge; and
u = carbody tilt superelevation equivalent

[10] or $C = u/s$ (radians), where the amount of superelevation is small

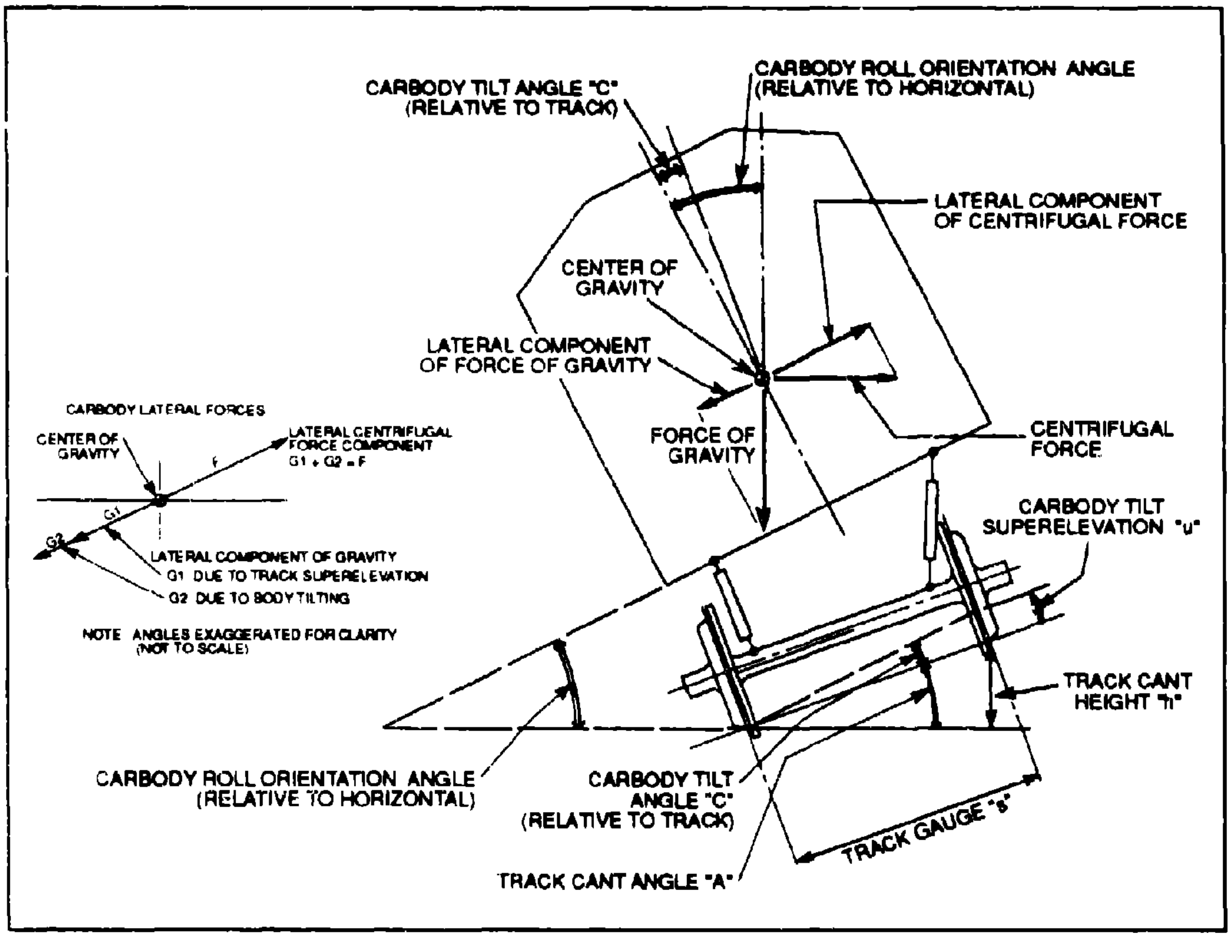

Figure A.13: Effect of Deliberate Body Tilting on Force Balance

Incorporating both deliberate carbody tilt and the roll induced by differential compression of the vehicle suspension into the expression for balance speed yields the balanced speed equation applicable to tilting railway vehicles:

$$V^{**} = \sqrt{(h-w+u)Rg/s} \qquad (A.10)$$

Traversing a curve at speeds either higher or lower than this balanced speed results in an unbalanced lateral force, F_y , relative to the tilt orientation of the vehicle. Incorporating the effects of differential compression of the vehicle suspension and deliberate body tilting into the expression for unbalanced force gives the complete unbalanced force equation for tilting railway vehicles:

$$F_y = (V^2/R) - g(h-w+u)/s \qquad (A.11)$$

APPENDIX B ACTUATION TECHNIQUES AND CONTROL STRATEGIES FOR TILTING OF RAILWAY VEHICLE CARBODIES

B.1 FUNCTIONAL REQUIREMENTS

Achievement of controlled tilting of a railway vehicle carbody requires hardware and software to fulfill the functions shown schematically in **Figure B-1**:

- o Rotation of the vehicle carbody about an appropriately-located roll center, by means of suitable mechanical linkages or roller arrangements between the carbody and the truck (bogie) of the vehicle, for either active- or passive-tilting systems;

- o Command-actuated angular rotation of the vehicle body about the roll center, for active-tilting systems; and

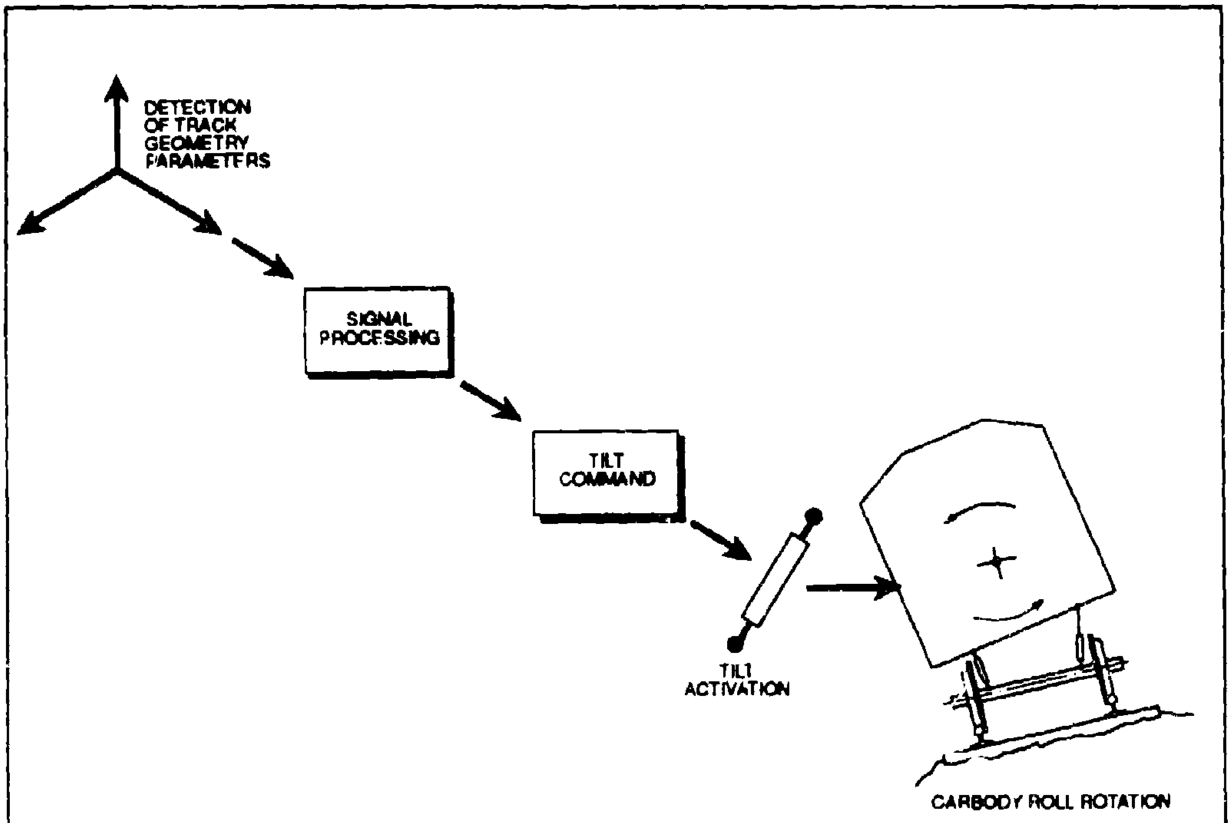

Figure B-1: Schematic Representation of Functional Requirements for Body Tilting

- Continuous and reliable detection and analysis of local track geometry parameters to determine the magnitude and rate of carbody tilt required to compensate for lateral curving forces, and to generate the command signals required to control the tilt mechanism (for active-tilting systems).

B.2 MECHANISMS FOR CARBODY ROLL ROTATION

There are several types of mechanical linkages between the carbody and the truck that have been used by developers of tilt-body technologies to provide carbody roll rotation. The simplest linkage arrangement is comprised of just two connecting links between carbody and truck. These links are connected by hinges to the body and truck, so as to define a carbody roll center that falls on the centerline of the untilted vehicle and at an appropriate point relative to the center of gravity of the vehicle, as shown in **Figure B-2**.

This type of simple two-link mechanical connection has been incorporated into the Fiat ETR 450 active-tilt EMU, the ABB active-tilt X2000 equipment, and the VT610 active-tilt DMU sets now under construction for German Federal Railways (DB), as detailed in Appendix C. This mechanism also formed the basis for active-tilting in the Advanced Passenger Train (APT) prototypes developed for British Rail but ultimately cancelled prior to fleet deployment.

The Bombardier-built LRC active-tilt coaches use a more complicated linkage arrangement, as shown in **Figure B-3** and detailed in Appendix C. This mechanism incorporates a bell crank with two pairs of connecting links; this minimizes the volume of space required for the mechanism and provides lateral displacement of the carbody in conjunction with body roll, so as to enhance vehicle rollover stability and minimize the dynamic clearance envelope while traversing a curve.

The Japanese Railways Series 381 passive-tilting EMUs differ with respect to the mechanism used to achieve carbody roll rotation. These trainsets incorporate truck-mounted rollers which support a cross-beam mounted on the carbody. The lower surface of the cross-beam, which rests on the rollers, takes the form of a constant-radius curve, as shown in **Figure B-5** and detailed in Appendix C.

The Talgo *Pendular* passive-tilting coaches operated by Spanish National Railways (RENFE) incorporate another, quite distinct mechanism for carbody roll rotation. A vertical supporting structure mounted on the articulated trucks and located within the ends of each carbody carries the secondary suspension air bags, which are housed within roof-level pockets in the body end structure, as shown in **Figure B-4** and detailed in Appendix C. The differential compression of the suspension air bags through curves provides for roll rotation of the carbody relative to the trucks, but with a roll center close to the roof of the carbody.

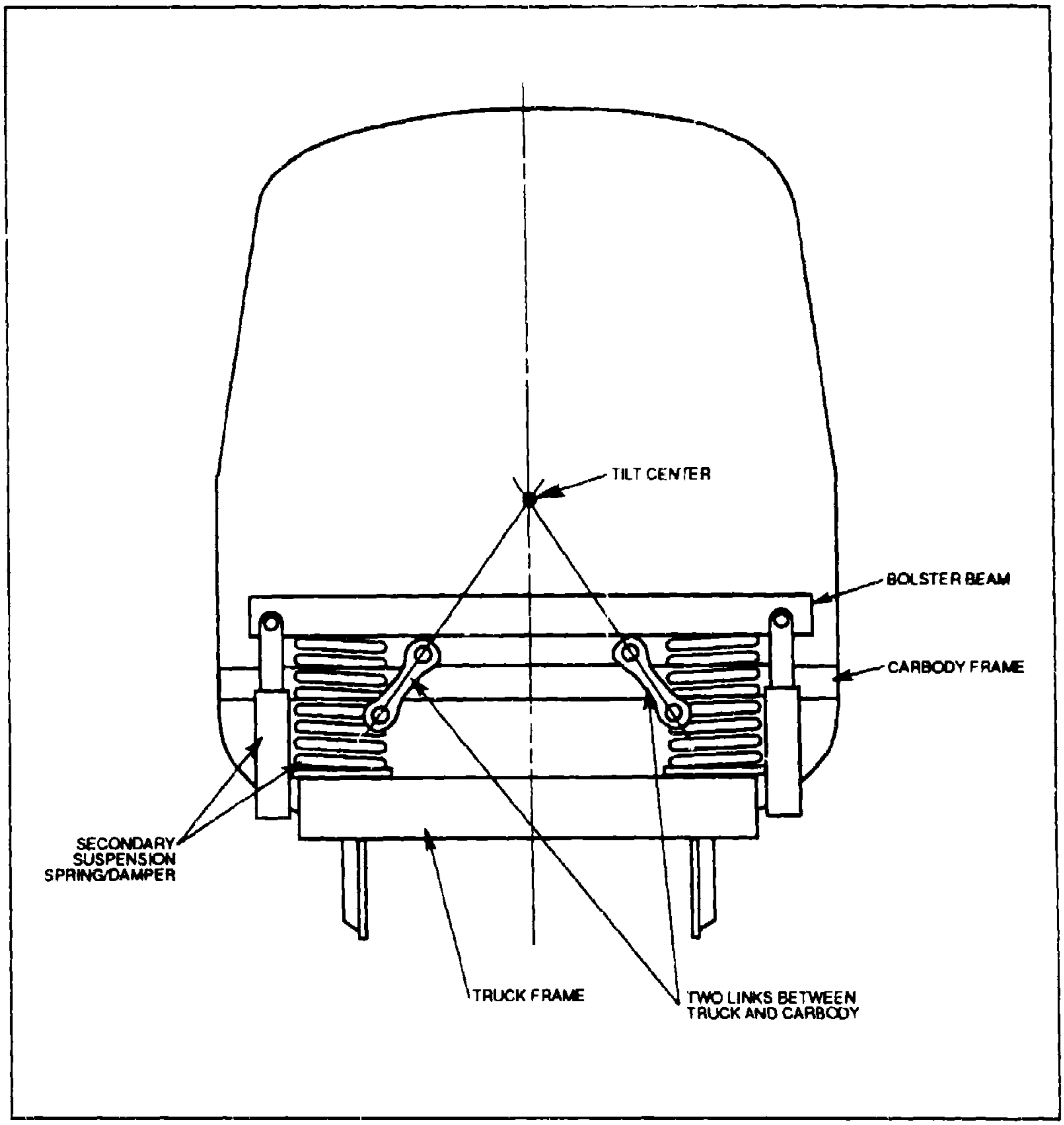

Figure B-2: Simple Two-Link Mechanism For Carbody Tilting

The mechanisms used to achieve carbody roll rotation are relatively simple, and should not pose reliability or maintainability problems beyond those associated with conventional rail passenger vehicle trucks. These conventional trucks typically incorporate linkages and/or other components which are very similar in operational loading, reliability and maintainability requirements to those elements which are used to provide for roll rotation capability in tilting trains. However, there is no question that the tilting mechanisms add complexity relative to the conventional (and

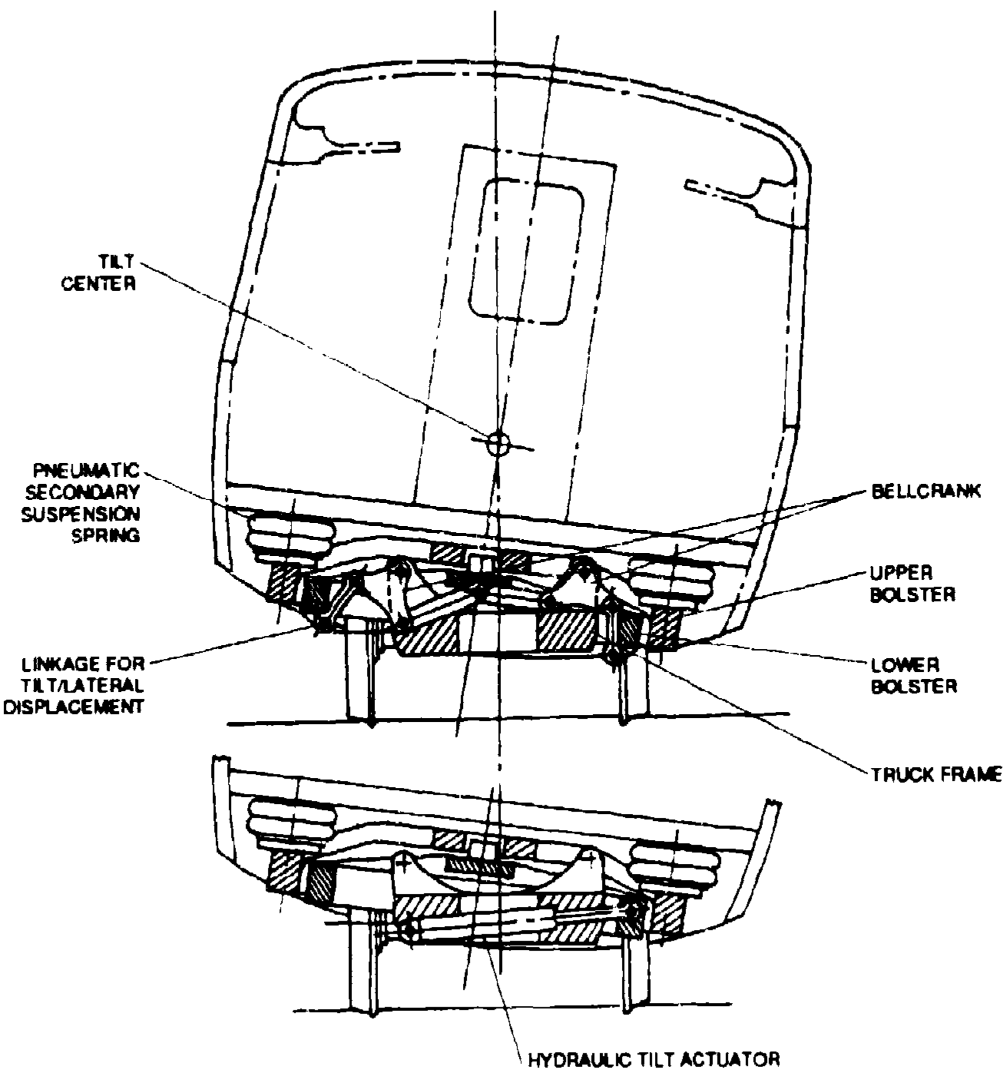

Figure B-3: The LRC Carbody Tilting Mechanism

very simple) passenger truck, which for the most part, is based on the three-piece freight truck which often forms the *de facto* datum for the North American railroad industry.

B.3 TILT ACTUATION MECHANISMS

The active tilting of a carbody about its center of roll can be achieved by means of either hydraulic or pneumatic linear piston actuators connected between the vehicle truck and the carbody so as to exert a torque about the tilting carbody roll center. With one exception, all the active-tilting trainsets which are currently in revenue service or which have been tested in a full-

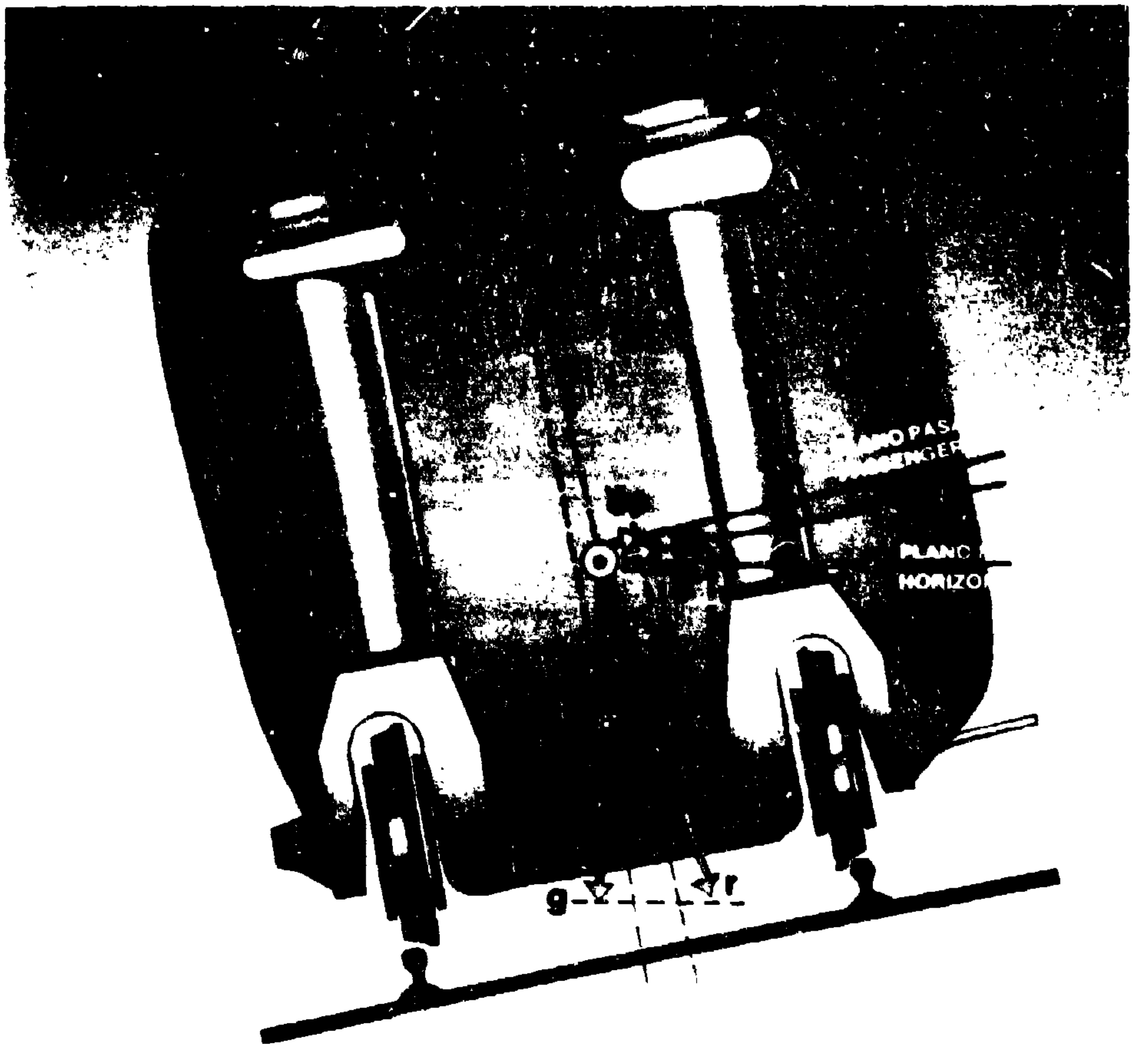

Figure B-4: Talgo Pendular Passive-Tilting Mechanism

scale prototype incorporate either one or two hydraulic actuators mounted on each truck to rotate the carbody. The sole exception to the use of hydraulic actuators is the pneumatic actuator originally developed as a possible retrofit package for the JR Series 381 EMU sets, and later incorporated in the JR Series 2000 active-tilt DMU equipment.[1] **Figure B-5** shows a typical hydraulic actuator (left) and the JR pneumatic actuator (right).

Whether hydraulic or pneumatic, the roll actuators are connected into a servo loop which permits precise control of the force applied by the actuator in response to command signals from the tilt

[1] The pneumatic actuator was originally developed in an attempt to overcome persistent problems with tilt nausea experienced by passengers on the Sr 381 trainsets ("Active-Tilting Tested as JNR Plans Narrow Gauge Speed-up," Railway Gazette International, April 1985).

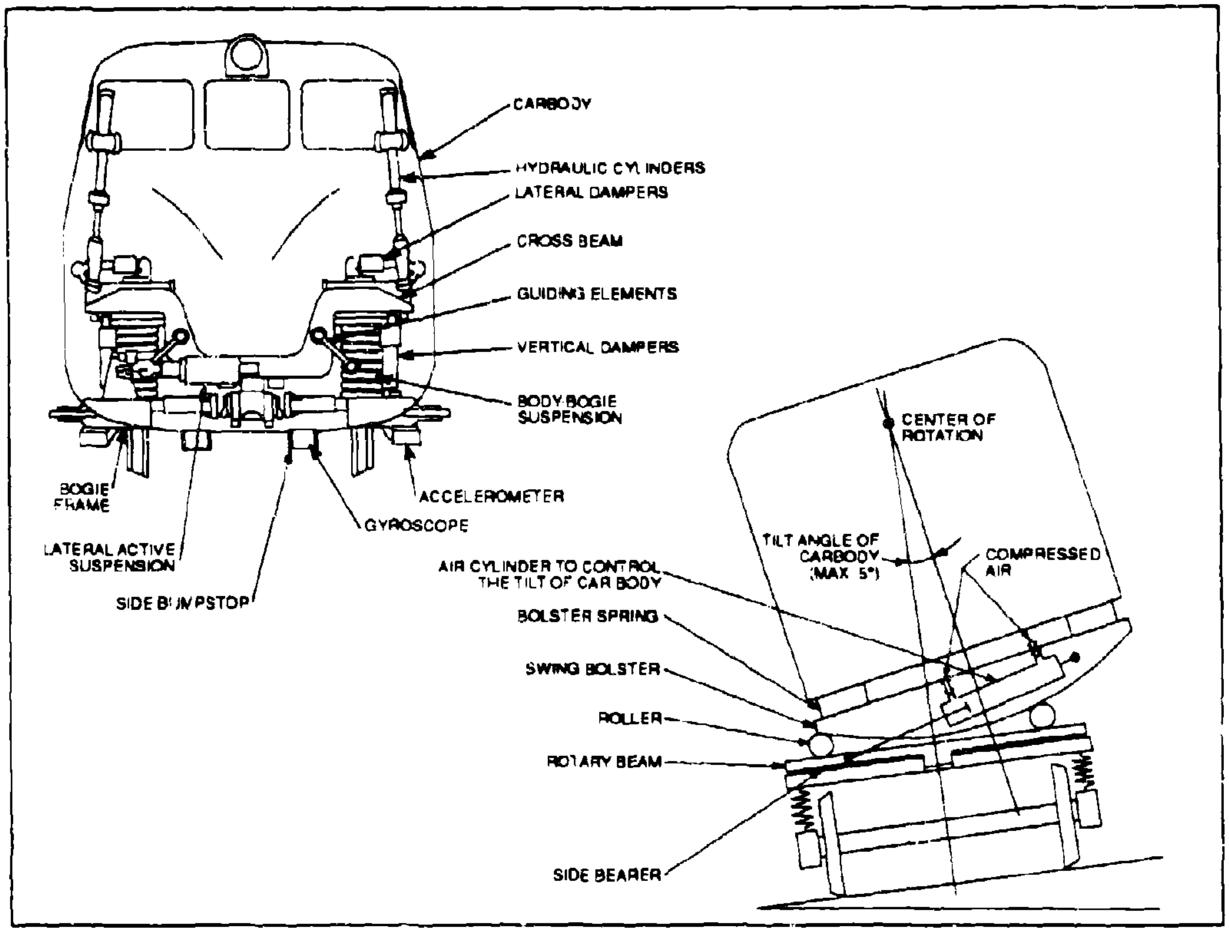

Figure B-5: Typical Tilt Actuation Mechanisms

control system. The tilt actuation servos are designed to provide fast response while retaining an adequate stability margin, using techniques which are commonly applied to servomechanisms. To enhance the vehicle ride quality and also to minimize the power consumption of the actuation mechanism, the input signal to the control mechanism is filtered to eliminate noise caused by random irregularities in local track geometry (as opposed to changes in actual alignment geometry). The schematic for a typical servo loop is shown in **Figure B-6**.

Power packs for the supply of high-pressure hydraulic oil or compressed air typically employ self-contained compressor units driven by electric motors mounted on each vehicle. These units incorporate the fluid filters, redundant pumps, and other design safety factors that are normal industry practice.

Servoed hydraulic actuators provide greater inherent resistance to unwanted carbody roll motion induced by inertial forces, and a faster command response time than do servoed pneumatic actuators. This reflects the much higher viscosity of hydraulic oil (which is effectively incompressible) relative to even high-pressure air, which remains quite compressible. However,

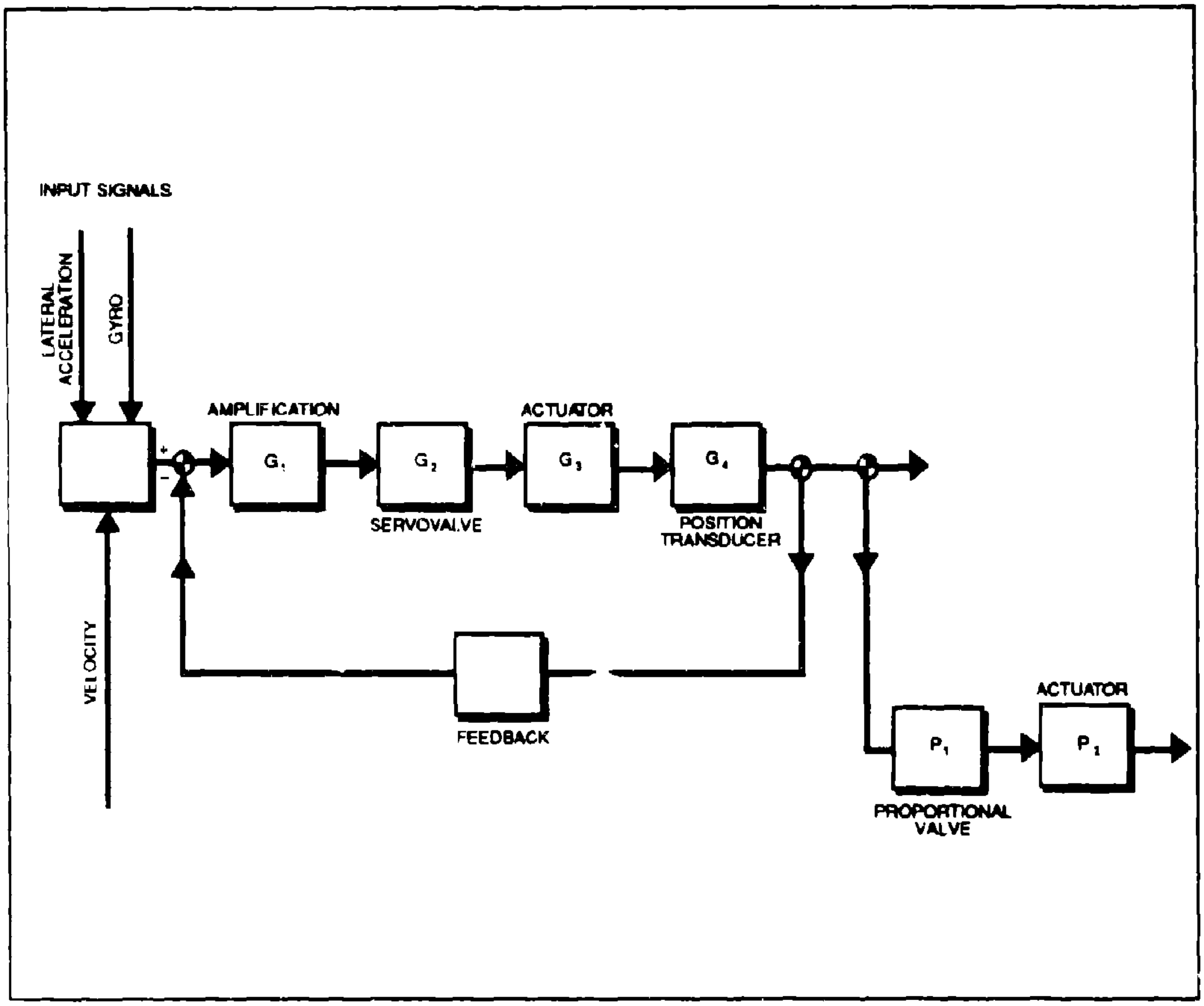

Figure B-6: Schematic Diagram for Typical Tilt Control System

compressed-air systems are generally less demanding to fabricate and maintain. Of perhaps even greater importance, compressed-air technology is "familiar ground" to every railroad and transit property in the U.S. through the ubiquitous air brake, and so would pose no technical or institutional hurdles. Hydraulic systems, on the other hand, have traditionally been regarded as maintenance "headaches" by U.S. railroads (with some justification), and could pose jurisdictional problems where shop trades are concerned.

B.4 CONTROL SYSTEMS

B.4.1 Requirements for Input Parameter Detection

The control of active-tilt carbody roll rotation typically requires continuous real-time sensing of one or more of the following parameters by means of a vehicle-mounted sensor suite:

- o Lateral acceleration of the vehicle;
- o Carbody roll angle relative to the truck;
- o Track superelevation; and/or
- o The location of the vehicle relative to defined curves.

Vehicle lateral acceleration is detected by accelerometers mounted on the carbody and/or on the truck(s) of the vehicle. Carbody-mounted accelerometers provide a direct measure of compensated lateral acceleration (and thus force) while truck-mounted accelerometers measure the uncompensated acceleration at track level while traversing curved track. Ruggedly constructed, reliable accelerometers suitable for this application are available commercially. All of the active-tilt trainsets which are described in Appendix C to this report use vehicle-mounted accelerometers to detect lateral acceleration. The acceleration data form an essential input for control of carbody tilting. **Figure B-7** shows the schematic for a typical tilt control system including the accelerometers.

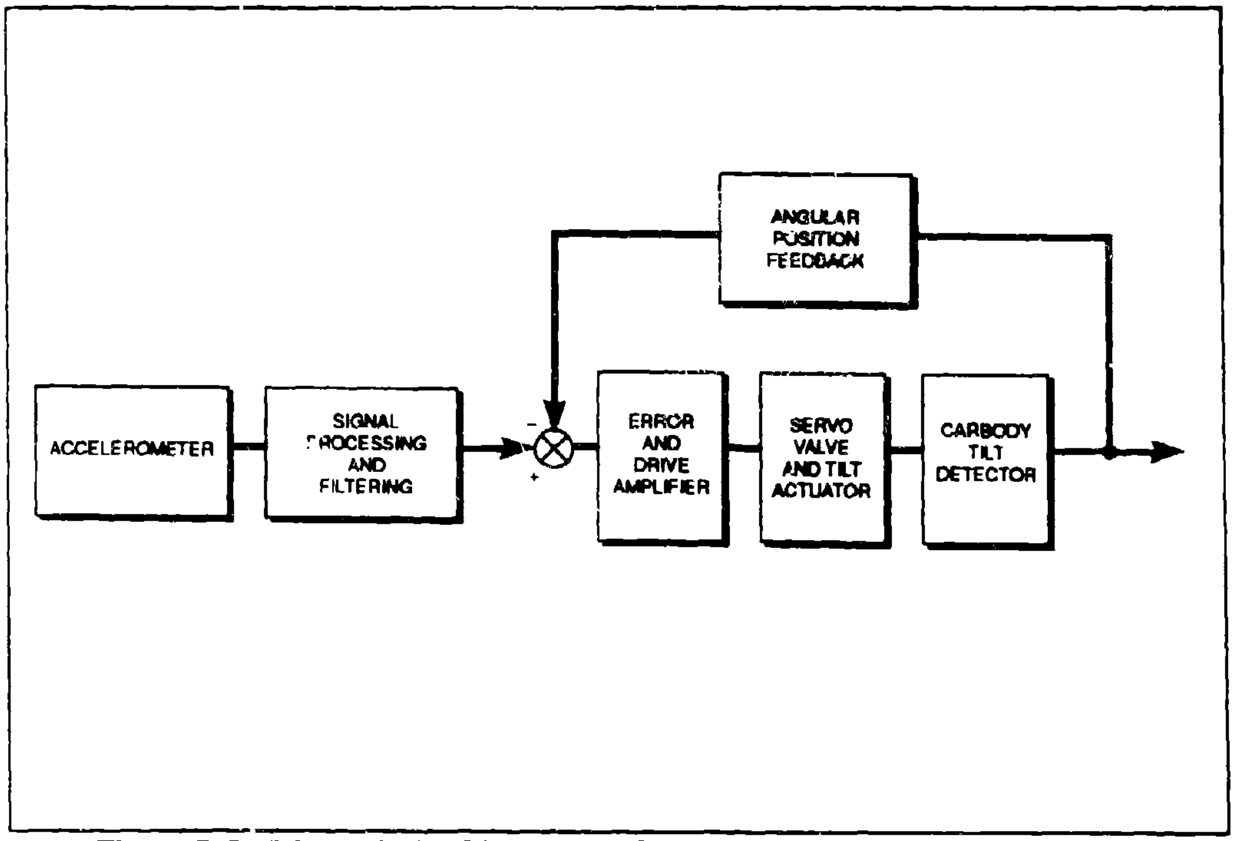

Figure B-7: Schematic for Tilt Control System Incorporating Accelerometers

The ABB X2000, the modified JR Series 381 EMU and the JR TSE-2000 DMU active-tilt equipment also incorporate sensors to measure the roll orientation of the carbody; the now defunct BR APT also carried this type of sensor. This is accomplished using displacement transducers, such as differential transformers, mounted between the vehicle carbody and the truck. As with accelerometers, suitable displacement transducers are commercially available. **Figure B-8** illustrates the schematic for a control system incorporating displacement transducers.

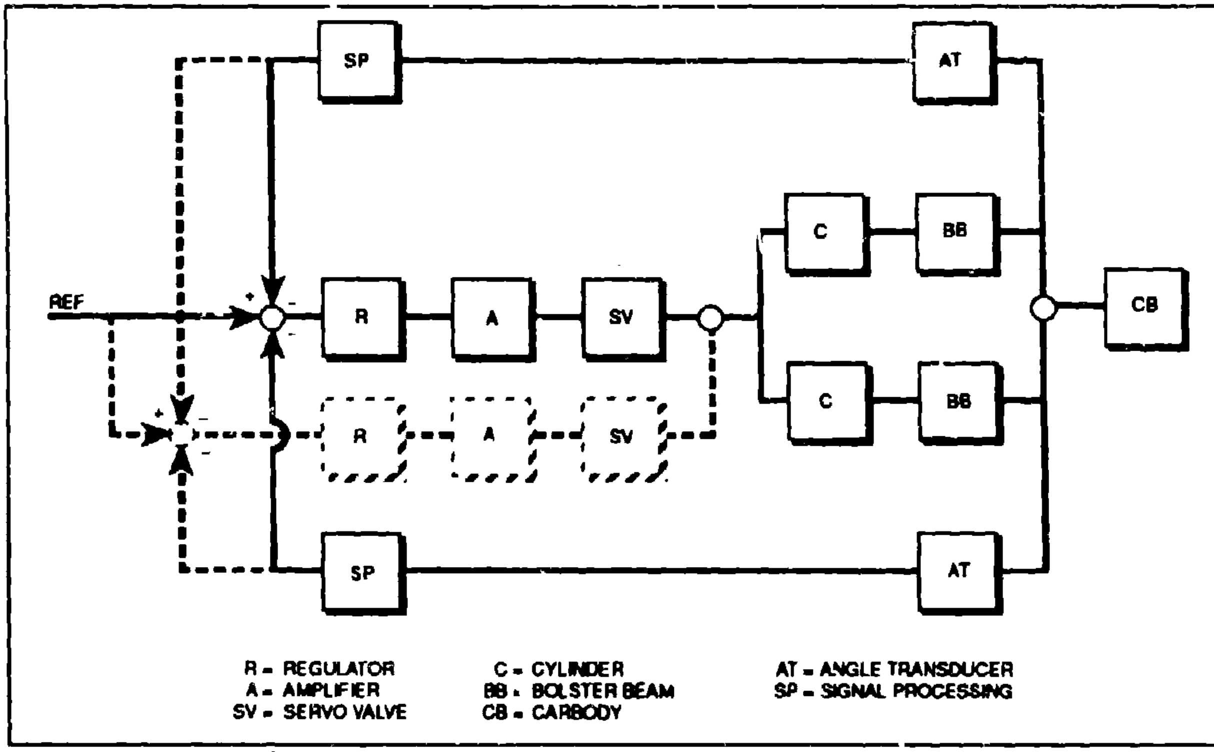

Figure B.8: Schematic for Control System Using Displacement Transducers

The data generated by these sensors serves a variety of functions, depending on the specific technology. On the JR equipment, the roll orientation (tilt angle) signal forms the feedback signal for the pneumatic servo-actuators, allowing the control system to follow the computerized vehicle roll data file for a particular route. These programmed data are correlated with the measured location of the vehicle on the track (as discussed in Appendix C) to ensure correct timing and rate of tilt onset.

In the ABB X2000, the roll orientation signal is used to monitor the tilt angle of each vehicle in the trainset relative to the tilt angle of adjacent vehicles, in order to detect and correct any inappropriate response on an individual vehicle. The control system in the APT used the roll orientation data as the feedback signal for the hydraulic servo-actuators, relative to the required roll angle derived from measured lateral acceleration.

On the Fiat ETR 450 EMU trainsets, track superelevation is sensed using gyroscopes mounted on the leading truck of each control car, as shown in **Figure B-9**. The gyroscopes provide an absolute horizontal datum against which to measure the roll orientation of the truck.

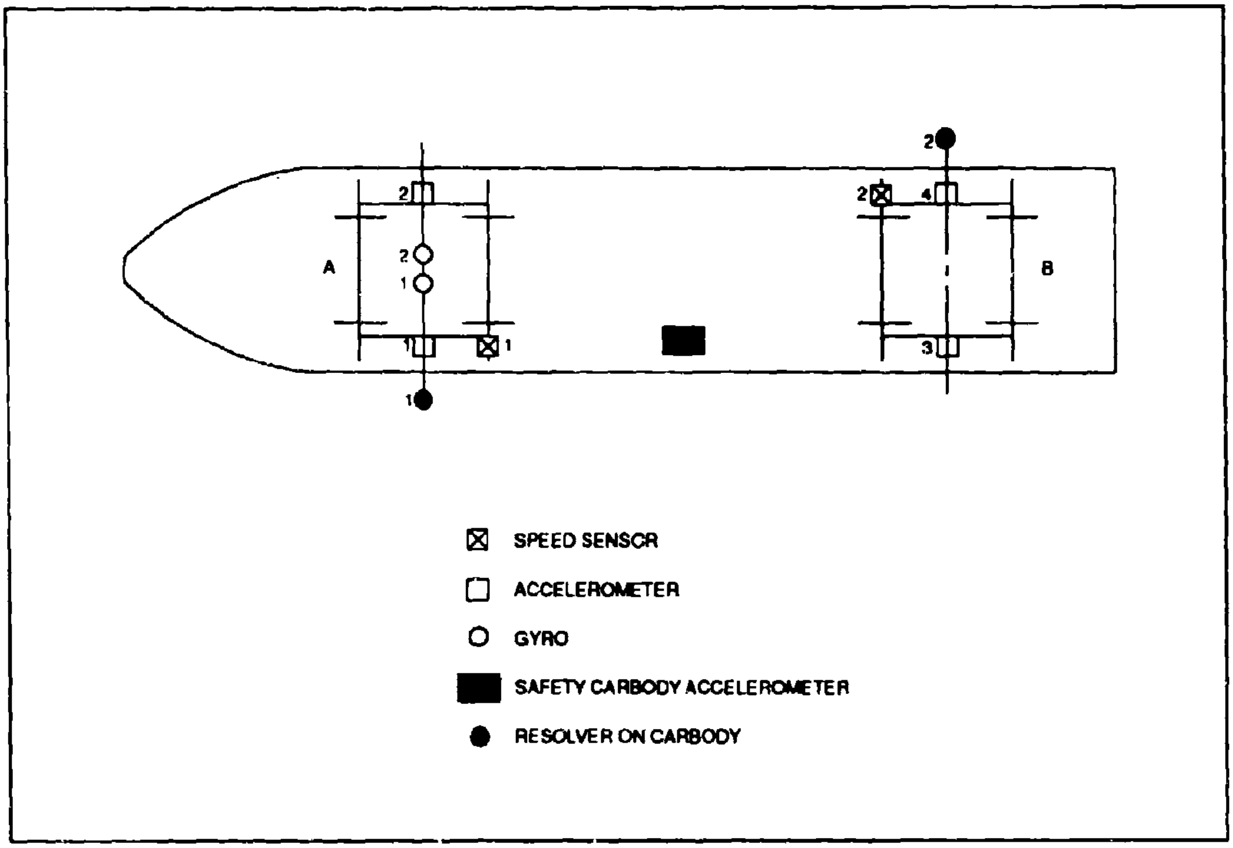

Figure B-9: Location of Gyroscopes on ETR-450 EMU

Measurement of the truck roll orientation provides a relatively close approximation to actual track superelevation, provided that the differential compression of the truck suspension springs is not large. The differential compression is relevant because the gyroscopes are mounted on the truck frame above the primary suspension; they sense the orientation of the frame rather than that of the wheelsets in contact with the track.

The control system for the ETR-450, shown schematically in **Figure B-10,** uses the truck roll orientation signal as a proxy for track superelevation to ensure that tilt initiation occurs at the correct time, slightly in advance of the actual entry of a given vehicle into a curve. This compensates for the delay caused by filtering to remove random noise caused by irregularities in local track geometry or the presence of switches and cross-overs. In contrast, the gyroscope signals are relatively noise-free, since the roll motion of the vehicle truck is largely insensitive to track irregularities.

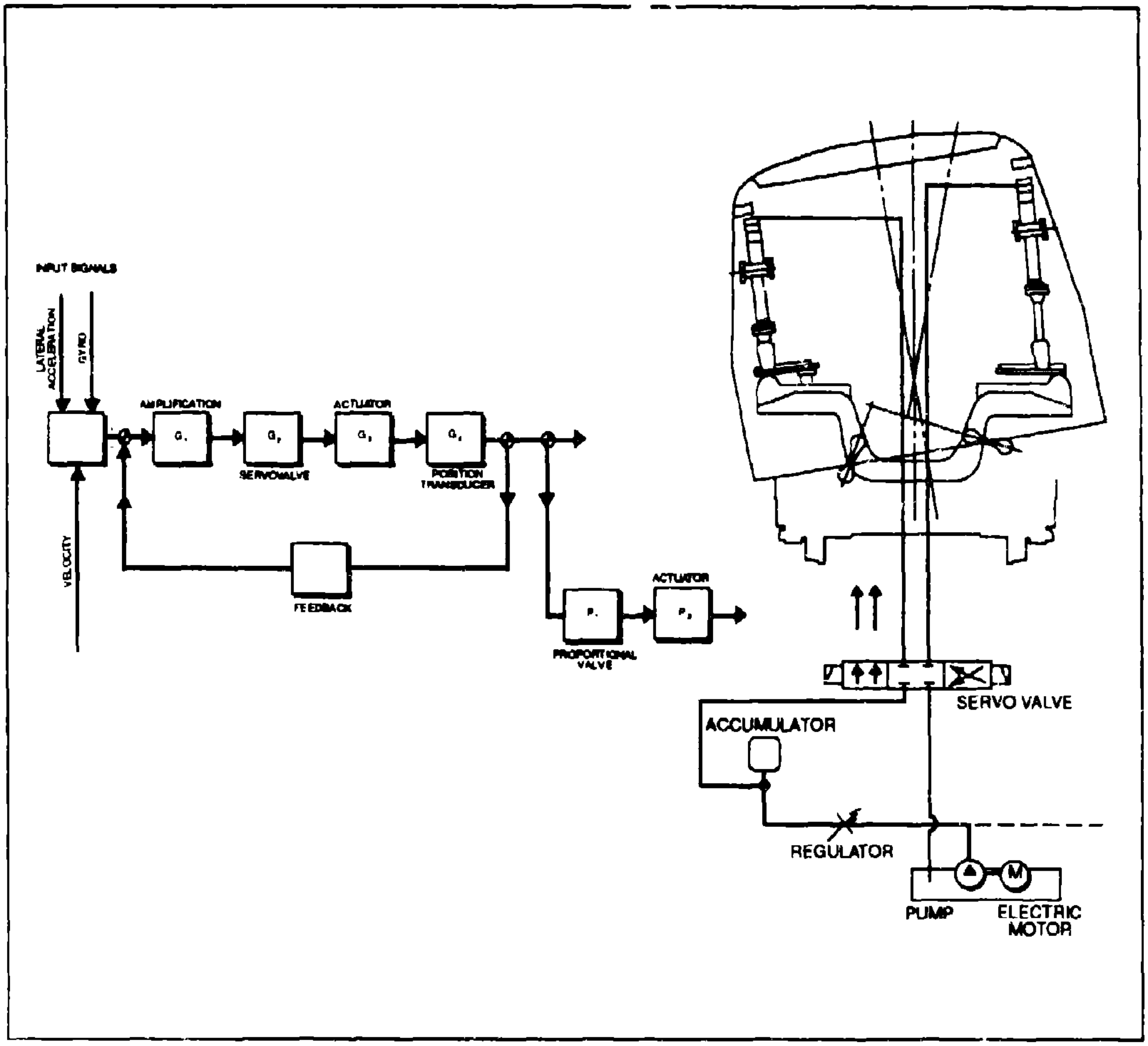

Figure B-10: Schematic Diagram of Fiat ETR 450 Tilt Control System

B.4.2 Control System Signal Processing

The control of active body tilting requires appropriate processing of the various sensor inputs to generate the correct commands to the actuation mechanism. This processing employs on-board electronics, especially microcomputers. The modification and manipulation of the input data is necessary to achieve acceptable control system response times and to ensure stable operation with an adequate safety margin. The electronic manipulation of sensor suite data includes the feedback of signals and the summing or subtracting of signals to permit comparison of signal amplitudes. Input signal modification includes amplification and filtering, as well as signal compensation to enhance the stability of feedback control loops.

The control systems for active-tilt technologies are quite complex and are not addressed in detail in this Appendix. However, there is no question that appropriate response time with acceptable stability can be achieved through application of well-established control system analysis and optimization techniques.

The key to successful control of railway carbody tilting is maintenance of acceptable ride quality through transition curves, where the parameters which affect the required magnitude of body tilt and the rate of change in tilt vary continuously. As a consequence, the overall strategy for railway vehicle carbody tilt control systems relates almost exclusively to the optimization of the vehicle ride quality through transition curves, as discussed in Section B.5 below. The requirements of tilt managment through transition curves are reflected in the control system schematics detailed in Appendix C.

B.5 TILTING TRAINSET RIDE QUALITY OPTIMIZATION

Railway vehicle passenger ride quality as it affects the curving performance of tilt-body railway vehicles falls into two distinct categories:

- o Ride quality while traversing track with a constant radius of curvature (and thus, a constant balance speed, amount of superelevation, and required carbody tilt angle); and

- o Ride quality while traversing the run-on or run-off transition curves between straight (tangent) track and constant-radius curves (where the local radius of curvature, the amount of superelevation, the balance speed and the required carbody tilt angle vary continuously, and where the rate of change of curvature and/or amount of superelevation may also vary).

These ride quality categories are discussed below.

B.5.1 Ride Quality Through Constant-Radius Curves

This ride quality category deals with passenger comfort in the context of exposure to sustained lateral acceleration while traversing a curve of constant radius. This situation will arise whenever the lateral component of gravitational force resulting from the net carbody tilt angle (superelevation plus body tilt minus differential suspension compression) does not fully compensate for the force unbalance arising from track curvature.

As noted earlier in this Appendix, a passenger comfort limit of 0.08g (approximately 2.6 ft/sec^2 or 0.8 m/sec^2) for sustained lateral acceleration is generally accepted i passenger railway practice worldwide. Accordingly, the ride quality performance of vehicle carbody tilting systems will be acceptable provided that the body roll angle is sufficient to reduce unbalanced lateral acceleration within the ± 0.08g limits while traversing constant-radius curves.

B.5.2 Ride Quality Through Transition Curves

The second vehicle ride quality category deals with the much more complex issue of maintenance of acceptable passenger comfort in the context of constantly changing track geometry parameters and consequent levels of uncompensated lateral acceleration. This environment demands transient roll motion of the carbody as the vehicle traverses run-on or run-off transition curves between straight (tangent) track and constant-radius curved track. In essence, the magnitude, rate and direction of change in roll motion of the carbody must follow changes in the magnitude and rates of change in superelevation and suspension compression. This will maintain as uniform an exposure to uncompensated lateral acceleration as is possible, both with respect to the magnitude of the acceleration and especially the rate of change of the acceleration (the so-called *jerk*).

Transient roll motion can and often does produce very adverse effects on passenger ride comfort, although passenger sensitivity to transient roll motion in combination with transient lateral acceleration is still not well defined. Much fundamental research remains to be done in this area, particularly as regards the effects of rapid reversals in the direction of acceleration (as occurs in reverse curves).

The issue of passenger ride quality on tilt-body equipment traversing transition curves is further complicated by the absence of any meaningful objective measure of ride quality. The rather meager literature is dependent on highly subjective passenger perceptions of ride comfort - in essence, little better than anecdotal information. While there is no doubt that the effects are real - JNR, the predecessor of the JR Group companies, was forced to issue anti-nausea pills to passengers and crew on its Sr. 381 passive-tilt EMUs prior to development of its pneumatic servoactuator[2] - it remains very difficult to correlate passenger perceptions with actual exposure to uncompensated acceleration and roll transients. This issue, which represents the most contentious aspect of tilting trainset technology, is considered in some detail in the following subsections.

B.5.2.1 <u>Characteristics of Passenger Ride Quality on Tilt-Body Trainsets on Transition Curves</u>

The effects on passenger ride quality induced by the tilting of the vehicle body while traversing a transition curve differ fundamentally from those induced by exposure to vibrational motion. Passenger discomfort induced by vibration relates primarily to the sympathetic vibration of the

[2] Op. Cit., <u>Railway Gazette International</u>, April 1985.

human body with emphasis on the resonant vibration of internal body organs. The ISO ride comfort specifications shown in **Figure B-11** are most restrictive for vertical acceleration in the four to eight cycles per second frequency range, which corresponds to the resonant frequencies of the internal body organs.

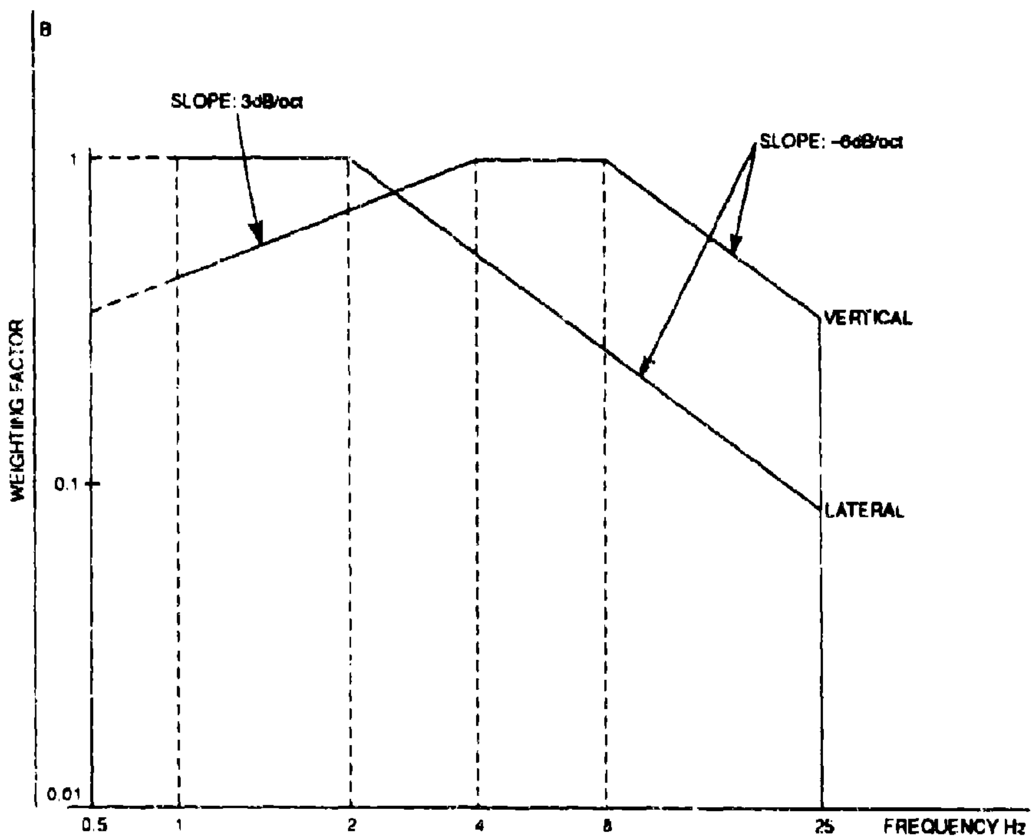

Figure B.11: ISO Ride Quality Specification

In contrast, passenger discomfort induced by transient carbody roll rotation while traversing transition curves relates primarily to variability in whole-body force levels, which disturbs the equilibrium or balance of passengers and, in extreme cases, causes motion sickness.

It is straightforward to quantify railway vehicle motion in terms of acceleration amplitude and frequency, especially for steady-state acceleration. There have been many tests of ride quality conducted over the last 40 years, and the population of passengers, observers and other test subjects is quite large. This body of data with respect to human response to vibrational motion has resulted in identification of key parameters and determination of passenger sensitivity over the ranges associated with railroad operations. These parameters are codified in the International Standards Organization (ISO) Standard 2631 for human exposure to vibratory motion, and form the basis for most ride quality standards outside North America, where the Peplar standards are commonly used.

However, the ISO and Peplar criteria do not address the relationship between passenger comfort and the transient motions that arise when rail vehicles traverse transition curves. These transient motions are of such low frequency as to fall outside the range of the existing ride comfort standards. When an attempt is made to correlate quantified measurements of the accelerations associated with transient motion with ride comfort as reported by passengers, the whole area becomes very subjective. The subjective nature of reported ride quality results in the performance of tilting railway vehicles being conditional and sometimes controversial.

Only very limited data are available in the literature with respect to railway vehicle passenger ride comfort during curving. Some limited data relating to this area has been published in connection with the design of the vehicle tilting system for the ABB X2000 tilting trainsets.[3] The most comprehensive data so far relating to passenger ride comfort during curving resulted from the ride quality testing program carried out by British Rail in 1983-84, using a number of volunteer test subjects riding both tilting and non-tilting trainsets through curves over a range of unbalanced speeds.[4] This British Rail program to investigate ride quality during curve negotiation was conducted primarily to assess the effectiveness of the active-tilt APT relative to non-tilting trainsets. A limited program of ride comfort testing during curving was carried out by JNR at about the same time, in connection with retrofitting of the Series 381 passive-tilt EMU fleet with a form of active-tilt control.[5]

The results of these test programs can be summarized as follows:

- o Passenger perception of ride comfort with respect to the transient accelerations and roll motions experienced when traversing a curve is very different than the perception of equivalent levels of conventional vibrational motion,

- o A run-on transition curve is generally perceived by passengers to result in a significantly less acceptable level of ride comfort than a run-off transition curve having exactly the same geometry,

- o Passengers are particularly sensitive to the rate of change of lateral acceleration (the lateral *jerk*) while traversing a curve,

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[3] "Swedish Body-Tilting Electric Set for Very High-Speed on Severely-Curved Main-Lines," N. Nilstam, Rail Engineering International, May-Sept. 1982.

[4] "Passenger Tolerance of High-Speed Curving," M.G. Pollard, Railway Gazette International, Nov. 1984.

[5] "Ride Quality Evaluation of a Pendulum Car," S. Koyanagi, Quarterly Reports of the Railway Technical Research Institute, Japanese National Railways, Vol. 26, No. 3, 1985.

- o Standing passengers are much more sensitive to variations in acceleration and jerk than are seated passengers,

- o Variations in lateral acceleration and jerk totally dominate variations in both the vertical and longitudinal directions as a determinant of perceived ride quality, and

- o The motion parameters which determine perceived ride quality when traversing a curve with an unbalanced lateral acceleration of less then the 0.08g ride comfort limit appear to be:

  - the rate of change of lateral acceleration, or *jerk*;
  - the tilt roll velocity; and
  - the tilt roll acceleration.

In relation to the apparent asymmetry of sensitivity to transition curve geometry, humans apparently can compensate for variations in applied lateral force and/or roll rate as induced by vehicle motion, provided that sufficient time is available for the equilibrium control system of the body (the inner ear) to respond and prevent the person from being thrown off balance. The ability of a passenger to anticipate variations in lateral acceleration or roll rate both with respect to time of onset and magnitude of variability appears, from the very limited information in the literature and from intuition, to produce a very significant reduction in the degree of ride discomfort perceived by a passenger for any given variation in lateral acceleration or roll rate.

The ability of a passenger to anticipate the onset of variation in lateral acceleration and/or roll rate appears to result from the (typically) gradual onset of parameter variability. In other words, once steady-state conditions cease, a rider is not surprised if the transient effects get worse.

The much greater degree of perceived ride discomfort associated with a vehicle traversing an unanticipated run-on transition curve, as compared with a run-off transition curve having the same geometry and at the same speed, has been attributed to passenger anticipation of the run-off curve while the vehicle is still traversing the constant-radius segment of the curve.[6]

In essence, passengers tend to be caught more "off balance" by entry onto the run-on transition curve than for the run-off transition curve where some degree of anticipation exists. The degree of gradualness (the *smoothness*) of lateral acceleration and roll rate variations tends to be the measure of the "anticipation" factor in ride quality as perceived by vehicle passengers. The "smoothness" of lateral acceleration and roll rate variations is indicated by the absence of inflections or sudden changes in the slope of a plot of the parameter variations as a funtion of the distance travelled along the curve. Such inflections may be more easily identified and

[6] Op.Cit., Pollard.

quantified by determining the variation of the slope of the lateral acceleration and roll rate variations, as the derivative of the lateral acceleration and the roll rate to give the lateral rate of change of acceleration or *jerk* and the roll acceleration, respectively. The more uniform the variation in these jerk and roll accelerations, the greater the "anticipation" factor in ride quality as perceived by vehicle passengers. This supports the dependency of perceived ride comfort on lateral jerk and roll acceleration as indicated above.

In relation to the greater sensitivity of standing passengers, one would intuitively expect a standing person to be much more sensitive to the disruption of equilibrium or balance caused by whole-body force variation. This expectation is consistent with the perceived differential in level of discomfort reported by Pollard.[7] Finally, investigators have consistently found that lateral force variations dominate passenger perceptions of ride comfort. Variations in vertical or longitudinal acceleration apparently do not trigger any significant disturbance in passenger equilibrium or balance such as could lead to motion sickness.

B.5.2.2 Tilt-Body Performance Requirements for Enhanced Passenger Ride Quality Through Transition Curves

Given that the sudden disturbance of lateral balance or equilibrium represents the primary source of passenger discomfort while traversing a transition curve, and that the ability of a passenger to anticipate the onset of variations in lateral acceleration or roll motion significantly reduces the perceived level of discomfort, one can define several performance criteria for tilt-body vehicles that will improve the ride quality as perceived by passengers:

- The rate of change of the carbody roll rate, and the rate of change in the unbalanced lateral acceleration must be jointly optimized so as to reduce the perceived disturbance from variations in both roll motion and lateral force to a minimum; and

- Variation of lateral acceleration and roll motion through the transition curve should be made as gradual and consistent as possible, so as to maximize the ability of the passenger to anticipate the onset of changes.

These criteria for enhanced passenger ride quality are consistent with the geometric design standards for transition curves for dedicated high-speed alignments, as described in Appendix A. In particular, **Figures A.6** and **A.7** show transition curve characteristics which, if emulated by a trainset tilt control system, would satisfy the performance criteria for enhanced ride comfort defined above. The tilting system would compensate for deficiencies in transition curve alignment geometry so as to produce ride quality approaching that achievable on dedicated high-

[7] Op. Cit., Pollard. See especially Figures 4 and 6.

speed track constructed with optimized transition curve geometry. The criteria are also consistent with the preference for smoothed roll rate variation advocated by the Japanese Railways for their modified Series 381 tilting trainsets.[8]

Evidence from full-scale tests of ride comfort in curves clearly establishes the strong influence of anticipation of the onset of lateral acceleration and/or roll motion on the perceived level of discomfort reported by the test subject. This evidence supports application of a tilting strategy that is capable of delivering smooth and continuous variation in lateral acceleration and roll

motion, rather than one with perceptible discontinuities in the rate of change of roll rate and unbalanced lateral acceleration.

B.5.2.3 Ride Quality and The Effects of Track Geometry

The potential advantage of tilt-body vehicle systems depends on achieving an acceptable level of ride quality *as perceived by the passenger* while traversing curves in existing track at speeds significantly above the limits set by ride comfort requirements for non-tilting vehicles. Basically, the thrust of tilt-body technologies is to raise the average speed over a given track segment without necessarily altering the allowable maximum speed on the segment.[9] Experience with tilting trainsets, both in test and in revenue operation in a number of countries, has shown that passenger ride comfort through *constant-radius curves* can be maintained or even improved if the curving speed is increased well above the balance speed for the curve geometry, up to the tilting limit of a given technology. However, the ride quality through transition curves tends to be very dependent upon the transition curve geometry.

Short transition curve lengths, coupled with sudden variations in the track curvature and superelevation rates of change at the start and end of the transition curve, as is usually the case for North American railway track, can and does degrade the ride quality associated with tilting trainsets to the extent that the market advantages of improved trip time through an increase in average speed are overwhelmed.

This means that achievement of the potential benefits from carbody tilting is affected not only by the quality of the tilting system used on a given vehicle but also, and to an even greater degree, upon the the geometry of the track alignment, and in particular, on the geometry of the transition curves through which the equipment must operate. Good ride quality while operating on track built to a given set of geometric standards in no way guarantees acceptable performance

[8] See Figure 9 of "Ride Quality Evaluation of a Pendulum Car," S. Koyanagi, Quarterly Reports of the Railway Technical Research Institute, Japanese National Railways, Vol. 26, No. 3, 1985.

[9] Insofar as the maximum speed will reflect safety criteria related to L/V force ratio, truck stability, and signalling and train control, none of which are enhanced by body tilting.

when running on track built to different geometric standards. Since all tilt technologies except the LRC have been designed, tested, and operated outside North America, assertions regarding the transferability of ride quality and technology performance should be treated with caution. North American track reflects local alignment and construction practices intended to address quite different geotechnical, operational and commercial considerations, as well as different utilization and maintenance histories and maintenance practices.

B.5.2.4 Tilt Control Strategies to Enhance Ride Quality Through Transition Curves

Transition curves often include one or more geometric deficiencies that make it difficult or impossible for an active-tilt control system to track changes in curve geometry so as to compensate for variation in lateral acceleration, without subjecting passengers to unacceptable jerk and roll rates. These deficiencies include:

- o Short transition curve lengths relative to the overall change in curvature and superelevation, so that the rate of change of these parameters is high,

(Short transition curves force the tilt control system to apply a high roll rate through the transition curve. High roll rates are known to result in high levels of passenger discomfort, especially in combination with high roll acceleration rates.[10])

- o Discontinuities in the rate-of-change of curvature and superelevation, particularly at the entrance to and exit from transition curves,

These discontinuities are inherent in clothoid transition curves, which are ubiquitous in freight and mixed-use track, particularly in North America. The notable exceptions to the foregoing statement are the dedicated high-speed passenger tracks in Japan and France, which incorporate more sophisticated transition curve geometries. These discontinuities give rise to sudden variations in the rate of change of lateral acceleration (and thus force) acting on passengers, and of the roll rate, at the entrance to and exit from clothoid transition curves (see Appendix A for discussion of geometric effects). These sudden variations result in a high level of passenger discomfort, stemming from the need to either rapidly compensate for the unexpected force and motion, or else lose one's balance.

- o Differences in the variability of curvature and superelevation along the length of transition curves

Variability in curvature and superelevation arises from poor track construction and/or

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[10] See Figure 8 of "Ride Quality Evaluation of a Pendulum Car," S. Koyanagi, Quarterly Reports of the Railway Technical Research Institute, Japanese National Railways, Vol. 26, No. 3, 1985.

maintenance, resulting in differential settlement of the subgrade or track structure which creates a variety of localized track defects, especially warp defects. These defects can give rise to some additional variability in roll motion perceived by passengers if the tilting system is designed to respond to vehicle lateral acceleration, as is typically the case. However, this type of variation tends to have a relatively gradual and smooth onset and taper, and also typically is of modest amplitude relative to the intentional changes in track superelevation, so that the effect on perceived ride quality is normally quite limited. Severe problems with local track geometry can, of course, exacerbate ride quality degradation on tilting or conventional equipment.

B.5.2.4.1 Passive Tilting System Design Strategies to Enhance Ride Quality Through Transition Curves

There is very limited potential for modification of the design strategy for passive tilting systems to compensate for deficiencies in transition curve geometry. The response time of passive-tilting systems is dictated primarily by the location of the roll center about which the vehicle carbody swings and secondarily by the rotational inertia of the carbody about this roll center. Both attributes are essentially fixed for a given vehicle design.

The only technique compatible with passive-tilting that can affect system performance entails the addition of hydraulic dampers to reduce the rate of roll generated by the pendular motion of the carbody swinging about the roll center. Roll motion damping could increase the extent to which passengers are able to anticipate the onset of motion, and so enhance the perceived ride quality. However, the use of dampers to control roll motion increases the delay in the reponse of the carbody to changes in lateral acceleration, and so could in fact increase the level of unbalanced lateral acceleration (force) which would act on a passenger through a given transition curve.

A carefully optimized design for roll damping could result in an overall improvement in perceived ride quality but the potential for meaningful changes would be very limited, and any assessment would have to be undertaken on a technology- and route-specific basis to yield valid results. In practice, passive-tilting systems have not incorporated roll motion dampers. This implies that the technology developers and operators have found that the rotational inertia of the tilting carbody provides for sufficient (or possibly excessive) damping. With passive body tilting, about the only solution to unacceptable ride quality through transition curves on a particular route is to reduce the speed(s) at which the curves are traversed.

Exactly this sort of situation arose with respect to passenger ride comfort on the then-JNR Series 381 passive-tilting narrow-gauge EMUs. These pendular trainsets were introduced into fleet service in 1973 in an attempt to increase maximum speed (to 130 km/h from 120 km/h) and especially to raise allowable speed in curves by 20 to 25 km/h (i.e., to 85 km/h from 65 km/h, for a 300m radius curve) on the Nagoya-Nagano line. The revenue service experience of JNR, and later its successor company JR-Shikoku, with this equipment on severely curving narrow gauge track through mountainous terrain is very informative with respect to ride comfort considerations in general and to tilt motion sickness in particular.

In any event, neither objective was realized. Maximum speed could not be increased, and persistent and widespread tilt motion sickness led JNR first to issue motion-sickness medication to all passengers, and to then restrict curving speeds to the levels used for conventional equipment. Ultimately, JNR was forced to develop a retrofit package incorporating a pneumatic servomechanism to provide a form of active-tilt control.[11]

The passive-tilting trainset operating experience of the JNR and its successors is quite extensive; only that of RENFE in Spain is similar. In 1984, there were 277 passive-tilting cars in the Series 381 EMU fleet. This equipment is still in revenue service, but has been modified with the addition of the active-tilt pneumatic controller. In addition, there are 38 active-tilt diesel multiple unit (DMU) trainsets which began operation on the Takamatsu to Matsuyama line on Shikoku Island in 1990. The DMUs use essentially the same active-tilting system as the upgraded Series 381 trainsets.[12]

The only other revenue operation of passive-tilting equipment has been by the Spanish National Railways (RENFE) using Talgo *Pendular* coaches hauled by diesel or electric locomotives. The Talgo *Pendular* design is detailed in Appendix C. Although based on design principles enunciated in the 1940's, the equipment now in fleet service was delivered starting in 1980 with 16 pairs of trainsets in service on domestic routes in Spain by 1987. An additional 73 Talgo *Pendular* cars were delivered to Spanish Railways in 1989 and a further 200 cars are currently entering service as they are delivered. As well as domestic services within Spain, Talgo tilting trainsets are also used to operate run-though services to destinations in France and Switzerland.

As noted above, there are well-documented reports of poor ride quality and motion sickness on the unmodified Sr. 381 passive-tilt trainsets. The necessity of issuing motion sickness pills to passengers to control tilt nausea on severely curving track[13] is mentioned repeatedly. Apparently, there were many complaints regarding the discomfort felt on these pendulously-tilting trainsets, by both passengers and crew, particularly when standing while entering and exiting curves.[14]

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[11] Op. Cit., Railway Gazette International, April 1985. See also "Speedup on JNR 1067mm Gauge Lines," Y. Yukawa, Japanese Railway Engineering, Vol. 24, No. 2, 1984.

[12] "Development of an Active Tilt-body Diesel MU," K. Matsuda, JR-Shikoku, Railway Technology International, 1991.

[13] "Active Tilting Tested as JNR Plans Narrow Gauge Speed-Up," Railway Gazette International, April 1985.

[14] "Ride Quality Evaluation of a Pendulum Car," S. Koyanagi, Quarterly Reports, Railway Technical Research Institute, Vol. 26, No. 3, 1985; and "Technical Point for Speeding-Up on Narrow Gauge Lines," A. Maruoka, Japanese Railway Engineering, No. 112, Dec. 1989.

These ride discomfort and motion sickness problems led to the investigation and subsequent retrofitting of active-tilting systems to the Series 381 trainsets starting in the mid-1980's. The retrofitted active-tilting control system limits the carbody roll rate to about five degrees per second and the roll acceleration to about 15 degrees/sec^2 to achieve acceptable ride comfort. These roll motion limits reflect the apparent dependency of "tilt nausea" on the carbody roll motion in combination with reversals of roll direction as the vehicle enters and exits transition curves.

The issue of ride discomfort and motion sickness induced by vehicle roll motion, although not easily definable due to the very subjective perception of discomfort and to the well-established wide variation in motion sickness sensitivity among any given passenger population, is considered to be of primary importance in determining the practicality of tilting technologies. This issue is judged to represent the main deterrent to the more widespread application of tilting trainsets for operation on existing track.

The apparent absence of any reported ride discomfort problems with the Talgo *Pendular* passive-tilt trainsets operating by RENFE may be due to the smaller designed maximum tilt angle (3.5 degrees as compared to five degrees for the Series 381 trainsets). This difference would permit a reduced vehicle roll rate through transition curves with appropriate roll motion damping by the tilting carbody rotational inertia. However, the apparent lack of ride discomfort problems could also reflect the less demanding transition curves on the broad-gauge Spanish lines and the standard-gauge lines in France and Switzerland on which the Talgo trainsets operate, relative to the severely curved narrow gauge lines through the mountains of central Honshu.

B.5.2.4.2 Active-Tilt Control Strategies to Enhance Ride Quality Through Transition Curves

The potential for improving active-tilt control systems to better compensate for deficiencies in transition curve geometry and so enhance perceived ride comfort is very significant. Input data to permit control of carbody tilt motion can be readily gathered and processed on-board the vehicle including.

- o Lateral acceleration of the carbody and/or of the truck using accelerometers;
- o The tilt angle between the carbody and the truck, using displacement transducers; and
- o The track superelevation using truck-mounted gyroscopes.

Signals from these sensors are used, often in combination, as the input to the tilt control system for almost all of the active-tilting trainsets developed to date (See Appendix C). The notable exceptions are the active-tilt version of the JR Series 381 EMU, and the JR Series 2000 active-tilt DMU equipment, both of which use essentially the same tilt control system design.

The JR tilt control system is unique in that its on-board computer generates the command signal for the tilt servoactuator from a data file containing the predetermined track alignment and geometry information for the particular route being served on a given run. The current trainset location on the route is determined from trackside transponders (which form part of the "automatic train stop" traffic control system used in Japan) in combination with trainset wheel tachometers which permit calculation of the distance between transponders. The stored track information includes curve locations relative to the trackside transponders, the degree of curvature and superelevation variation through the curves. This tilt control system allows accurate warning of the approach of a curved section of track and the corresponding activation of the tilting system in anticipation of the onset of the transition curve. This system also allows for a high degree of flexibility with respect to compensation for deficiencies in transition curve geometry, as discussed above.

Since the only justification for the added complexity and cost of body-tilting capability stems from maintenance of acceptable ride quality through track curves at speeds in excess of the designed balance speed, it is imperative that tilt control strategies be able to maintain ride quality even when faced with significant deficiencies in transition curve geometry.

There are a number of approaches with the potential to deliver this capability. Three of the most promising techniques, which could be applied individually or in combination, are discussed below:

1) Extend the duration of transition tilting:

This technique essentially involves increasing the length of track (and thus, the period of time) over which the carbody is rotated from its normal position to its equilibrium roll angle for traversing a constant-radius curve, so that the "tilting track" length is greater than the length of the actual transition curve built into the track. This is illustrated in **Figures B.12** and **B.13**.

By increasing the duration of tilt onset, this allows a reduction in the tilting roll rate, which in turn enhances the transition phase ride comfort. **Figures B.12** and **B.13** clearly show that the penalty for extension of the duration of transition tilting is the emergence of variable lateral acceleration (and thus force) during this phase.However, the advantage of being able to anticipate the start of a transition curve, so as to initiate tilting prior to curve entry, is evident from the lateral acceleration curves. The maximum lateral acceleration, during the transition phase, can be reduced by as much as a factor of two, all other factors being equal.

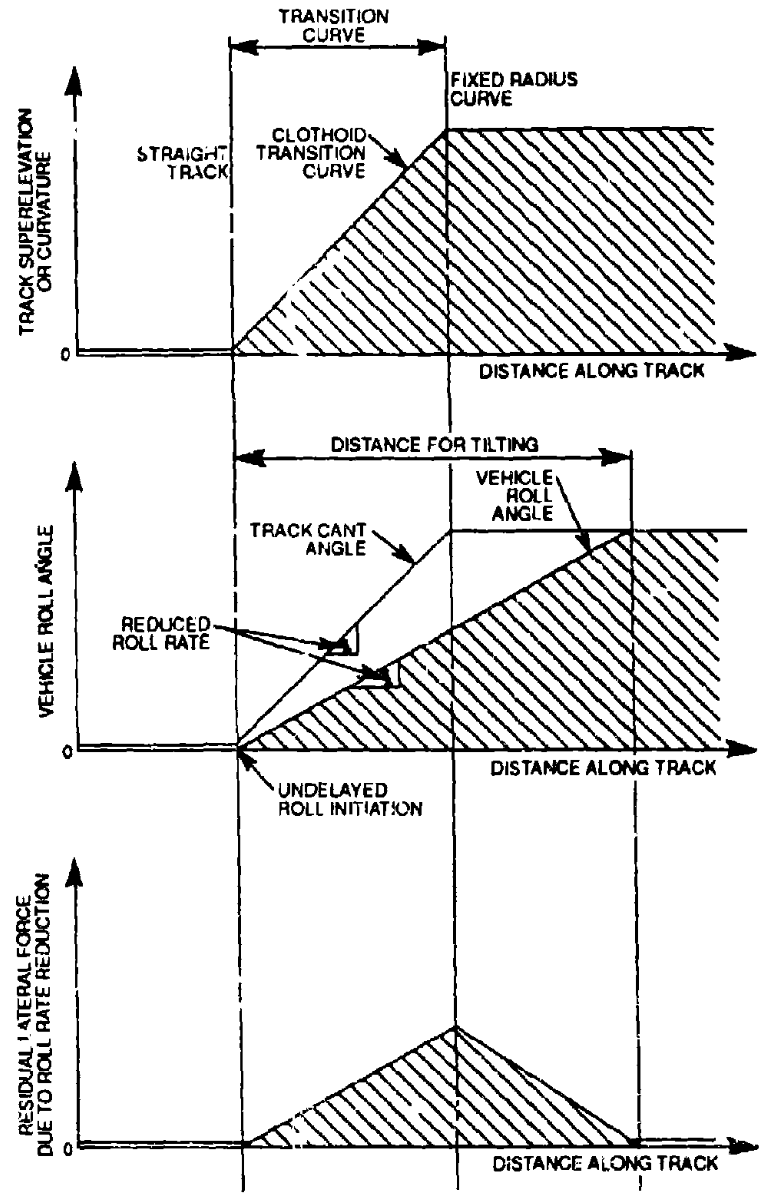

Figure B.12: Effect of Reduction in Roll Rate with No Change in Timing of Tilt Initiation

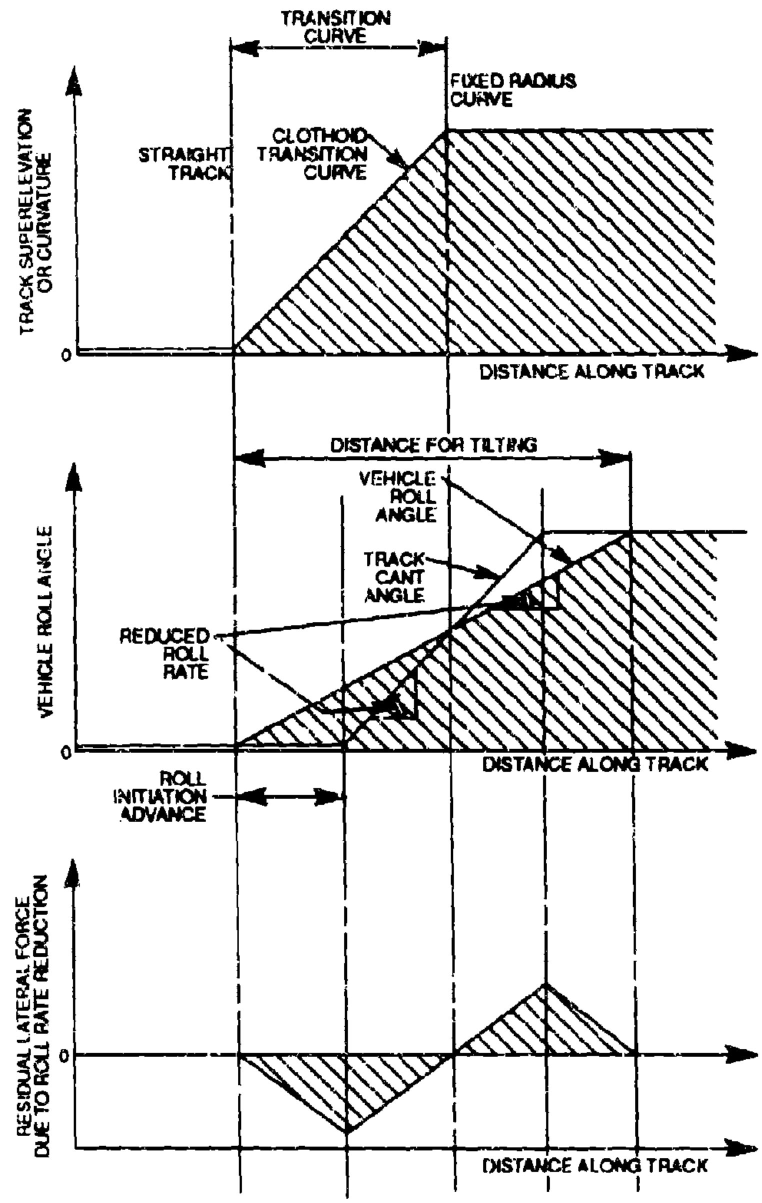

Figure B.13: Effect of Reduced Roll Rate With Tilt Initiation in Advance of Curve Entry

Conversely, if there is a delay in tilt initiation due to lags in the response of the tilt controller and/or the tilt actuator, the level of variable lateral acceleration will be even higher than is the case for tilt onset coincident with curve entry, as shown in **Figure B.14**.

2) **Reduce the rate of change of tilt roll rate during transition entry and exit:**

This technique requires a more gradual variation of the tilt roll rate at the beginning and the end of the tilt transition phase, relative to the rate which would correspond to exact lateral force compensation for the geometry of a given transition curve, as illustrated in **Figure B.15**.

Reduction in the rate of change of the tilt roll rate at the onset and termination of transition curving allows passengers some degree of anticipatory reaction to tilt initiation and termination, which should lead to an associated reduction in the perceived level of ride discomfort. However, easing the rate of change during curve entry and exit results in the emergence of variable lateral acceleration (and force) during entry and exit, as can be seen in **Figure B.15**.

3) **Limit the degree of body tilt to that required to reduce uncompensated lateral acceleration to the maximum acceptable level rather than to the minimum achievable level:**

This technique essentially trades off the degree to which body tilt compensates for lateral acceleration on the constant-radius portion of a curve against control of transient acceleration and roll rate effects during the transition portions of a curve. Instead of setting the tilt limit at the maximum compatible with safe operation, the tilt limit is set to reduce uncompensated lateral acceleration during transition of the constant-radius segment of the curve to the 0.08g limit, as shown in **Figure B.16**.

This strategy allows reduction in the tilting roll rate (since the maximum tilt angle that must be achieved in a given time period is reduced) with the associated enhancement of transition phase ride comfort. A maximum non-varying lateral acceleration of 0.08g is generally accepted by the railway community as a realistic ride comfort limit, with 0.04g acceleration regarded as undetectable by the majority of passengers. The obvious penalty associated with this strategy is the potential reduction in perceived ride comfort during traversal of the constant-radius portion of the curve.

It is quite feasible to combine aspects of all three strategies, as shown in **Figure B.17**. Clearly, there will be trade-offs between the rate and magnitude of vehicle tilt and variations in uncompensated lateral acceleration through the transition phases of curving at any given speed in excess of the balance speed; however, there will be some combination of these strategies which will be optimal for passenger ride comfort. Identification of this optimized strategy represents ***the*** design challenge facing developers and operators of tilt-body rail vehicles.

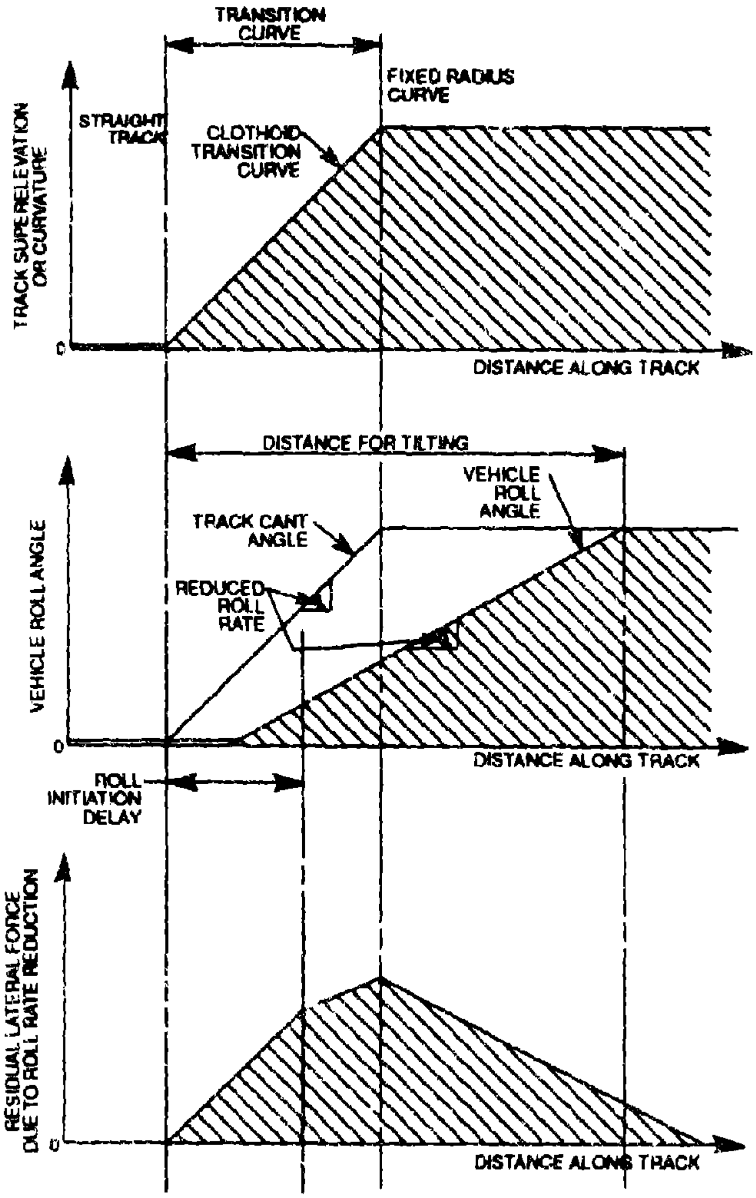

Figure B.14: Effect of Reduced Roll Rate With Tilt Initiation After Curve Entry

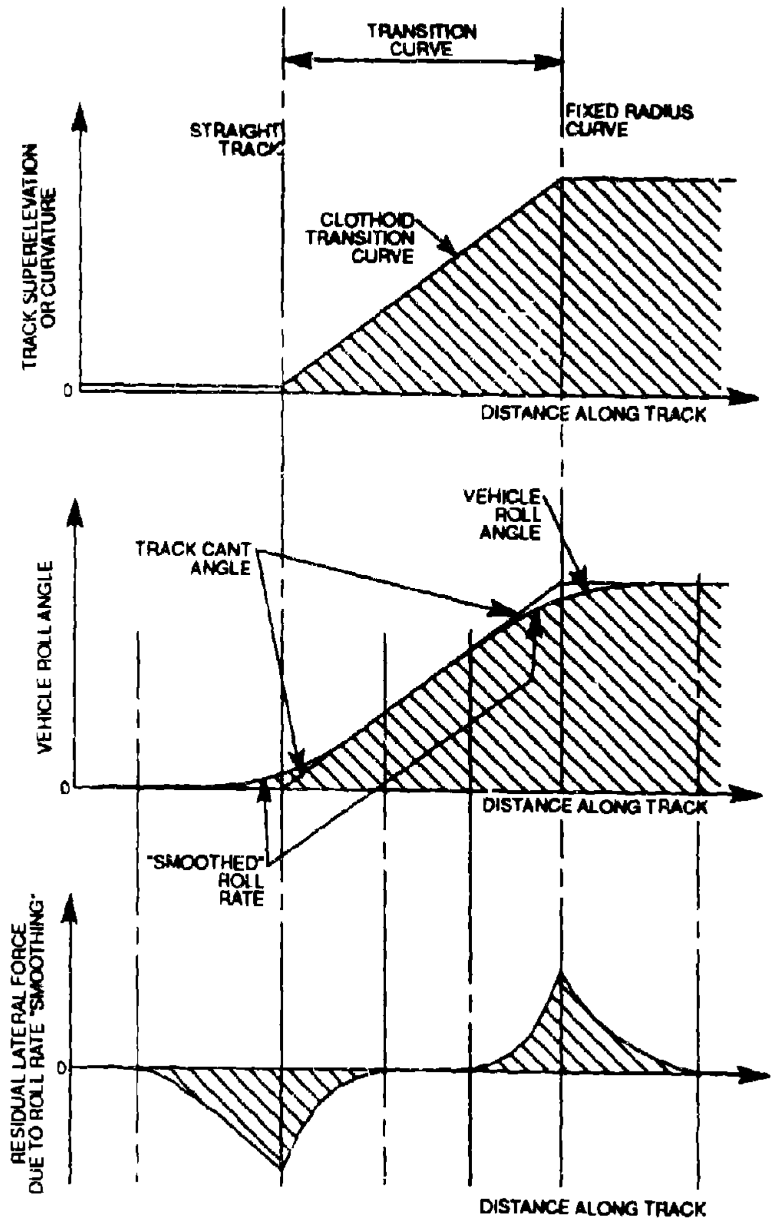

Figure B.15: Effect of Reduced Rate of Change in Roll Rate During Curve Entry and Exit

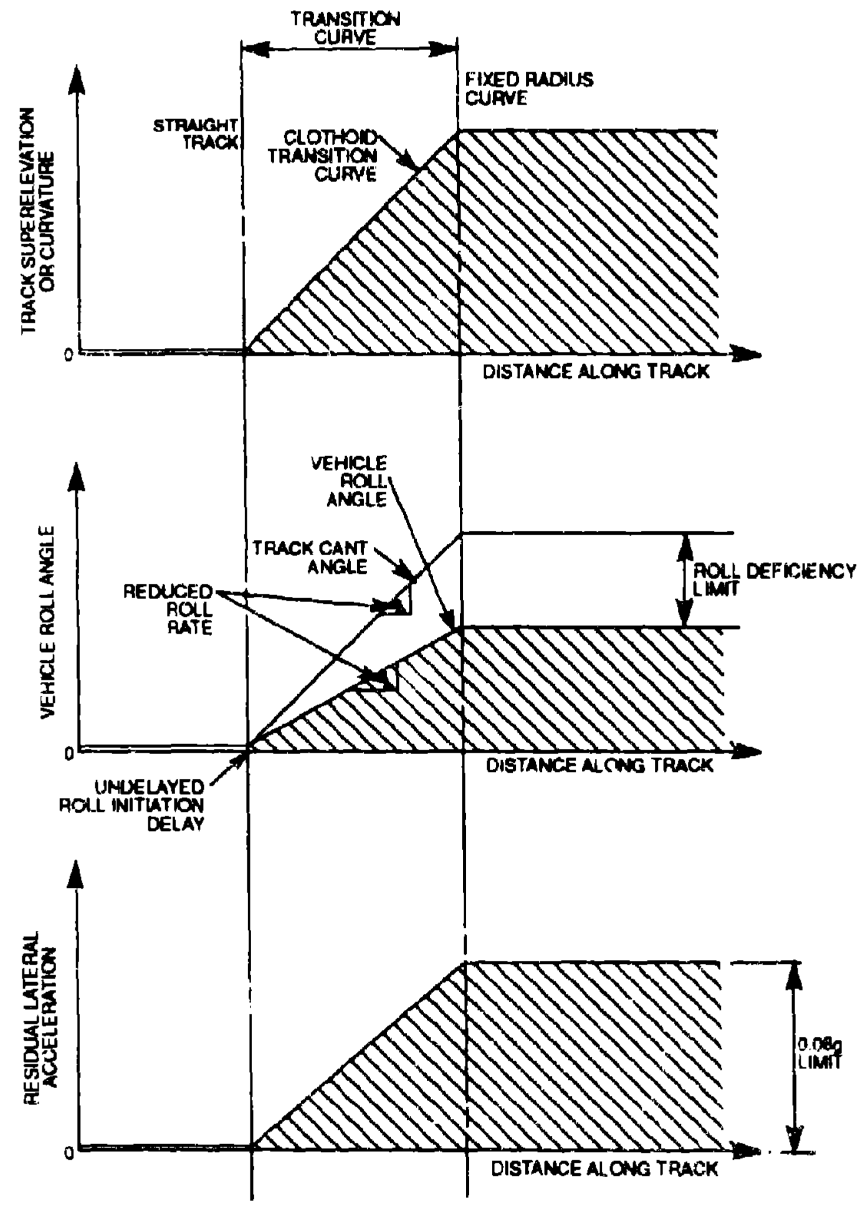

Figure B.16: Effect of Minimizing Body Tilt Angle to Uncompensated Lateral Acceleration Limit for Constant-Radius Portion of Curve

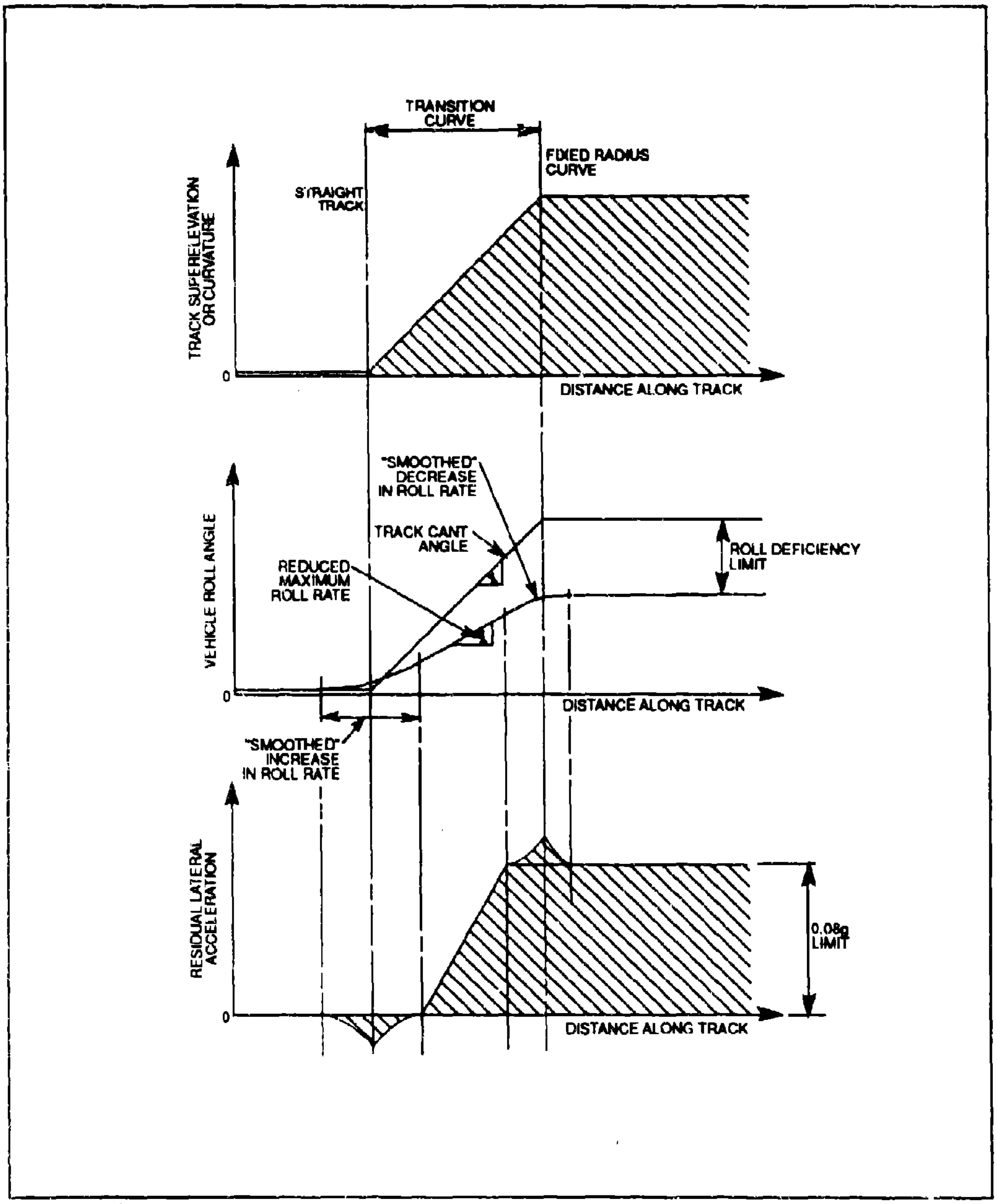

Figure B.17: Integrated Tilt Control Strategy To Enhance Ride Comfort During Transitions

B.5.2.4.3 Active-Tilt Control Strategies As Incorporated In Operational Vehicles

The actual implementation of the tilt control strategies discussed above is complicated by several practical considerations:

- o The presence of defects in local track geometry and track and vehicle components give rise to significant random lateral accelerations which register on the accelerometers that form part of the tilt control system sensor suite; and

- o The inevitable and unavoidable lag between detection of curve onset and the response of the tilt actuation servomechanism.

The random accelerations induced by irregularities in track and vehicle components can be considered as "noise" in the accelerometer signals to the tilt controller. This level of noise can be significant relative to the unbalanced lateral acceleration caused by track curvature, particularly when the track is not well maintained. Using an unfiltered lateral acceleration signal for the control of vehicle tilting systems in combination with a tilt actuator with a rapid reponse time would typically result in unacceptable variations in vehicle roll motion in response to the lateral acceleration signal noise even when *not* traversing track curves. Electronic filtering of the accelerometer signal is incorporated in all of the active-tilting systems based on lateral acceleration detection (the Bombardier/MLW LRC, the Fiat ETR 450 and its derivatives, the German V-610 and Austrian Class 4012, and the ABB X2000) to eliminate this noise and so control unwanted tilt response to track irregularities.

However, the processing of the raw input signal required to achieve this filtering necessarily delays the response of the tilting system to actual track curvature, as illustrated in **Figure B.18**.

This has the unwanted effect of introducing unbalanced lateral acceleration and associated lateral force on passengers during the transient roll phase of vehicle tilting. The degree of filtering of the raw accelerometer signal which is required will depend upon the condition of the track on which the trainset is operating as well as upon the location of the accelerometers on the vehicle. Accelerometers which are mounted on the wheelset axle boxes will generate lateral acceleration signals that are not affected by the vehicle suspension but will contain all of the noise induced by track irregularities. Accelerometers mounted on the truck frame will yield lateral acceleration signals that have been "filtered" by the primary suspension while accelerometers mounted on the vehicle carbody will result in lateral acceleration signals filtered by both the primary and secondary suspensions.

The tilt control system of the Fiat ETR 450 active-tilt EMU trainsets compensates for the delay due to the filtering of the lateral acceleration signal by also sensing the track superele-vation. The track superelevation is detected by gyroscopes mounted on the leading truck of each trainset end car. The gyroscope provides an absolute horizontal datum against which the roll orientation of the truck can be measured continuously. The truck orientation serves as an excellent proxy for the actual angle of superelevation. These roll orientation signals tend to be relatively free of contamination by noise from track irregularities, and so do not need to be filtered. The truck roll orientation signal, which is undelayed by filtering, is used in conjunction with the delayed lateral acceleration signal to control tilting of the ETR 450, so that some of the adverse effects of lateral acceleration signal filtering on ride comfort can be avoided.

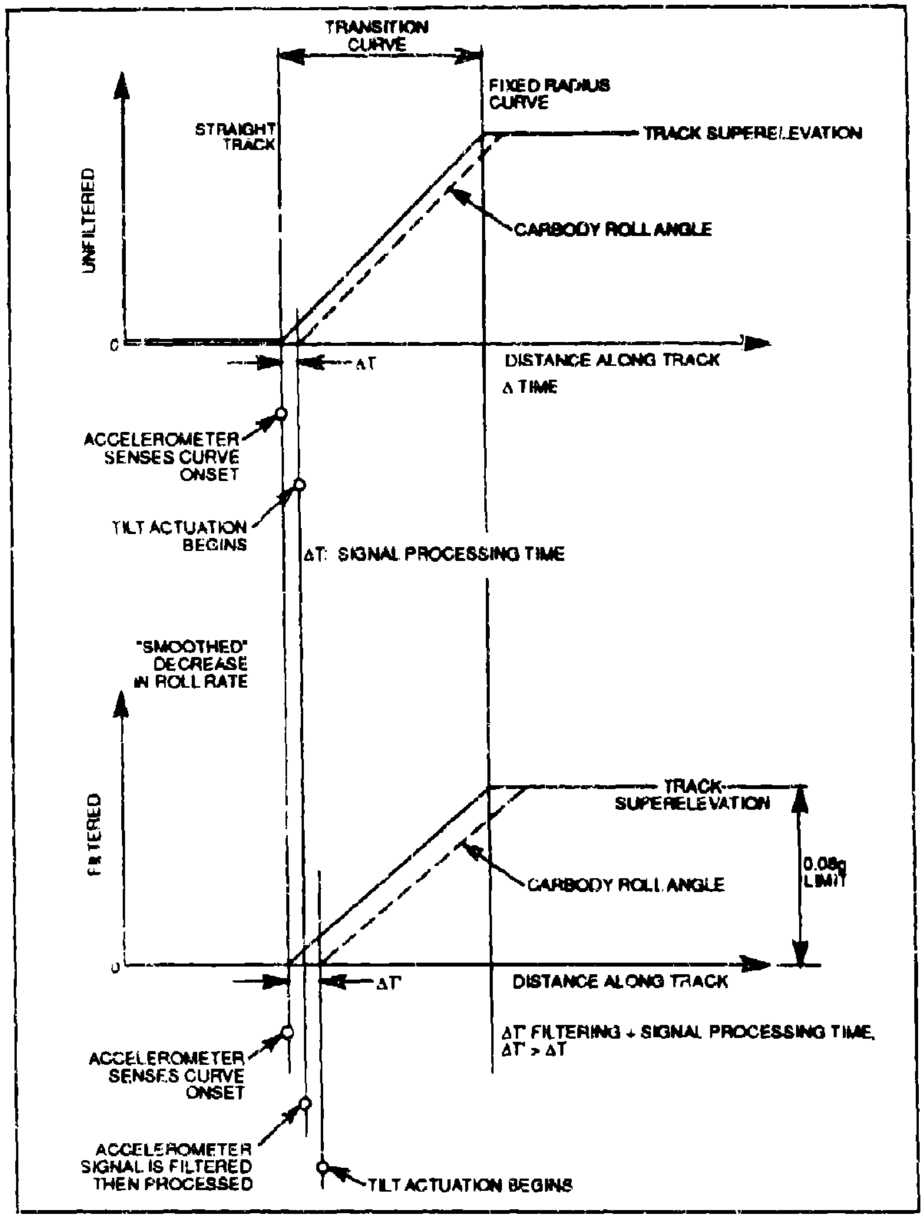

Figure B.18: Effects of Filtering of Accelerometer Signals on Tilt System Response

Another method of compensating for the delay due to signal filtering takes advantage of the fact that trains are made up of a number of discrete vehicles. By using a lateral acceleration sensor mounted on the vehicle next ahead - whether tilting or non-tilting - to provide input for the controller on the following car, the tilting system of each trailing car can be activated in anticipation of curve onset, rather than in response to it. A varient of this approach uses lead vehicle sensors to control tilt onset for all cars in the train. The latter version compensates for signal delays with increasing effectiveness from the front to the rear of the trainset. This type of anticipatory tilt control is incorporated in both the Fiat ETR 450 and its derivatives and the ABB X2000 tilting trainsets.

Perhaps the most promising technique for dealing with the effects of signal noise and lags in control and actuator response time is that used by JR on its modified Sr. 381 EMU and Series 2000 DMU equipment. By applying what amounts to numerical control to its tilting mechanism, the JR strategy decouples tilt onset from real-time curve detection, as shown schematically in **Figure B.19**. Provided that the mathematical representation of the three-dimensional geometry of a specific route in the data file supplied to the on-board computer is accurate, and that the location of the lead vehicle in the trainset also can be determined accurately, this approach allows incorporation of any or all of the techniques for ride quality enhancement in transition curves discussed above.

The use of trackside transponders to update the absolute position of the vehicle at relatively short intervals (the length of a signal block) ensures that minor locational errors cannot propagate to significant levels, while multiple on-board tachometers permit calculation of vehicle position between transponder locations. In combination, this apparently allows very satisfactory and accurate matching of tilt motion to curve entry and exit. Presumably it would not be difficult to incorporate a time-dependent function in the controller to allow a single route data file to be used with different speed profiles, should that be desirable.

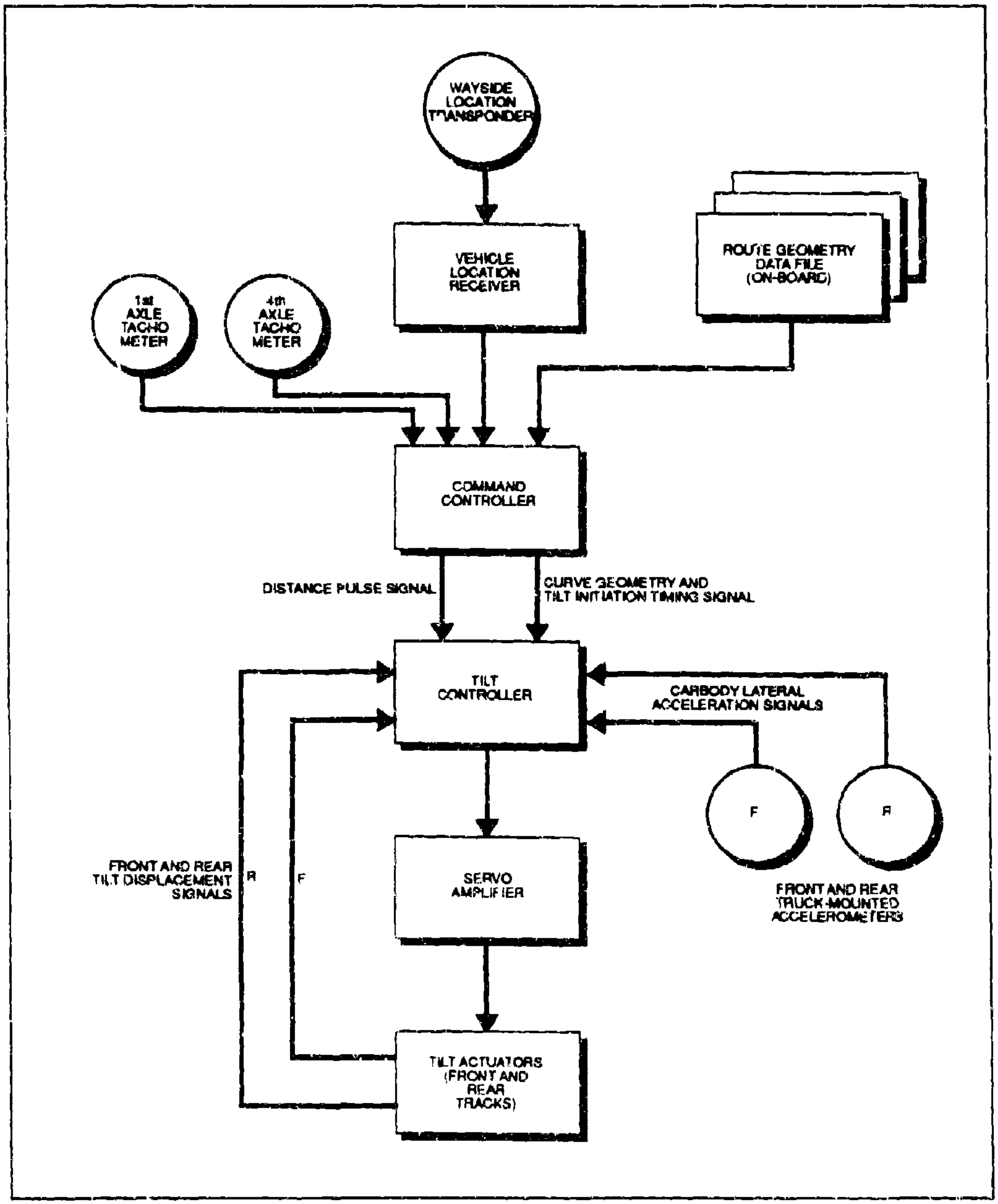

Figure B.19: Schematic of JR Tilt Control System Using Track Data File

APPENDIX C TILT-BODY TECHNOLOGIES DESCRIPTION AND CHARACTERIZATION

C.1 INTRODUCTION

This Appendix presents more or less detailed descriptions and characterizations of tilt-body railway vehicles or subsystems that are currently in service or are under development. The level of detail provided reflects the nature and scope of the technical and operational data made available to the authors by technology suppliers and operators and obtained from the literature. The description and characterization of each technology begins with an overview of its development rationale and test and/or operational history, including the salient characteristics of the infrastructure on which it operates and the types of service provided. A detailed specification of the tilt-body equipment is presented in tabular and graphical form, followed by examination of major subsystems as appropriate (Tilt Control and Actuation, Trucks, Traction, Braking, and Other Features). Where there is substantial commonality of major subsystems among different technologies (as, for example, is the case with the Fiat ETR-450, the German Railways VT 624 DMU and the Austrian State Railways Class 4012 EMU, all of which share the Fiat active-tilt mechanism and truck), these technologies have been grouped for the purposes of this Appendix, to minimize repetition of material.

The reader should be aware that the nature of what is being described in the following subsections varies considerably, from complete trainsets to subsystems. For example, the Fiat ETR-450 is an active-tilt electric multiple-unit (EMU) trainset, while the Talgo *Pendular* passive-tilt coaches could be hauled by electric, diesel-electric or gas-turbine locomotives or power cars. The MLW/Bombardier LRC equipment includes both diesel-electric locomotives and active-tilt coaches, but the locomotives are of essentially conventional (North American) design; VIA Rail Canada, the only fleet operator of the LRC, presently operates their active-tilt coaches with both LRC and other locomotives. The SIG *Neiko* passive-tilt feature forms part of an advanced steerable truck that is under consideration for use by Swiss National Railways, and which could potentially form part of a retrofit package for existing Amtrak passenger equipment or be included in new equipment orders.

The Appendix is structured in three sections. Section C-2 details active-tilt technologies, while Section C-3 contains material on passive-tilt technologies. Section C.4 provides much less detailed capsule summaries of tilt equipment and concepts that for one reason or another have not been pursued.

Note that this Appendix does not deal with tilt-body technologies that have been withdrawn from service (the UAC Turbotrain) or cancelled prior to revenue deployment (the BR APT). While the histories of these technologies are interesting and offer the common lesson that the simultaneous introduction of multiple new and untested subsystems into the railroad environment invites failure, they lie outside the scope of the present investigation.

C.2 ACTIVE-TILT TECHNOLOGIES

C.2.1 THE MLW/BOMBARDIER LRC'

C.2.1.1 Background

Development of the Canadian LRC (Light, Rapid, Comfortable) equipment began in 1968, as a joint venture among Montreal Locomotive Works (MLW), Dominion Foundries and Steel Company (a railway vehicle and truck manufacturer), and the Aluminum Company of Canada (Alcan). This initiative was inspired by a perceived need for fast but rugged and reliable passenger equipment to serve passengers in the Quebec City-Windsor corridor and permit the railways to compete effectively with the airlines and the automobile, without the need for large investments in new alignments and track structure. The initial attempt to meet the need for high-speed equipment to operate on existing tracks - the passive-tilt United Aircraft Turbotrain purchased by Canadian National Railways (CN) - was plagued with poor in-service reliability and high O&M costs, defects that would continue throughout its operating life with CN and later VIA Rail Canada. The consortium recognized the market opportunity for a passenger rail technology that would combine the positive attributes of the *Turbo* (low axle load and unsprung mass. relaxation of passenger comfort limitations on speed through curves, rapid acceleration and deceleration, and a high top speed) with the reliability and maintainability of conventional (i.e., diesel-electric) motive power. Substantial financial support was provided by the Canadian federal government through the former Department of Industry, Trade and Commerce, with both technical and additional financial support from Transport Canada.

The original concept for equipment capable of sustaining 200 km/h (125 mph) operation on existing track encompassed several innovative features, most notably the use of large-section aluminum extrusions for the coach body structure, a suspension system that balanced high-speed stability and curving capability, and a servo-controlled active tilt system. Unlike most high-speed rolling stock, the equipment was designed from the outset to fully comply with AAR, FRA and Transport Canada safety and compatibility requirements.

The development process for this ambitious technology eventually spread over 12 years. Between 1968 and 1972, the LRC concept progressed from preliminary designs to a full-scale prototype locomotive and tilting coach (shown in **Figure C.1**). The coach was unveiled in October 1971, with the locomotive following some five months later, in March 1972. **Table C.1** summarizes the key characteristics of the prototype consist. Note that the original powerplant was the MLW Sr. 251 V-12 diesel, rated at 2163 kW (2900 hp), with 1491 kW (2000 hp) for traction and the balance for the auxiliary alternator. This relatively low traction power was predicated on an

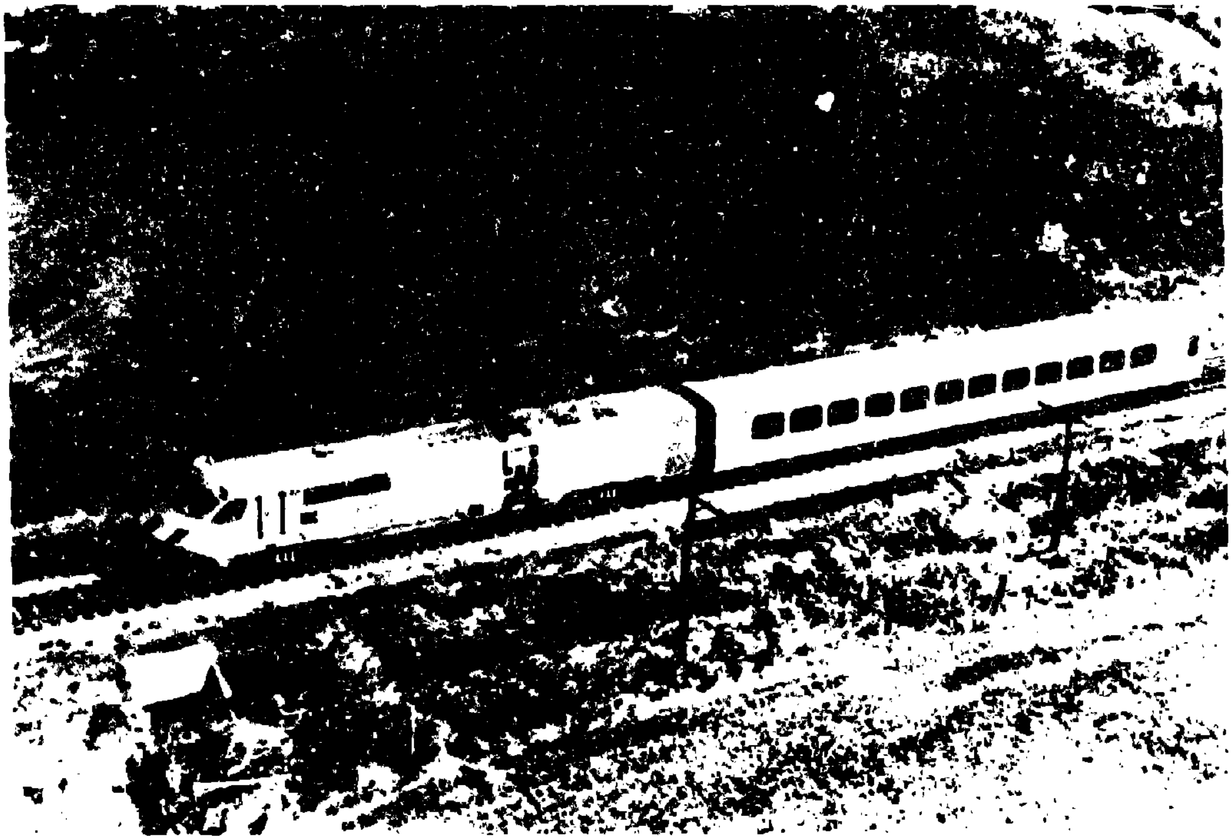

Figure C.1: The Prototype LRC Locomotive and Coach

initial presumption that the equipment would operate in push-pull consists, with two locomotives and four to eight intermediate coaches[1].

Between the Spring of 1972 and 1975, the prototype consist underwent extensive testing, initially in Canada, then at the Transportation Test Center in Pueblo, Colorado during 1974, where it reached a top speed of 210 km/h (130 mph) during some 33,600 km (21,000 miles) of high-speed running. Subsequently, the prototype consist was used to "show the flag" with demonstration visits to major centers across North America

Beginning in March 1975, six months of revenue service operational testing was carried out as part of CN passenger service between Toronto and Sarnia, Ontario. The LRC locomotive hauled the prototype coach, still fully instrumented, plus six lightweight aluminum *Tempo* coaches.

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[1] "Canada's LRC Prototype Coach on Trial," Railway Gazette International, November 1971. In any event, this presumption proved incorrect, to the great disadvantage of the LRC and its developers.

TABLE C.1: LRC PROTOTYPE ATTRIBUTES

ATTRIBUTE	LOCOMOTIVE	COACH
DIMENSIONS	20.7m *l* x 3.19m *w* x 3.48m *h* (67'11"*l* x 10'5.625"*w* x 11'5"*h*)	25.6m *l* x 3.19m *w* x 3.48m *h* (84'*l* x 10'5.625"*w* x 11'5"*h*)
STRUCTURE	Steel with Aluminum Shell	Aluminum
SUSPENSION	Rubber Chevron Primary, Flexicoil Secondary	Rubber Chevron Primary, Airbag Secondary
WEIGHT	81,650 kg (design); 97,520 kg (actual) (180,000 lb) (215,000 lb)	40,820 kg (90,000 lb)
PROPULSION	SR. 251 V-12 2163 kW diesel, 4 axle-mounted dc traction motors	N/A
TILT MECHANISM	N/A	Accelerometer, hydraulic actuation, up to 10° active bank

The acquisition of MLW by Bombardier, Inc., in 1975 signalled the start of an aggressive continent-wide marketing campaign for the LRC. This strong initiative led to two apparent early triumphs. VIA Rail Canada, Inc., the Canadian national passenger operator created in 1977, ordered 22 locomotives and 50 coaches for delivery starting in 1981, as the first step in replacement of the largely antiquated equipment (the equivalent of Amtrak's Heritage Fleet) purchased from CN and Canadian Pacific (CP). Amtrak itself agreed to lease a pair of LRC trainsets (one locomotive and five coaches each) for a period of two years, with delivery in 1980, with the option for a subsequent fleet purchase.

However, while the production LRC coaches were essentially similar to the original design, the locomotives as ordered by (and delivered to) Amtrak and VIA Rail Canada differed substantially from the prototypes tested prior to 1978.

The principal difference was in the powerplant. To accommodate the hotel power requirements specified by Amtrak and VIA (for heating, air conditioning, lighting, galley equipment and so on), as well as the normal auxiliary loads, and to permit single-locomotive operation with up to five coaches, the original V-12 diesel was replaced by a much more powerful (and much heavier) 2780 kW (3725 hp) V-16 powerplant.[2] The gross weight of the locomotive rose from the 98 tonnes (107.5 tons) of the prototype[3] (itself well above the design weight of 81 tonnes (90 tons[4]) to about 115 tonnes (127 tons[5]) for the production equipment.

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[2] Progressive Railroading, August 1980.

[3] LRC - A System For Today, MLW Industries Marketing Brochure, 1975.

[4] Op. Cit., Railway Gazette International, November 1971.

The effect of this weight increase on static axle load was significant; from a design load of just 20.5 tonnes/axle (22.5 tons/axle[6]), the static load for the locomotive increased to 28.75 tonnes/axle (31.75 tons/axle[7]). This increase in static load exacerbated track forces caused by the already high unsprung mass of the LRC. Retention of the axle-mounted dc traction motors, that were then (and remain) standard practice in North American locomotive design, instead of opting for frame-hung traction motors as used in the contemporary HST diesel-electric power car, resulted in a maximum unsprung mass of 4.02 tonnes (4.42 tons) for the powered axles.[8] This compares to just 2.2 tonnes (2.42 tons) for the HST.[9]

This increase in locomotive weight and attendant track forces had the effect of neutralizing the principal advantage of the tilting coach - an increase in allowable speed in curves with limits imposed for reasons of passenger comfort - since the increased forces exerted by the locomotive would, in some instances, exceed *safety-related* limits with even a modest increase in speed.

Significantly, although the axle load of the LRC coach also increased - from a design load of just over 9 tonnes (10 tons) per axle to about 12 tonnes (13.25 tons) per axle, even the higher value is well within the upper limit for high-speed operation and the maximum unsprung mass of a coach axle (1.13 tonnes/1.24 tons) is lower than that of the Mk III Coach used by British Rail in HST consists (1.38 tonnes/1.52 tons).[10] Since it is the active-tilting coaches that are of primary interest in the current assessment, rather than the non-tilting locomotives, this difference is very important.

In any event, both the Amtrak and the VIA Rail deployment met with mixed success. Amtrak accepted and tested its two leased trainsets on revenue track in the Northeast Corridor during the period from 1980 to mid-1981. Although success of the LRC trainset from an overall system perspective is difficult to ascertain, they did obtain rather good results with respect to the tilt capability. The tests confirmed that the speed of the LRC trainset on curves could be increased to cant deficiencies of nine inches without exceeding safety criteria or sacrificing passenger comfort. (Parallel testing of non-tilting Amcoaches demonstrated safe operation at similar cant deficiencies but with much reduced ride quality). Nevertheless, at the end of the lease period,

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[5] Engineering Master Report for LRC-Loco and LRC-Cars, VIA Rail Canada, July 1985.

[6] Op. Cit., Railway Gazette International, November 1971.

[7] Op. Cit., VIA RAIL Canada.

[8] Sub-Study E: Rolling Stock Assessment, Calgary-Edmonton Intercity Passenger Rail Study - Phase III, Alberta Department of Economic Development, January 1985.

[9] Ibid.

[10] Ibid.

Amtrak, with its limited budget, elected not to exercise its option to purchase, and returned the sets to Bombardier.[11]

Although VIA continues to operate the LRC coaches, and even purchased a second batch of 10 locomotives and 50 coaches with delivery in 1984-85, the overall operational history of this equipment with VIA has not been an outstanding success. Shortly after the commencement of revenue operations in the fall of 1981, several persistent problems became apparent. The active-tilt mechanism proved to be unreliable, the automatic plug doors and steps failed to operate correctly, there were major problems with the train-line power distribution subsystem, and the locomotives themselves suffered from frequent in-service breakdowns.

These failures were exacerbated by speed restrictions imposed by CN and later by CP on operation of the LRC over their respective tracks. Where conventional equipment was permitted to run at the 155 km/h (95 mph) limit imposed by the CTC for operation on track with at-grade road crossings, the LRC was limited to 128 km/h (80 mph), due to the excessive track forces generated by the high axle load and unsprung mass. This limit was later removed following modifications to the locomotive suspension, but the combination of these problems hindered the initially positive effects of the new equipment on market response. The acceptability of track forces generated by the LRC locomotive remains marginally acceptable to the operating railways.

VIA was forced to withdraw the equipment from service temporarily to deal with the door and electrical problems, and elected to "lock out" the tilt mechanism. The LRC equipment was reintroduced in mid-1982, but the problems had not been eliminated, and contributed to a major reliability crisis in the winter of 1983.

In fact, the locomotive problems continued to persist - LRC locomotive maintenance costs were significantly higher than those for the 30- to 40-year-old units obtained from CN and CP[12]. When VIA operations were reduced by the Canadian government, 10 LRC locomotives were kept in service, although more will be returned to service in the near future. The coaches remain the mainstay of the Quebec City-Windsor Corridor operations, pulled by both LRC and F40PH units obtained in the latter portion of the 1980s. On most Corridor routes, the coaches are now operated with fully active tilting, after Bombardier undertook redesign and retrofit of an improved tilt subsystem during 1985-87 [13]

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[11] "Amtrak Wins A Big One," p.14, Railway Age, 29 June 1981.

[12] Reliability, Maintainability and Safety - Prototype Train - Preliminary Report, VIA Rail Canada, August 1984, and Report of the Inquiry Into the On-Time Performance of VIA Rail Canada, Inc., Canadian Transport Commission, October 1984

[13] "Banking Performance Curves for the LRC Car Fleet," personal correspondence, R. Monette, Maintenance Operations, VIA Rail Canada Inc., 1992.

In the Spring of 1988, LRC cars with the improved tilt system were provided by Bombardier and VIA Rail Canada for demonstration on the Boston-New York segment of the Northeast Corridor, as part of the on-going CONEG high-speed equipment demonstration project. Tests conducted with the LRC trainset (among others) again verified that safe operation at cant deficiencies of eight inches could be sustained without undue effect on passenger comfort. The steady-state lateral acceleration of the LRC coach was sustained near zero "g" in curves by its active tilt system, with only some low-level jolts in curve entry and exit transitions. A passenger survey, taken as part of the test sequence, rated the ride quality as excellent.

Review of all test data led to the conclusion by Amtrak/CONEG that trip time could be significantly reduced by operating tilt-body coaches at high cant deficiency speeds without compromising passenger comfort and derailment safety limits.

Bombardier sold MLW to General Electric in 1989, purchased the North American license for the GEC-Alsthom TGV, and has since concentrated its efforts on achieving deployment of the latter technology in the U.S. and/or Canada. The LRC coach and its subsystems are available, but are not being marketed as aggressively as in the past.

In view of the conventional aspects of the (non-tilting) locomotive, the following detailed technical description focuses on the LRC coach.

C.2.1.2 Technical Specification - LRC Coach

The LRC coach as supplied to VIA Rail Canada, Inc., is illustrated in **Figure C.2**. The characteristics of the coach are summarized in **Table C.2**.

C.2.1.3 Coach Trucks

The trucks for the LRC coach were developed by Dofasco Ltd., through extension and refinement of then current railroad truck design practices, with the addition of an active banking system that itself was based in part on earlier patents.[14] The trucks and banking mechanism are shown in **Figure C.3**.

The truck frame itself is a rigid cast-steel H-frame. The angled, laminated metalastik rubber chevron-spring primary suspension is mounted on this frame to cushion axle motions in yaw and in lateral translation. Bounce and pitch motions are controlled by rotary hydraulic dampers mounted on the axle boxes. The trucks are equipped with forged-steel axles and 762 mm (30 in) rolled-steel wheels. Each coach axle is equipped with two ventilated brake discs located inboard between the axle journal bearings. Tread brakes are also fitted.

[14] "Canada's Latest Supertrain," Business Week, February 17, 1975.

Figure C.2: LRC Coach as Produced For VIA Rail Canada, Inc.

The secondary suspension is provided by large-diameter rolling diaphragm air springs. The air springs are separated by a distance of 2235 mm (7 ft, 4 in) laterally, the maximum allowed by the coach structure, and are mounted as close to the coach floor as possible to enhance stability. The lines connecting the air springs to the air source are equipped with chokes to provide pneumatic damping of roll motion; this is supplemented by lightweight hydraulic dampers. The air supply to the air springs is controlled to maintain a constant suspension height and frequency regardless of load. This suspension design reflects an effective solution to the tradeoffs among high-speed running stability (resistance to hunting), control of lateral and vertical track forces and L/V ratio during curving and maintenance of acceptable passenger ride comfort.

The LRC truck incorporates two bolsters, also shown in **Figure C.3**. The lower bolster is connected to the truck frame by modified swing links to permit roll rotation of the carbody relative to the plane of the truck frame. This linkage creates a virtual roll axis about 250 mm (10 in) above the coach floor, at a point slightly below the center of gravity of the coach. The geometry of this link arrangement avoids significant displacement of the roll center, and ensures static stability by forcing any displacement of the roll center to act upwards on the carbody against the force of gravity. This arrangement permits carbody tilting for lateral acceleration compensation through curves with the roll center close to seat level for maximum passenger comfort.

TABLE C.2: TECHNICAL SPECIFICATION OF LRC PRODUCTION COACH

ATTRIBUTE	DETAILS
Exterior Dimensions	Overall Length: 28m (85'4"); Height: 3.658m (12'); Width: 3.191m (10'5.625"); Distance between truck centers: 18.136m (59'6");
Body Structure	Welded all-aluminum stressed-skin design with large-section tubular side sills and fabricated horizontal shear structures; main longitudinal framing members are continuous over car length; structure fully complies with FRA safety regulations and AAR requirements
Weight (Empty/Loaded)	42.7 tonnes (47 tons)/48.1 tonnes (53 tons)
New Wheel Diameter	0.762m (30")
Brakes	Pneumatic disc and caliper tread, 2 discs per axle
Truck Design	Two-axle, rigid one-piece frame; metalastik chevron primary suspension; widely-spaced, low-rate airbag secondary suspension; dual bolster design
Tilt Mechanism	Accelerometer-controlled sensor on each truck; servo-controlled roll bolster for up to 10 degrees unbalanced tilt (8.5 degrees net of differential suspension compression); hydraulic tilt actuator; roll center is 25 cm (10") above coach floor and slightly below C_g.
Couplers	Standard type H tightlock couplers, .876m (34.5") above rail
On-board Power	Head-end electrical, 480v 60Hz 3-phase AC; DC transformers for door and step operation
On-board Amenities	Heating 36kW/car; hot air flow rate 1000ft^3/min; AC 12-ton unit 2800ft^3/min, 1:2 fresh/recirculated; at seat and general lighting; lockable airline-style overhead luggage bins.
Seating	Coach: 84 seats/car, 2 + 2 seating; club car (1st class) 68/72 seats/car, 2 + 1 seating; both car types have provision for wheelchair tiedown, disabled-accessible toilets
Design Speed	200 km/h (125 mph)
Service Speed	155 km/h (95 mph)

The linkage between the lower bolster and the truck frame also provides for some lateral displacement towards the inside of the track curve to reduce any destabilizing tendancy that could increase the risk of car rollover due to the tilting action.

The air springs rest on a transverse spring plank connected to the carbody by traction bars and a transverse locating link. The spring plank is supported by the banking bolsters through four laminated-rubber bearing pads. The truck center-post projects above the banking bolster to engage precompressed laminated-rubber traction pads attached to the spring plank. Lateral suspension is provided by the combined shear resistance of the traction and spring-plank pads; the latter also control truck rotation as the spring plank is fixed and cannot pivot.

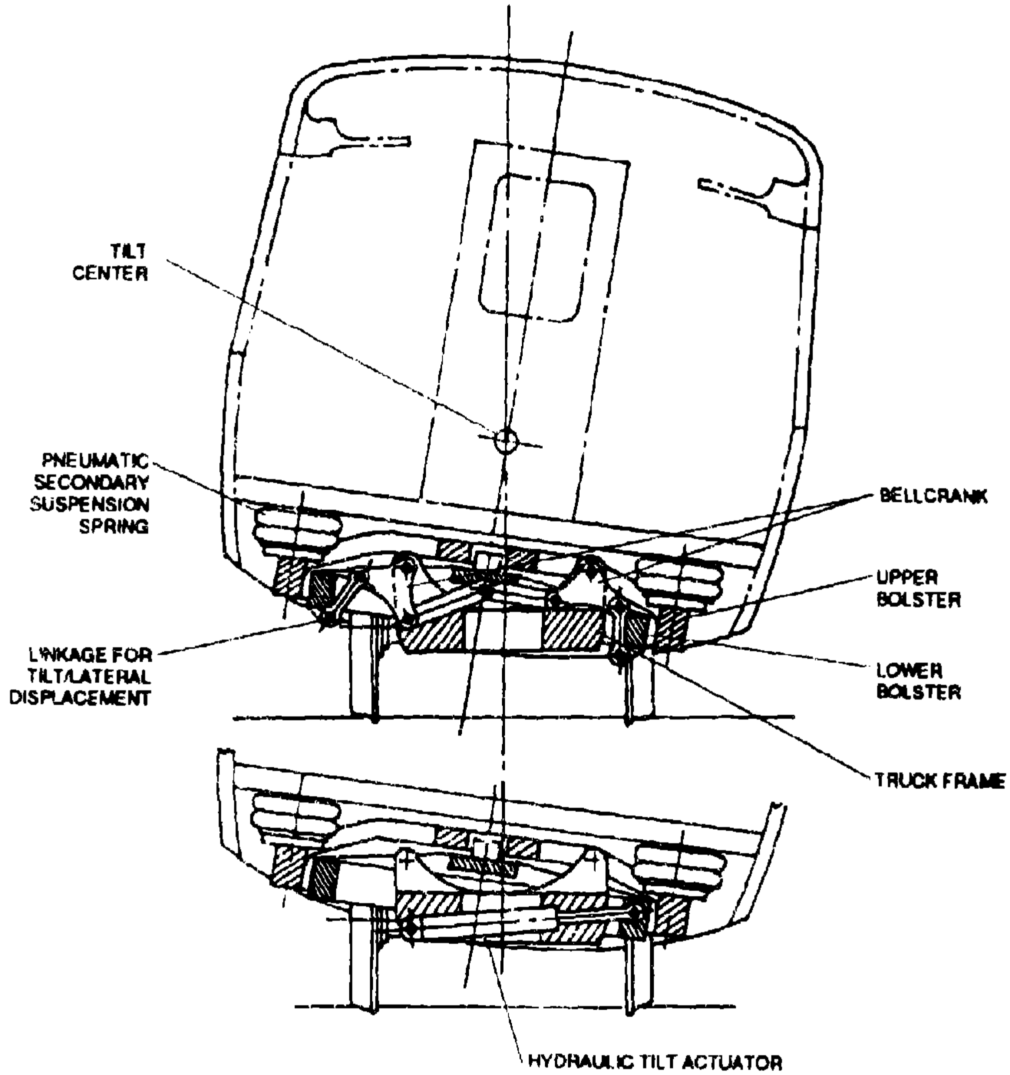


Figure C.3: The LRC Coach Truck and Active Body Tilting Mechanism

When traversing a small-radius curve at speed, the differential compression of the suspension elements will result in an effective negative superelevation of as much as 1.5 °.

C.2.1.4 Tilt Control and Actuation Subsystem

The tilt control and actuation subsystem as originally fitted to the LRC coach was based on an electro-hydraulic servo-loop driven by signals from accelerometers mounted on each truck, as shown in **Figure C.4**.

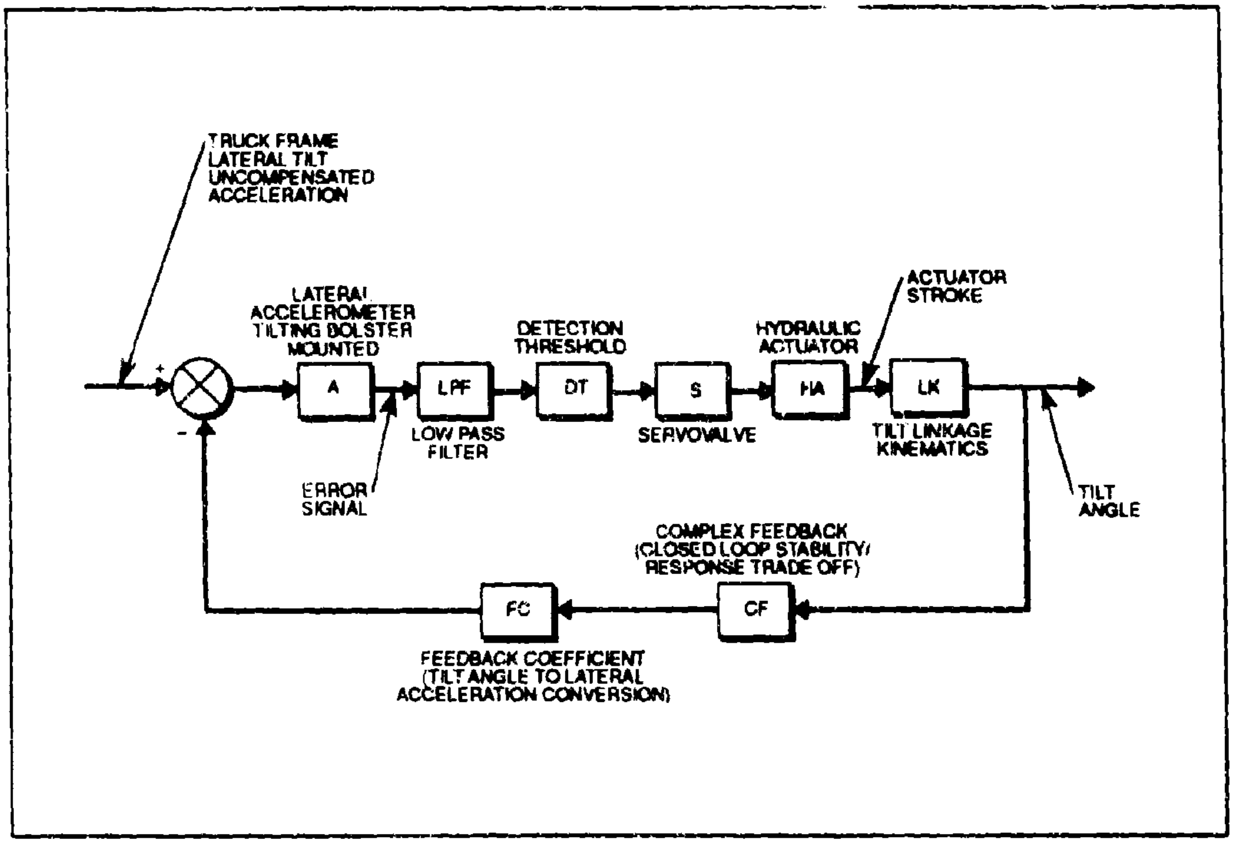


Figure C.4: LRC Feedback Tilt Control Schematic

The accelerometer was mounted on the banking bolster with its sensitive axis aligned so as to measure lateral acceleration in the plane of the bolster.[15]

The control strategy for the LRC tilt mechanism is based on reduction of the uncompensated lateral acceleration, as detected by the accelerometer mounted on the tilting bolster, to a predetermined limit (between 0.05g and 0.08g) through tilting of the carbody. The accelerometer signal opens the electro-hydraulic servovalve, which permits high-pressure hydraulic fluid to enter the hydraulic tilt actuators and bank the carbody.

As the bank angle of the carbody increases, the uncompensated lateral acceleration sensed by the accelerometer also decreases. The tilt controller incorporates a feedback control, also as illustrated in **Figure C.4**, so that the rate of banking is reduced as the difference between the

[15] Improved Passenger Equipment Evaluation Program - Train System Review Report, Volume 8, LRC (Canada), U.S. Department of Transportation, Federal Railway Administration, Report No. 80/14.VIII, March 1979; and The LRC Coach Trucks and Suspension, W.H. ElMaraghy, J.A. Gaiser and H.J. Bexon, ASME Paper No. 79-RT-4, 1979 (paper presented at Joint ASME/IEEE Railroad Conference, Colorado Springs, April, 1979.

measured uncompensated lateral acceleration and the preset threshold approaches zero, and banking stops once the limit is achieved.

The resulting closed-loop tilt control system, with the accelerometer within the feedback loop, is quite sensitive and in principle will compensate not only for cant deficiency, but also for an excess of superelevation at low-speed. The sensitivity of the controller has made it essential to apply appropriate feedback compensation techniques in an attempt to achieve an acceptable balance between closed loop stability and response sensitivity and time.

The banking system was designed to provide up to 10 degrees of bank angle, excluding the effects of suspension compression, which in practice restricted the effective tilt to about 8.5 degrees. This permits operation with a maximum uncompensated acceleration of 0.23g, based on specification of 0.08g as the uncompensated lateral acceleration limit in the feedback loop.

Tilt actuation is achieved by means of two hydraulic cylinders on each truck mounted diagonally between the truck frame lower bolster and the upper tilt bolster, as shown in **Figure C.3**. The high-pressure hydraulic fluid is provided by a hydraulic power pack, consisting of an electric motor, pump, fluid reservoir, accumulator, pressure regulator and filter mounted under the floor of each tilting coach.

The mechanical linkage between carbody and truck frame is quite complicated, comprising interconnecting hanger links, bell cranks and swing links, again as shown in **Figure C.3**.

C.2.1.5 Braking

As noted above, the coaches are equipped with conventional airbrakes acting on two discs per axle, plus tread brakes on each wheel. The brakes were designed to provide a service braking rate of 0.08g and an emergency braking rate of 0.11g.

C.2.1.6 Other Features

The other major feature of the LRC coach is its welded all-aluminum body structure, which combines both relatively light weight and complete compliance with FRA and Transport Canada safety standards and also with the requirements set out in Section C of the AAR Manual of Standards and Recommended Practices for trains weighing in excess of 272,200 kg (600,000 lb).

Structurally, the LRC coach is a stiffened stressed skin aluminum tube with continuous large-section extrusions of AA-7004 alloy forming the tubular side sills and other framing members, AA-5083 alloy plate in the sides and bottom plating, and AA-5052 alloy plate in the roof. The skin thickness varies from 2.6 mm (0.102 in) in the roof to 4.76 mm (.1875 in) on the coach sides.

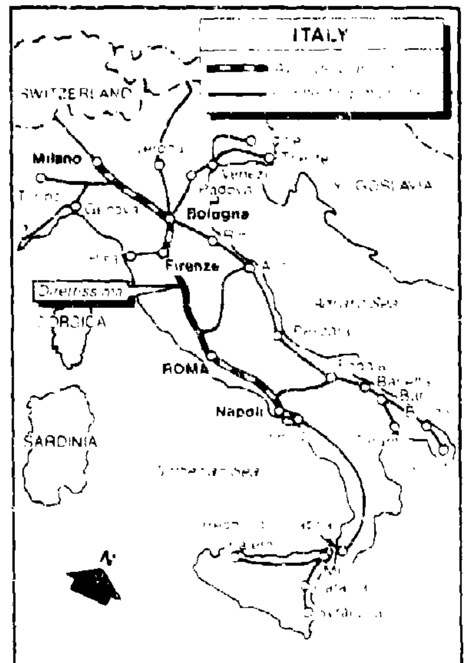
To comply with the FRA requirement for resistance to a 45.5 tonne (50 ton) vertical load applied between the bolster and the car end, the LRC design team avoided the conventional solution of reinforcing the draft sill to carry the load. Instead, the draft sill was eliminated, and deep-section transverse horizontal plate girders were used at the car ends to transfer the compressive force to the strengthened solebars. This approach resulted in much stronger and stiffer side frames, a very desireable outcome in a coach designed for high-speed operation. It also had the effect of creating, at floor level, a very strong perimeter frame that is resistent to lateral impact, while creating an unobstructed and well-protected space between the solebars for installation of auxiliary equipment.

Under test at the full 363,300 kg (800,000 lb) compressive load, the body structure shortened elastically by about 50 mm (2 in).

C.2.2 THE FIAT ETR-450 AND DERIVATIVES[b]

C.2.2.1 Background

In the early 1970s, Italian State Railways (FS) were faced with the same problems encountered by passenger carriers in prosperous countries worldwide - burgeoning automobile ownership and highly competitive air carriers, compounded by growing congestion on main lines forced to carry both passenger and freight traffic. FS responded with a plan to build what they then described as "Europe's first Shinkansen" - a new high-speed line between Rome and Florence, called the *Direttisima*, that would eliminate the severe curvature and gradients that characterized the conventional line up the Italian boot, and allow 156 mph operation.



(Source: Railway Gazette International, June 1989)

Figure C.5: Existing and Planned Italian High-Speed Lines

This original segment of what is now planned as the FS' *Alta Velocita* network - shown in **Figure C.5** - has encountered almost endless problems, ranging from opposition by Communist trade unions to a premium-priced service through geological surprises in tunnels and to cost overruns and financial difficulties. Almost two decades later, about 28 miles of the line, including a viaduct over the Arno river and a section of tunnel, are still not complete, although FS has already started preliminary engineering for extensions north to Bologna and Milan and south to Naples and Salerno. Eventually, an east-west line will be added, connecting Turin and Venice to the north-south spine.

Interestingly, the new lines are not dedicated to high-speed operations. FS claims that mixed operations are essential if it is to achieve a satifactory return on its investment. However, mixed in this context refers primarily to 80 to 125 mph passenger services rather than to freight, although some low-axle-load, high-speed freight is also operated. FS attempts to "fleet" trains with similar operating characteristics to improve dispatching efficiency and system throughput.

The mainstay of the FS high-speed services at present is the Fiat-built ETR-450 tilt-body EMU (**Figure C.6**), whose layout is illustrated in **Figure C.7**.

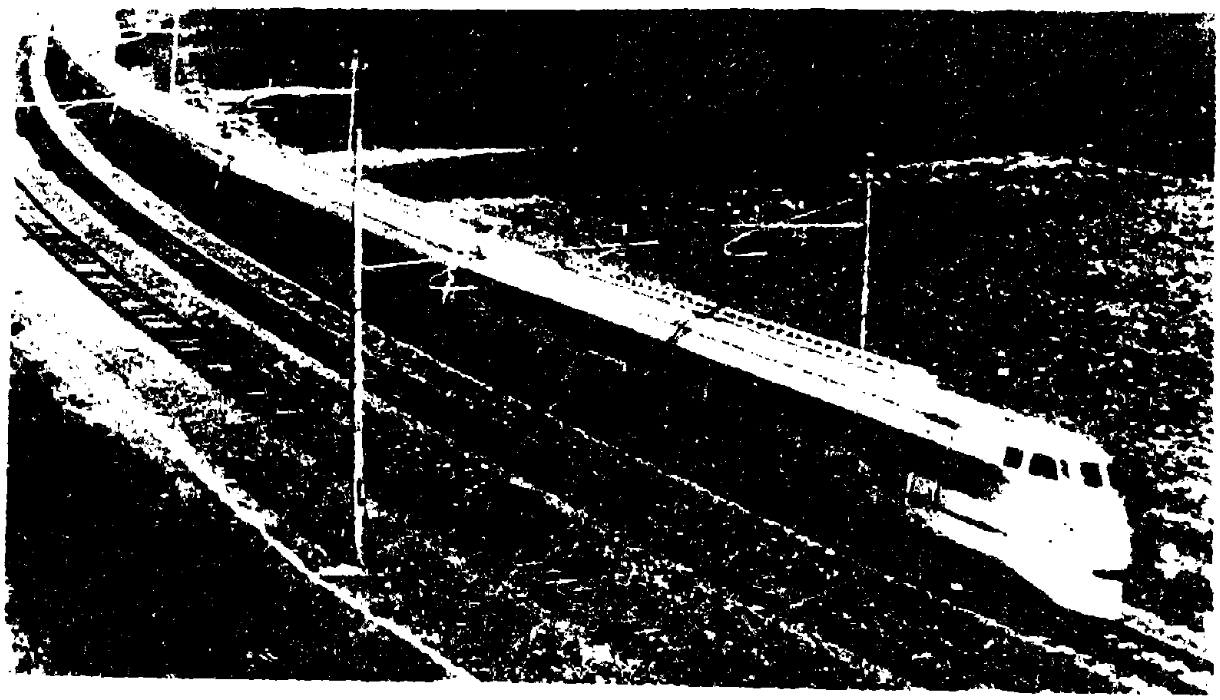


Figure C.6: The Fiat ETR-450 Tilt-Body EMU Trainset

This trainset was derived from the ETR-401 prototype. Commissioned during the early stages of *Direttissima* construction, the ETR-401 eventually operated for almost 220,000 miles in revenue service over a six-year period between 1976 and 1982, while waiting for enough new line to become operational for high-speed tests. By 1986, it had covered a further 156,000 miles in high-speed/high curvature testing.

In 1986, FS ordered a total of 130 vehicles, configured as ten 11-car trainsets and four 5-car trainsets. The production ETR-450s entered service between Rome and Milan (partly on *Direttissima*, partly on conventional lines) in 1988. Service between Rome and Naples was

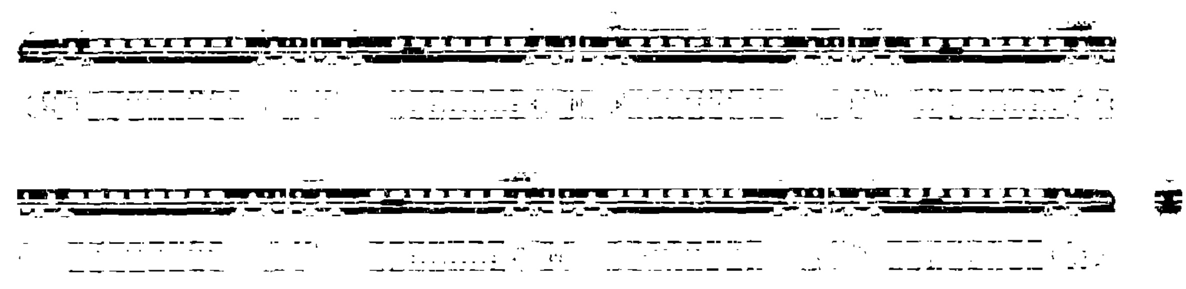


Figure C.7: Layout of the Fiat ETR-450 Tilt-Body Trainset

added in 1990, as delivery of the ETR fleet neared completion. **Table C.3** summarizes the major milestones in the development and operational history of Fiat tilt-body equipment. There have been trials on several railways outside Italy.

Active-tilt DMU and EMU sets based on the Fiat truck and body-tilting mechanism are under construction for Germany and Austria, respectively, and the Swiss are considering acquiring a dual-voltage version of the ETR-450 for cross-border services to Milan.

The characteristics of the *Direttissima* track and alignments over which this equipment operates are summarized in **Table C.4**. **Figure C.8** illustrates typical cross-sections for track and 3kV dc electrification on embankment and in cut (top), on viaduct (middle), and in tunnel (bottom).

C.2.2.2 Technical Specification

Table C.5 summarizes the key characteristics of the ETR-450 and also the conceptual design for the next-generation "Avril" (from *Alta Velocita a Ruote Indipendenti Leggero* - high-speed, independent-wheel, lightweight) tilt-body EMU.

Clearly, the most interesting feature of the ETR-450 is its active body-tilting mechanism. Achievement of reliable tilting with an acceptable level of passenger comfort is no small accomplishment, as witnessed by the demise of the APT in Britain, and the on-going problems with coach tilting on the LRC. As discussed at some length in Appendices A and B to this report, a major part of the challenge is ensuring that the tilting process is initiated at exactly the right time (slightly before entering the run-on or run-off transition) and that the rate of change of tilt parallels exactly the rate of change in curvature and superelevation of the track.

Table C.3: MILESTONES IN DEVELOPMENT AND OPERATION OF FIAT TILT-BODY RAIL VEHICLES

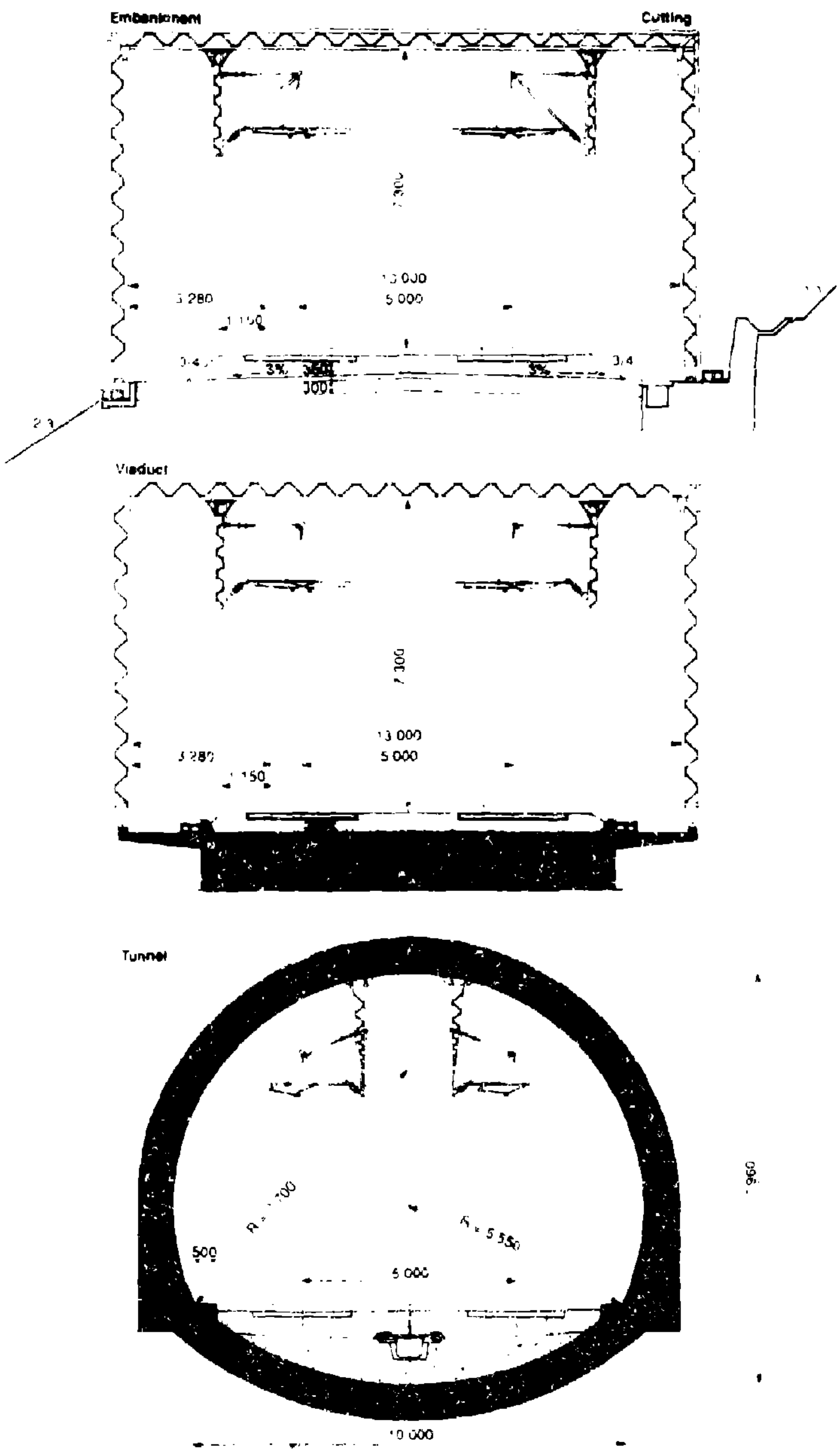
o 1968-70	Survey of Italian railway lines to identify potential for tilting train application; parametric analysis of tilt system performance requirements
o 1971-74	Extensive testing of active-tilting single-car Y0160 prototype
o 1974-76	Design and construction of 4-car 171-seat ETR 401 prototype active tilting revenue trainset
o 1976-82	Extensive in-service operation by the Italian State Railways (FS) of the ETR 401 prototype (approximately 350,000 km [219,000 miles]) at conventional speeds
o 1984-86	High speed (up to 250 km/h [156 mph]) testing of the ETR 401 prototype trainset (approximately 250,000 km [156,000 miles])
o 1985	FS orders four ETR-450 production trainsets of 11 cars each, based on the ETR-401 prototype
o 1986	FS increases original order to 14 ETR 450 trainsets, ten consists with 11 cars each and four consists with 5 cars each
o 1987	ETR 450 trainsets enter revenue service with FS on Rome-Milan route
o 1987-89	ETR 401 prototype trainset tested by state railways in Austria, Germany, Yugoslavia and Czechoslovakia
o 1989	FS extends ETR 450 services to Venice, Naples, Turin and Salerno
o 1989	German State Railways (DB) orders 10 two-car tilt-body DMU sets equipped with Fiat trucks and tilt mechanisms
o 1989-90	Feasibility study of dual-voltage 8-car tilting trainsets for Swiss Federal Railways (SBB); Austrian State Railways (OBB) orders three six-car EMU sets with Fiat trucks and tilt mechanisms
o 1990	Fiat completes construction of experimental high-speed coach with independent-wheel bodies; proposes development of next-generation 320 km/h (200 mph) tilt-train designated Avril
o 1991	FS orders six additional ETR-450 trainsets

TABLE C.4: CHARACTERISTICS OF ITALIAN HIGH-SPEED LINES

CHARACTERISTIC	ROME - FLORENCE	FLORENCE-BOLOGNA	BOLOGNA MILAN	ROME-NAPLES
Length	355 km (222 mi)	90 km (56 mi)	185 km (115 mi)	211 km (132 mi)
Status	310 km (194 mi) complete, 45 (28) under construction	planned	planned	planned
Design Speed	300 km/h (186 mph)	300 km/h (186 mph)	300 km/h (186 mph)	300 km/h (186 mph)
Service Speeds				
Conventional EMU	160 km/h (100 mph)	160 km/h (100 mph)	160 km/h (100 mph)	160 km/h (100 mph)
Conventional Locomotive	200 km/h (125 mph)	200 km/h (125 mph)	200 km/h (125 mph)	200 km/h (125 mph)
ETR-450	250 km/h (156 mph)	250 km/h (156 mph)	250 km/h (156 mph)	250 km/h (156 mph)
Minimum Curve Radius				
Design - 300 km/h (186 mph)	Not Available	5460m (17,800')	5460m (17,800')	17,800'
- 250 km/h (156 mph)	"	3720m (12,120')	3720m (12,120')	12,120'
- 200 km/h (125 mph)	"	2200m (7185')	2200m (7185')	7,185'
Actual and Limit Speed	"	3020m (9840') (225 km/h)	6040m (19,700')	
Controlling Gradient				
- Design		1.8% (1.5% in Tunnel)	1.8% (1.5% in Tunnel) 0.7%	1.8% (1.5% in Tunnel)
- Actual	0.85%			
Superelevation				
250 km/h (156 mph)	15 cm (5.9")	10.9 cm (4.3")	10.9 cm (4.3")	10.9 cm (4.3")
300 km/h (186 mph)		10.4 cm (4.1")	10.4 cm (4.1")	10.4 cm (4.1")
Track Center Separation				
250 km/h (156 mph)	4.0 m (13'1")	4.6 m (15'1")	4.6 m (15'1")	4.6 m (15'1")
300 km/h (186 mph)		5.0 m (16'5")	5.0 m (16'5")	5.0 m (16'5")
Rail	UIC 60 (121 lb/yd) CWR	UIC 60 (121 lb/yd) CWR	UIC 60 (121 lb/yd) CWR	UIC 60 (121 lb/yd) CWR
Ties	Concrete monoblock, .6m (24") centers	Concrete monoblock, .6m (24") centers	Concrete monoblock, .6m (24") centers	Concrete monoblock, .6m (24") centers
Fasteners	Elastic, 'K' or Pandrol Clips	Elastic, 'K' or Pandrol Clips	Elastic, 'K' or Pandrol Clips	Elastic, 'K' or Pandrol Clips
Ballast Depth	35 cm (14")	35 cm (14")	35 cm (14")	35cm (14")
Switches/Turnouts	250 km/h (156 mph) tangent, 128 km/h (80 mph) turnout	300 km/h (186 mph) tangent, 160 km/h (100 mph) turnout	300 km/h (186 mph) tangent, 160 km/h (100 mph) turnout	300 km/h (186 mph) tangent, 160 km/h (100 mph) turnout
Electrification	3kV d.c.	3kV d.c.	3kV d.c.	3kV d.c.
Catenary	Simple, double contact wire	Simple	Simple	Simple
Tunnel Cross-Section	Not Available	77m^2 (818 ft^2)	77m^2 (818 ft^2)	77m^2 (818 ft^2)
Signalling	Coded track circuit, five-aspect cab signals	Coded track circuit, five-aspect cab signals, continuous data transmission	Coded track circuit, five-aspect cab signals, continuous data transmission	Coded track circuit, five-aspect cab signals, continuous data transmission
Train Control	CTC	CTC	CTC	CTC
Other features	Stations are off high-speed line	Stations are off high-speed line	Stations are off high-speed line	Stations are off high-speed line

TABLE C.5: CHARACTERISTICS OF FIAT ETR-450 AND AVRIL HIGH-SPEED TRAINS

	ETR-450	AVRIL
Vehicle type and top speed	Two-car EMU with active body tilting; 6-car, 8-car trains with all units powered, or 5-car or 11-car trains with unpowered food service car; 250 km/h (156 mph) top speed	Four-car, permanently coupled non-articulated EMU, operated in paired (8-car) consist; 320 km/h (200 mph) top speed
Tilt mechanism	Active electronically-controlled hydraulic tilting; onset triggered by accelerometer/gyrocopic sensing of transition spiral	Active electronically-controlled hydraulic tilting; onset triggered by accelerometer/gyrocopic sensing of transition spiral
Maximum Tilt	10 degrees	10 degrees
Trainset Weight	232 tonnes (255 tons) (5 car), 285 tonnes (314 tons) (6 car), 372 tonnes (409 tons) (8 car), 511 tonnes (562 tons) (11 car)	309 tonnes (340 tons)
Axle load	12.5 tonnes (13.75 tons)	9.7 tonnes (10.6 tons) for each pair of wheels
Trainset Seating Capacity	178 (5 car), 240 (6), 340 (8). 450 (11) in 2 + 1 open 1st class seating	600; no details
Traction Motor	312 kW dc, body-mounted	200 kW, body-mounted, 4 per car
Power Conditioning	N/A	N/A
Pantograph	Two per trainset with 3kV trainline feeding other powered axles; pantographs are mounted on non-tilting frame attached to truck	1 per 4-car half-set, with 3kV trainline feeding 16 motors per set; pantographs are mounted on non-tilting frame attached to truck
Vehicle Dimensions	Cab cars: 26.9 m (107'7") l x 3.8m (12'6") h x 2.75m (9'4") w; Other Cars: 24.7m (84') l x 3.8m (12'6") h x 2.75m (9'4") w	N/A
Braking	Rheostatic plus two discs per axle	Rheostatic plus disc brakes
Body Structure	Light aluminum alloy transverse frame, stressed-skin construction; interior isolated from structural members; windows double-glazed; power-operated flush plug doors	N/A
Truck Design	Multi-piece unit with 8' wheelbase, coil-spring primary and secondary suspensions, limited radial steering capability.	Very light-weight independent-wheel "spider" truck; each wheel is independently sprung; similar to truck on Fiat experimental coach
Other Features	Complete pressure sealing to prevent overpressure effects in tunnels; chemical retention toilets.	Complete pressure sealing to prevent overpressure effects in tunnels



(Source: Railway Gazette International, June 1989)

Figure C.8: Typical Cross-Sections for Italian State Railways' High-Speed Infrastructure

The time period available for tilt actuation is very short. The minimum length of transition spiral on the *Direttissima* is about 1,100 feet, while transitions on conventional lines, where the tilt feature is especially important, can be as short as 330 feet. At 156 mph, the longer distance is traversed in just under five seconds, while at the reduced speed of 113 mph, the shorter spiral is covered in exactly two seconds. As detailed below, the ETR-450 controls the onset of tilt through an on-board electronic controller which analyzes inputs from accelerometers and gyroscopes in a closed-loop system, and actuates hydraulic cylinders connecting the vehicle bodies and truck frames. The controller also provides active lateral airbag suspension to reduce differential movement between the truck and coach body. The sensors on the ETR-450 are located only in the lead vehicle. Sensor signals are transmitted along a trainline to control units on each car, so that tilt onset/removal occurs sequentially as each vehicle reaches the appropriate location on the track.

FS attempts to limit uncompensated lateral acceleration to 0.08g. The ETR-450 tilting mechanism can provide up to 10 degrees, while the track has a maximum of 15 cm (5.9 in), or six degrees, of superelevation. In combination, these will compensate for about 0.29g, which in theory would allow total lateral acceleration in curves of 0.37g, which would translate into a 30% speed increase over what could be attained with non-tilting equipment. In practice, FS limits the maximum non-compensated acceleration to 0.21*g* at the truck, which with 0.08*g in the cabin* translates into a 20% speed gain. The amount of active tilt has been restricted to eight degrees (excluding the effects of differential suspension compression) largely because of the effect the larger total tilt has on window-seat passengers.

The conceptual design for the Avril is still not known in detail, but the limited information that has been released to date suggests that the most innovative aspect of the Avril will be its independent-wheel trucks, shown in **Figure C.9**.[16] The use of independently-sprung, independently rotating wheels permits an effective balance between high-speed stability and curving capability at a much higher speed than is attainable with conventional wheelsets. Fiat estimates that these trucks, in combination with improved trainset aerodynamics, will reduce train resistance by as much as 40%.

C.2.2.3 <u>Truck and Tilt Actuation Mechanism</u>

Figure C.10 illustrates the truck used on the ETR-450 active-tilt EMU. This truck has also been adopted for the tilting equipment being built for Germany and Austria. Each truck is built around a two-piece articulated steel frame, with rubber pads isolating the two sections of the frame. This permits use of a stiffer primary suspension to reduce pitching motion and improve high-speed performance.

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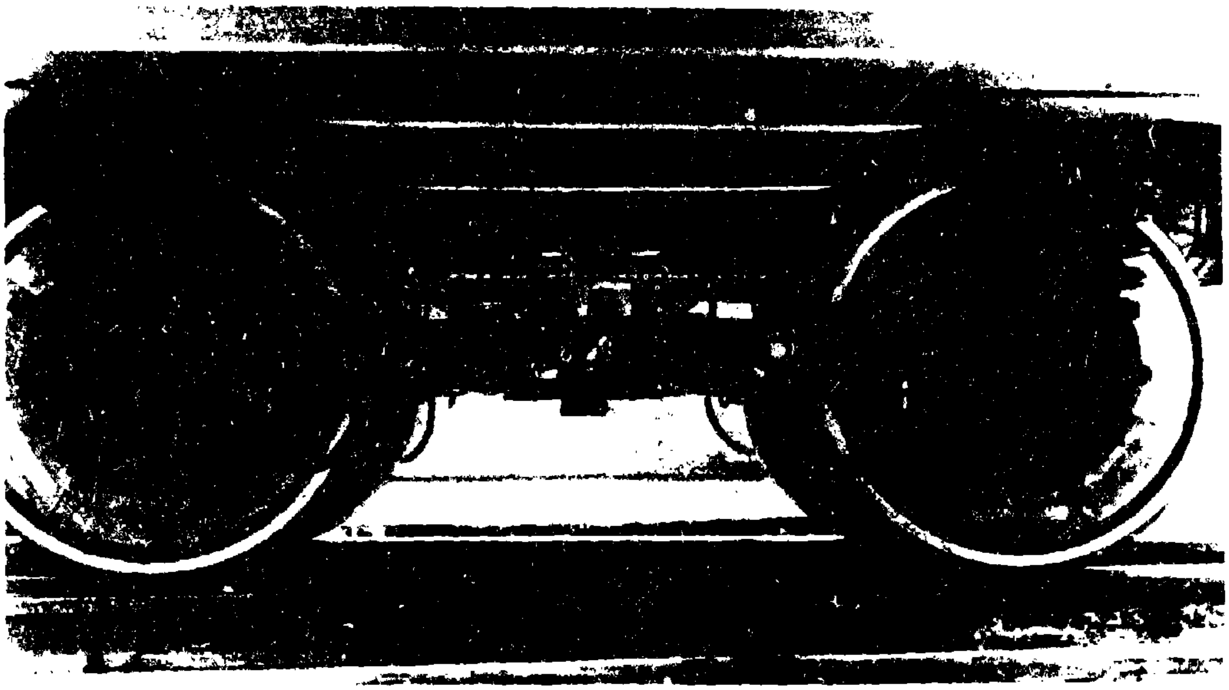
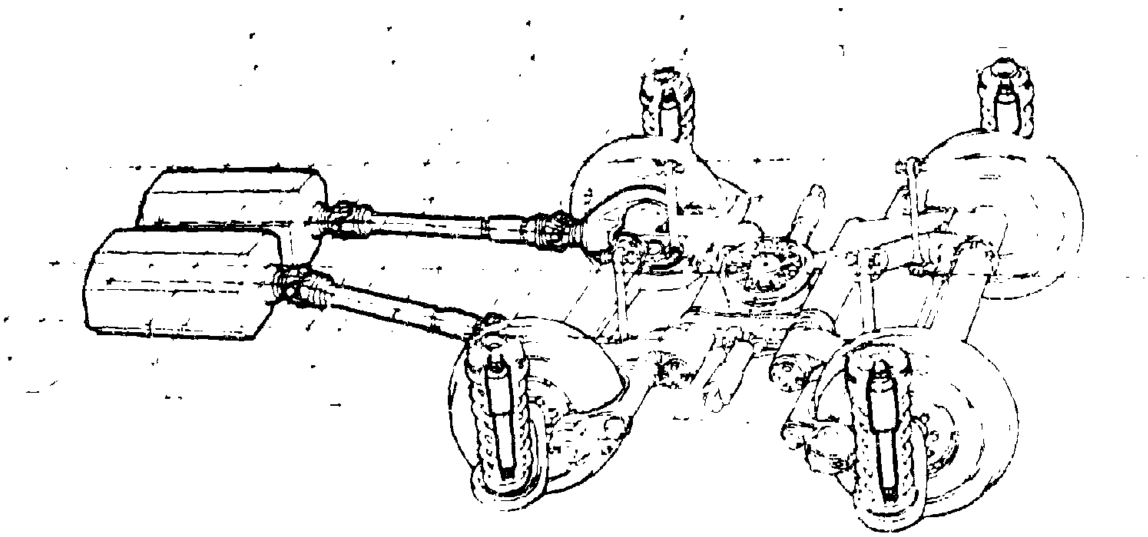
[16] "Fiat Plans Third-Generation Pendolino," Railway Gazette International, December 1990.





(Source: Railway Gazette International, December 1990)

Figure C.9: Independent-Wheel Truck on Experimental Fiat Coach (Top) and Conceptual Truck Design for Fiat Avril Advanced Tilt-Body EMU

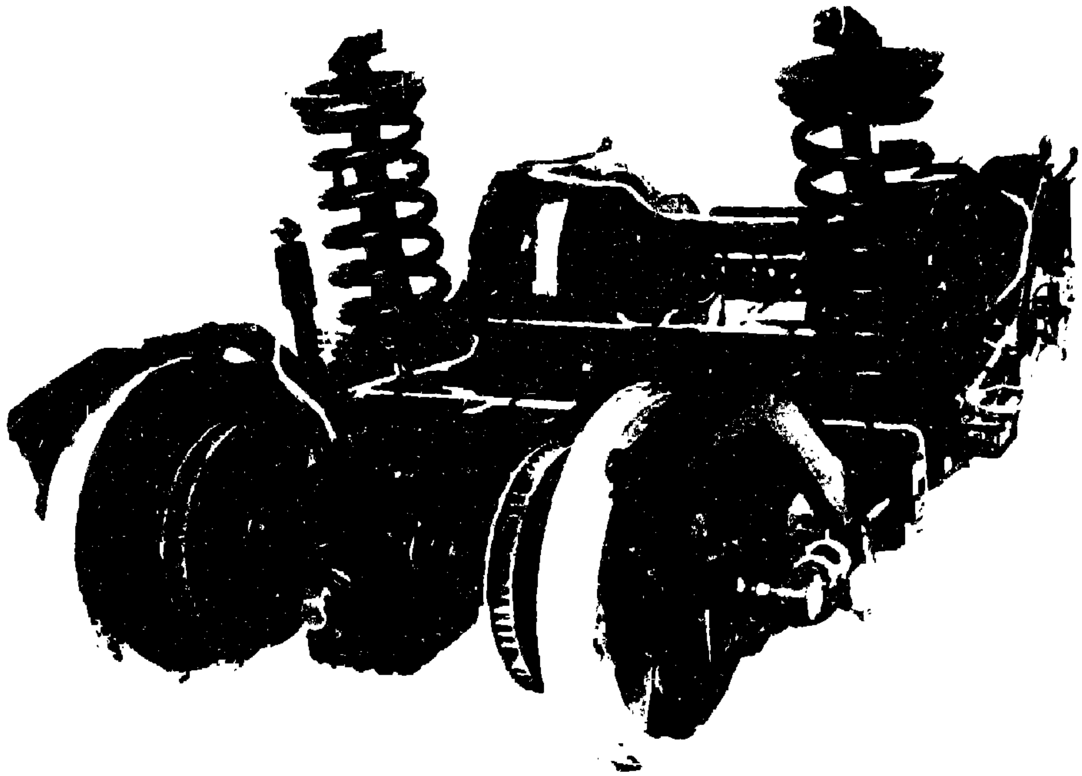


Figure C.10: The ETR-450 Truck

The articulated truck frame also allows the wheelsets to accommodate track twist while maintaining a uniform wheel/rail loading. This helps to control lateral force peaks due to track geometry defects in curves, thereby partially offsetting the higher lateral forces generated by the increase in speed permitted by active body tilting.

Self-aligning roller-bearing axle boxes are connected to the frame by the coil-spring primary suspension and tapered elastic links. The large coil springs of the secondary suspension are supported by the frame and linked to the carbody through a transverse bolster beam, highlighted in **Figure C.11**.

This configuration provides lateral and transverse suspension, as well as limited rotational movement of the truck relative to the bolster beam, as is required when the truck enters a curve.

The transverse bolster beam is connected to the carbody by tilt linkage swing arms and by the hydraulic tilt actuators, as shown in **Figure C.12**. The non-tilting pantographs fitted to each two-car set also are mounted on the bolster beam, as shown in **Figures C.16 and C.17** below.

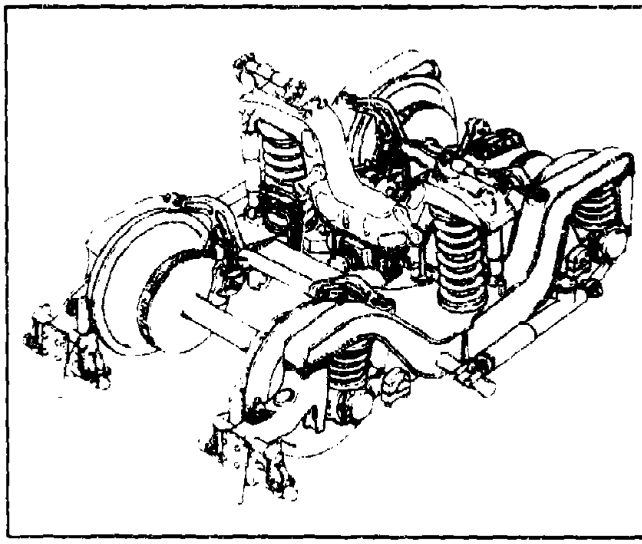
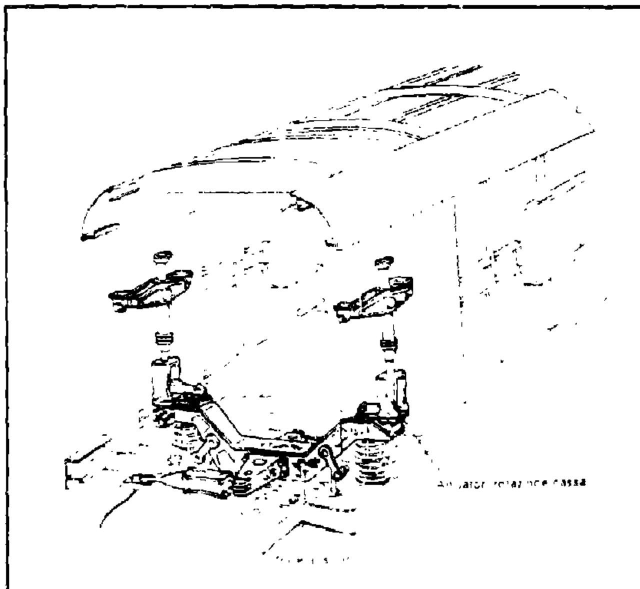


Figure C.11: ETR 450 Truck with Bolster Beam

The dynamic performance of the truck suspension (primary and secondary coil springs and conventional dampers and stabilizing links) has been optimized to achieve an effective 3-way compromise between hunting stability at high-speed, curving behavior and lateral loading, and vehicle ride quality.

There are two trucks per vehicle in the ETR-450, with the inner axle of each truck being driven by a body-mounted traction motor through a universal-jointed drive shaft and a bevel-gear final drive unit, as shown in **Figure C.13**. Each axle carries two cast-iron disc brakes.



(Source: LA Tecnica Professionale N.5 - Maggio 1989

Figure C.12: Connections Between Carbody and Bolster Beam

The wheels used on the ETR-450 have a double-dished cross-section that permits reduction in wheel weight (and thus, total weight and especially, unsprung weight). Each wheel is fitted with non-integral wear-resistant hardened tires.

The low total weight of the truck and especially of the unsprung components permits tuning of the suspension characteristics to minimize the dynamic load increment, a very important consideration for high-speed operation.

Each ETR-450 truck also carries a pneumatic lateral displacement actuator that is linked to the tilt system controller, as shown in **Figure C.14**. When the carbody is tilted to compensate for

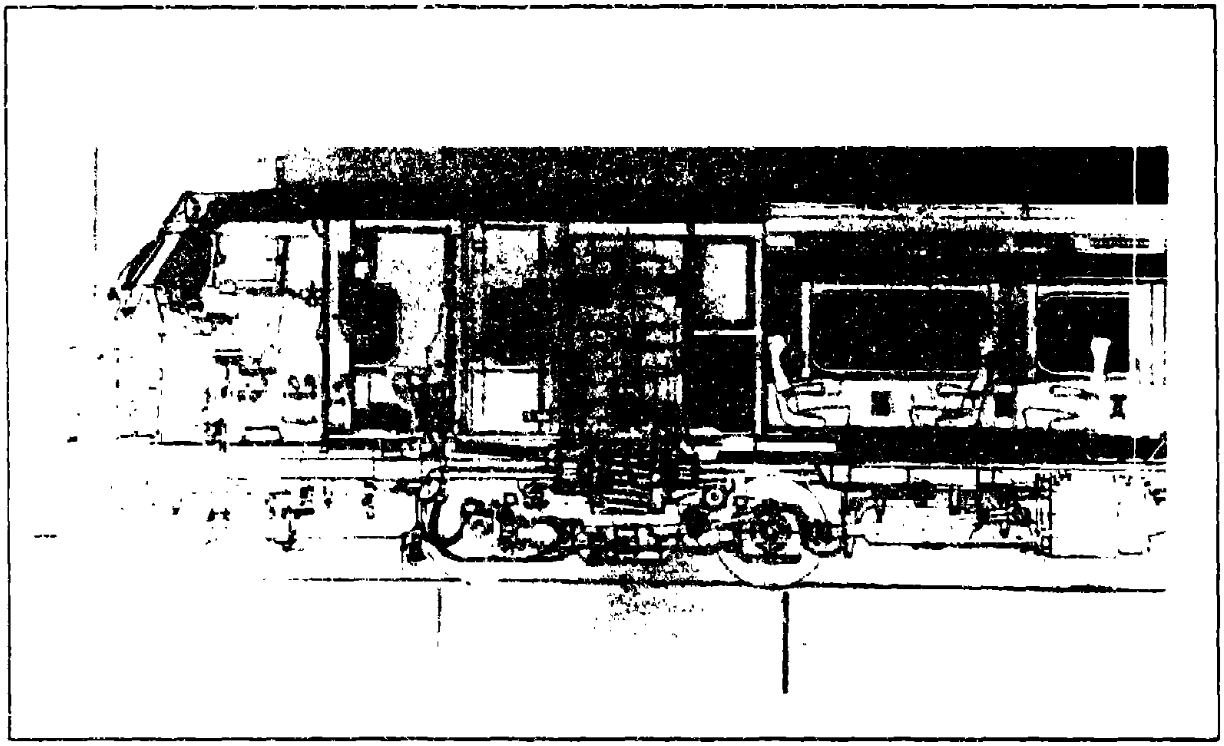


Figure C.13: Longitudinal Section of Cab Showing Body-Hung Traction Motor and Driven Inner Axle on Truck

lateral acceleration during curve negotiation, this component shifts the carbody laterally so that its center of gravity is displaced toward the center of curvature, without requiring any stiffening of the two-stage lateral suspension. This has the effect of improving the margin of safety with respect to vehicle rollover stability.

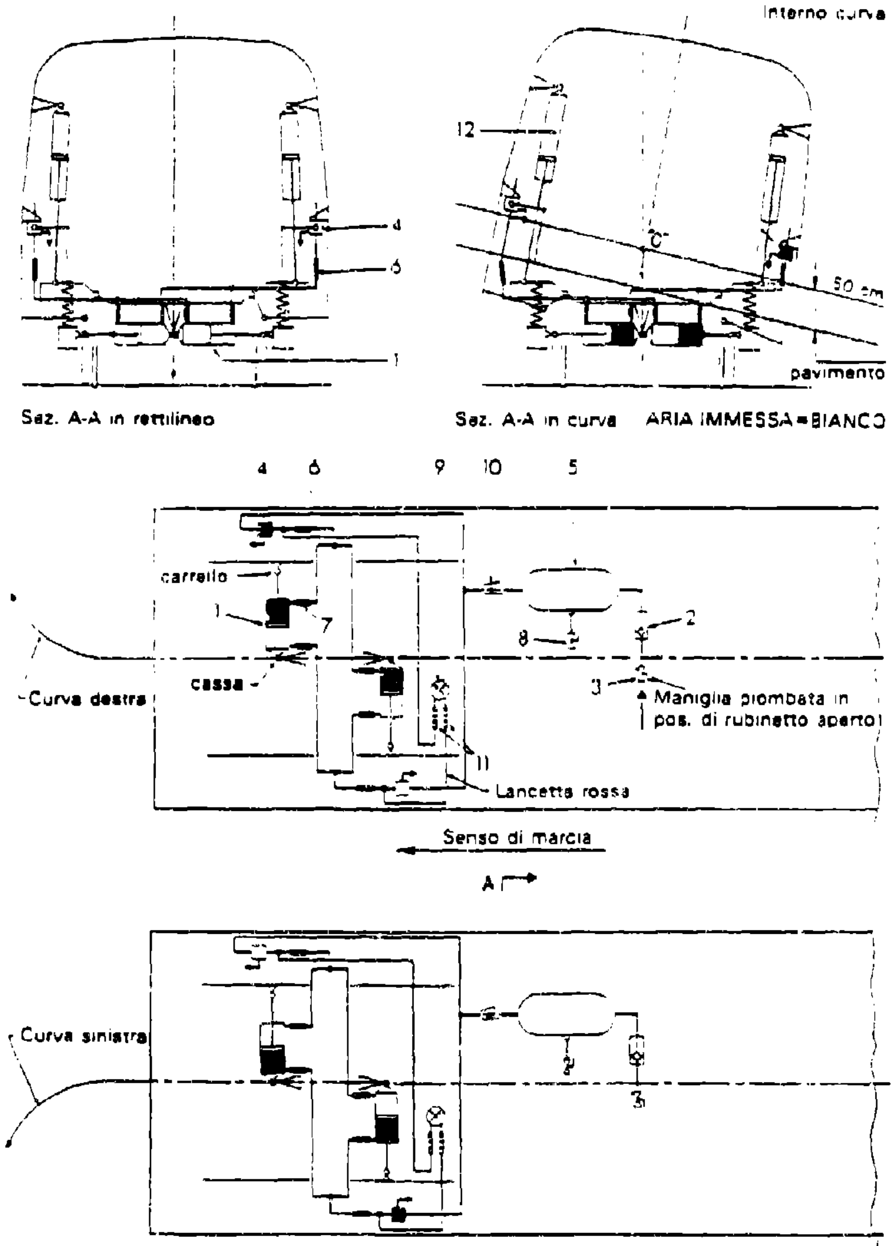
The trucks of each cab car also carry the sensor suite for control of the active tilt mechanism. The frame of the head-end truck of the cab car is equipped with two gyroscopes and two accelerometers, plus a speed sensor, while the rear truck mounts two accelerometers and a speed sensor.

C.2.2.4 Tilt Control and Actuation Subsystem

On the ETR-450, control of the active-tilt mechanism is achieved by means of the sensor and signal processing array shown schematically in **Figure C.15**.

The sensor suite is located in each cab car of the push-pull consist, and includes:

- o Two gyroscopes (one for backup in the event of a failure) mounted on the forward truck frame,



(Source: LA Technica Professionale N.5 - Maggio 1989

Figure C.14: Schematic for Active Lateral Suspension

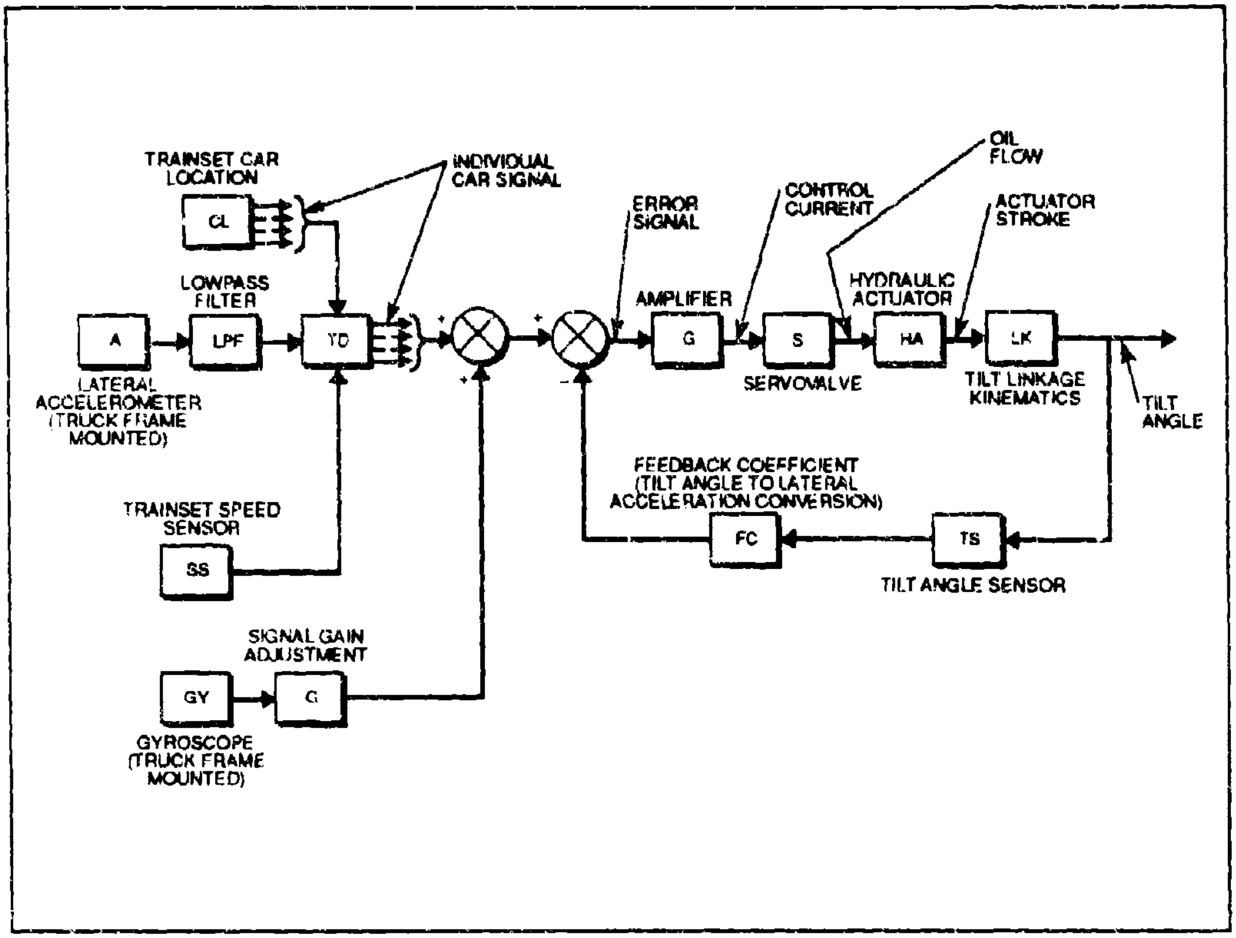


Figure C.15: Tilt Control Schematic for ETR-450

- o Four lateral accelerometers, two on each truck frame (again one on each as a backup) with the tilting control acceleration signal being the average between the two trucks,

- o A speed sensor on the inside axle of each cab car truck, and

- o One carbody-mounted accelerometer on each cab car.

The lateral acceleration signal generated by the accelerometers is lowpass filtered to remove noise induced by track geometry defects and consequent vehicle dynamic response, and then is corrected for time lag using the inherently low-noise gyroscope signal. To eliminate any effects of track warp (cross-level irregularities), the first 10 mm (0.4 in) of track superelevation detected by the gyroscope is ignored. Train speed is determined by the truck-mounted sensors. The on-board digital microprocessor in the lead car analyzes the speed and gyro-corrected lateral acceleration signals to determine the required timing of tilt onset and the magnitude of tilt required to reduce uncompensated lateral acceleration below the comfort threshold. The processor then generates the required command signal for each car and transmits the commands to the following cars through a high-reliability data link.

The command signal from the lead car causes the tilt actuation mechanism of each individual car to initiate tilting in sequence, so that tilt onset coincides with the predicted moment of entry into the transition curve.

The command sequencing, magnitude, and rate depends on the trainset speed and the geometry of the curve being traversed. Except for the lead vehicle, this control strategy provides the advantages of car-ahead curve sensing.

Figures C.16 and C.17 each show a cross-section of an ETR-450 vehicle in untilted and tilted states, respectively.

The command signal triggers operation of the electro-hydraulic servovalve on each truck. Opening this valve permits hydraulic fluid to flow into the hydraulic cylinders of the tilt actuators, as shown in **Figures C.16 and C.17**. The volume and flow rate through the valve is controlled by the feedback loop based on tilt angle displacement, as shown in **Figure C.15**. The hydraulic actuators are made up of two nearly vertical cylinders per truck connected to the truck bolster beam and to carbody anchor points on the car frame walls near the roof, as shown in **Figures C.16 and C.17**. The power supply for the tilt mechanism consists of a hydraulic power pack (motor, pump, reservoir, accumulator, pressure regulator, and filter) mounted on each car.

The ETR-450 tilt mechanism was designed to provide up to 10 degrees of tilt, excluding the (negative) effect of differential suspension compression, which in practice restricts the effective tilt angle to about eight degrees. The maximum tilting rate is limited to six degrees per second, which is close to the upper bound for passenger ride comfort. The ETR-450 is designed to accommodate a maximum level of non-compensated lateral acceleration (at the truck) of 2.1 meters (6 ft 10 in)/sec^2, or about 0.21g; at this limit, the uncompensated lateral acceleration in the passenger compartment can be reduced to an acceptable 0.08g by the active tilt mechanism.

The configuration of the tilting mechanism shown in **Figures C.16 and C.17** produces a roll center close to seat level (about 25 cm [10 in] above floor level) which itself improves the perceived level of passenger comfort by minimizing passenger exposure to lateral acceleration due to the rate of tilt onset. The location of the roll center 1.49 m (4 ft 10 in) above the rails, and about 0.28 m (11 in) above the carbody center of gravity allows the use of a passive gravity return to the untilted position. In the event of complete loss of hydraulic and electrical power, the tilt mechanism will still be "fail safe."

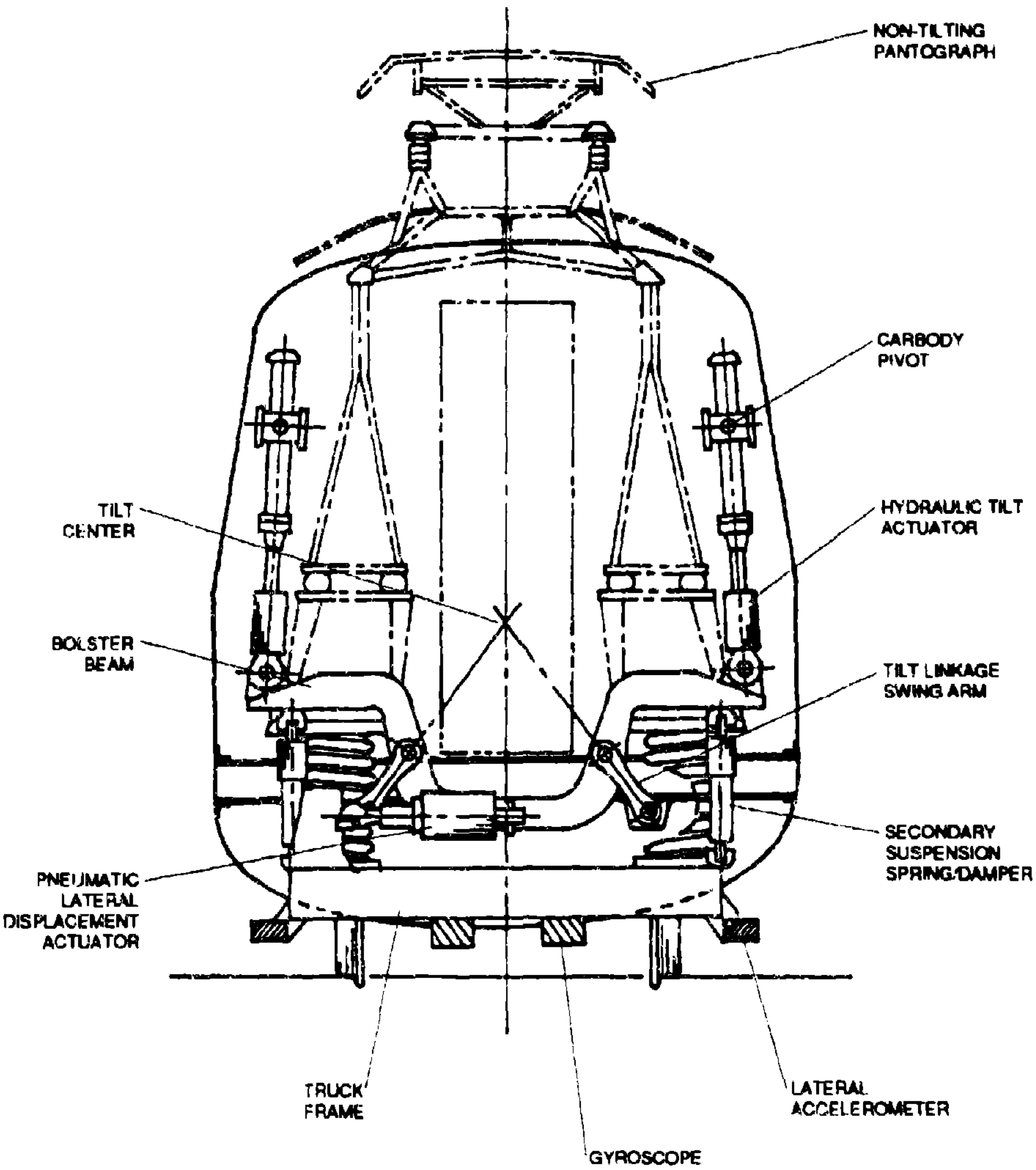


Figure C.16: Cross-Section of Untilted ETR-450 Vehicle

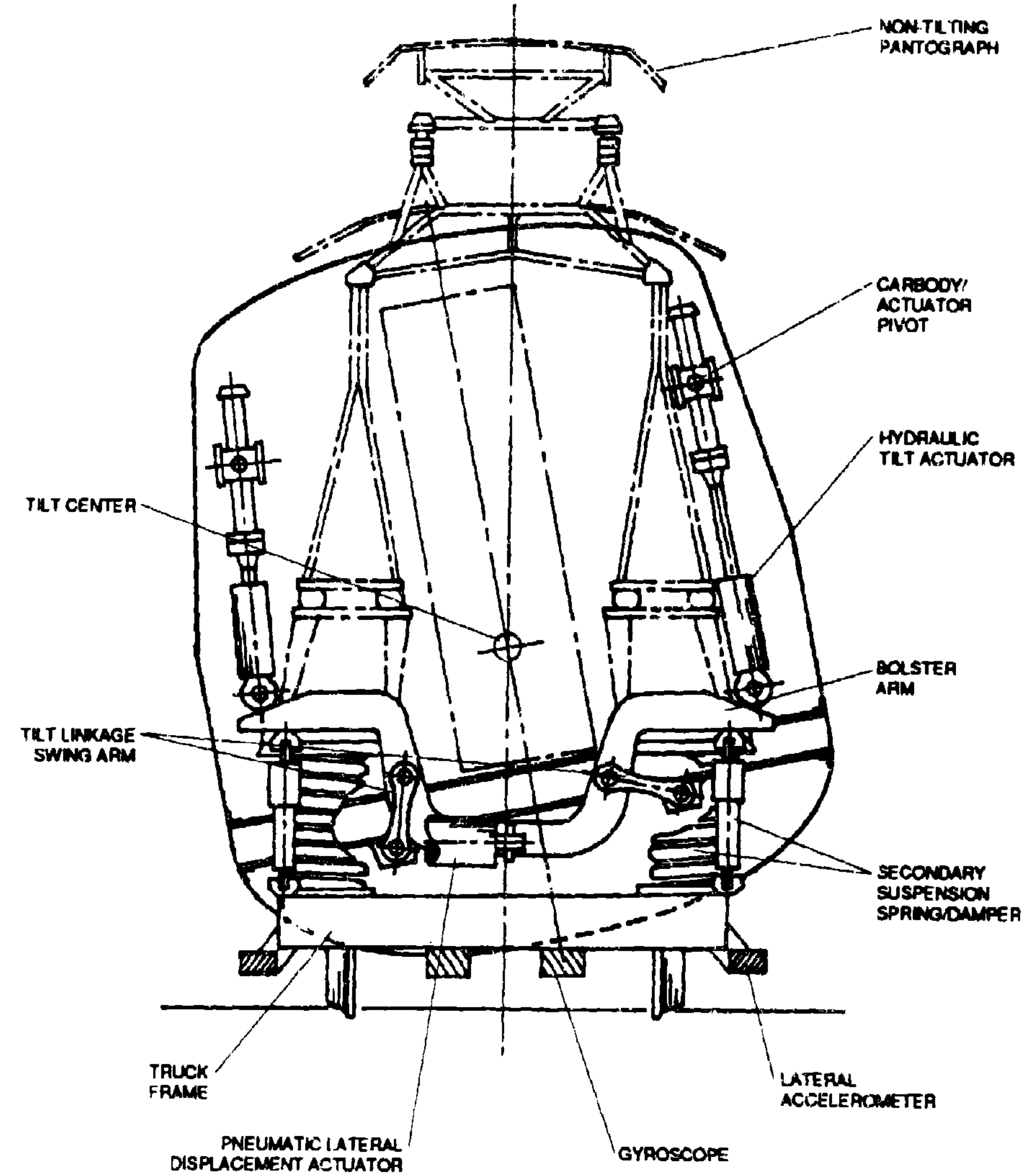


Figure C.17: Cross-Section of ETR-450 Vehicle in Tilted Position

C.2.2.5 Propulsion and Braking

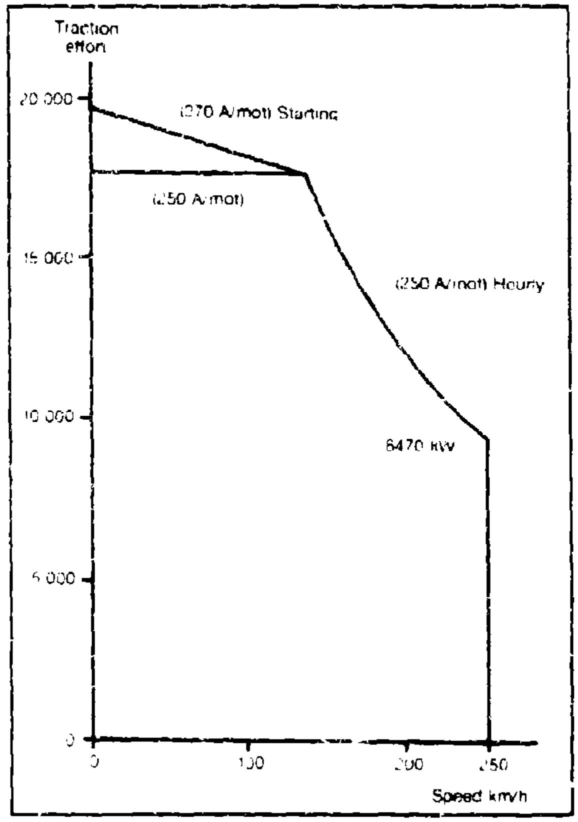


Figure C.18: Speed/Tractive Effort Curve

The ETR-450 is powered by two 312kW dc carbody-mounted traction motors per vehicle (1250 kw per two-car traction unit) driving the inboard axle of each truck through a cardan shaft and a right-angle drive bevel-gear final drive unit, as shown in **Figure C.13**. The series-excited four-pole motors are self-ventilated, operating at a maximum speed of 2,860 rpm at 250 km/h (156 mph). Each motor has a continuous power rating of 294 kW, and a 1-hour rating of 344 kW. Tractive effort at start-up is 192 kN; **Figure C.18** shows the tractive effort-speed curve for an 8-car ETR-450 trainset.

Power conditioning and excitation is provided by electronic control of two single-phase choppers per vehicle. The choppers operate at four frequency levels (65 Hz, 130 Hz, 260 Hz, and 390 Hz); the two choppers on a given vehicle are operated 180° out of phase to minimize 50Hz harmonics. The power converter semiconductors are located in drawers under the vehicle, along with the solid-state components of the traction-motor shunt system. The smoothing inductors are force-ventilated open-core units.

On FS, power is drawn from overhead catenary energized at 3kV dc, through a single-stage pantograph mounted on a framework connected to the bolster beam rather than to the carbody, as shown in **Figures C.16 and C.17**, so that the pantograph does not tilt. Each traction unit is equipped with one pantograph, but the trainsets are equipped with a 3kV trainline to permit single-pantograph operation, thereby avoiding problems arising from catenary dynamics which would affect trailing units.

Braking is achieved through a combination of dynamic and friction braking. The motors are configured to provide dynamic braking at speeds in excess of 80 km/h (50 mph), either alone or in concert with air-actuated disc brakes. Below 80 km/h (50 mph), braking is by means of the disc brakes only.

In the dynamic braking mode, the motor armature current is regulated through the shunt-chopper power conditioning circuits, with the electrical energy being dissipated through roof-mounted braking rheostats.

Disc braking is by means of air-actuated composite pads acting on two cast-iron brake discs carried on each axle (including powered axles). Each disc is equipped with an independent brake cylinder.

The ETR-450 traction and braking control system incorporates anti-skid and anti-slip features to ensure maximum use of available adhesion and to protect both trainsets and track against damage caused by wheelslip, skidding and consequent operation with out-of-round wheels.

The braking modes fitted to the ETR-450 are capable of stopping an eight-car trainset within 3,400 m (just over 11,000 ft) from 250 km/h (156 mph) on a 0.8% downgrade for service braking. Under emergency braking conditions, the disc brakes alone can stop the train within 3,200 m, again on a 0.8% downgrade.

C.2.2.6 Other Features

The body structure of the ETR-450 is formed principally from light aluminum alloy extrusions and fabricated components, and is designed and built to comply with the strength requirememts specified in UIC 651.[17] To control interior noise and vibration, and to facilitate heating and cooling, the body structure and exterior shell are isolated from the interior panels by composite pads. The interior panels are lined with thermal insulation and acoustic baffles. The windows are double-glazed, and the car bodies are pressure-sealed to avoid overpressure pulses in tunnels at high-speed. External doors are plug-type to facilitate pressure sealing and reduce aerodynamic drag.

C.2.2.7 Other Technologies Based on ETR-450 Truck and Tilt Mechanism[c]

In addition to the ETR-450 itself, there are two classes of active-tilt rail vehicles presently under construction which are based on the truck and tilting mechanism developed by Fiat. MAN-GGH, a subsidiary of AEG-Westinghouse, is about to deliver the first of 20 two-car VT-610 diesel-electric multiple unit (DMU) sets to German Federal Railways (DB) for acceptance testing and ultimate service on the Nurnberg-Bayreuth/Hof line, while three pre-production prototypes of the Class 4012 six-car EMU trains are being built for Austrian Federal Railways (OBB), with delivery scheduled for 1994.

In both instances, the selection of an active-tilt technology represents an attempt to improve the average speed and marketability of rail services without the expense of constructing new lines or rectifying the alignment of existing track, neither of which was practical for the very sinuous routes and low to moderate traffic densities that characterize services in northern Bavaria and much of Austria.

[17] "First ETR450 To Be Ready This Year," Railway Gazette International, January 1987.

DB interest in body tilting dates back to 1965, and in fact DB built and operated the VT 624 active-tilt DMU on the Koblenz-Dillengen line between 1972 and 1974. However, there were on-going technical problems with the accelerometer-based control system, and the use of the tilt feature was terminated. By 1975, DB had ceased development work on tilt-body equipment.

The development and successful application of passive- and active-tilting technologies since 1975, and especially activities in Italy, coupled with a clear shift of DB management towards a more businesslike approach to the provision of rail passenger services, prompted DB to re-examine its position with respect to tilt-body technologies. This reappraisal began in 1987 with a survey of German railway lines to identify those with geometric characteristics consistent with tilt-body capabilities, followed by a feasibility analysis to establish the technical and financial implications of tilt technologies on the candidate routes.

To verify the technical and performance assumptions used in the feasibility study, DB undertook a series of running trials during 1987-88. These trails took place on its Koblenz-Dillingen and Eichstatt-Treuchtlingen lines, using the prototype ETR-401 trainset hauled by a DB Class 120 electric locomotive (DB electrification is at 15kV 16 2/3Hz AC, rather than the 3kV DC used by FS). DB also undertook tests of a Talgo Pendular passive-tilt trainset. The test program investigated the behavior of both the locomotive and the ETR-401 at speeds up to 20% above the pre-existing limits. The results of these trials were quite favorable, insofar as curving forces remained within allowable limits under all but the least favorable circumstances. The active-tilt technology was preferred to the passive-tilt on the grounds that it could compensate for about 70% of the lateral acceleration measured at the truck, versus about 25% with passive tilting, at the same curving speed.

As a consequence of these successful and the feasibility study results, in November 1988 DB decided to develop the VT-610 diesel-electric tilting trainsets, using the Fiat active tilting mechanism and truck developed for the ETR-450. Orders for 20 two-car units were placed in early 1990, with the first delivery in December 1991. DB anticipates substantial improvements in trip times with introduction of the VT-610 fleet, with Nurnberg-Hof dropping from 117 minutes to 86 minutes, and Nurnberg-Bayreuth from 6[illegible] minutes to 56 minutes. The improvements result from a significant increase in line speeds. **Figure C.19** shows the changes for the Nurnberg-Hof line.

Austrian Federal Railways also followed the tilt-body developments of the 1970s and early 1980s with great interest, culminating in an extensive series of running trials using the ETR-401 prototype during 1988 and 1989. These trials were carried out on OBB's Innsbruck-Salzburg, Graz-Villach and St. Veit-Graz lines. On the basis of the results of these trials and an internal feasibility study, OBB issued a letter of intent for three pre-production prototype six-car electric multiple-unit trainsets based on the Fiat ETR-450 truck and active-tilt mechanism. This was converted to a firm order in 1990, and the first set is expected to be delivered in 1994.

Each EMU set will have 68 first-class and 198 second-class seats plus catering facilities. Each three-car half-set will form a traction unit. The design speed is 200 km/h (125 mph), and all cars will be pressure-sealed. The vehicles will be of welded aluminum construction. Maximum axle load will be 13 tonnes (14.3 tons) and total train weight will be 312 tonnes (343 tons). Power will be collected by a single pantograph from overhead catenary at 15kV 162/3 Hz, with traction distributed over three cars, with train-line power distribution. Power conditioning is by inverter, with GTO thyristors supplying four body-hung three-phase ac induction motors per half-set, each rated at 400 kW continuous and 450 kW peak. The drive train is identical to that on the ETR-450. Braking is computer controlled, involving a blend of regenerative dynamic, disc (two per axle) and electromagnetic rail brakes.

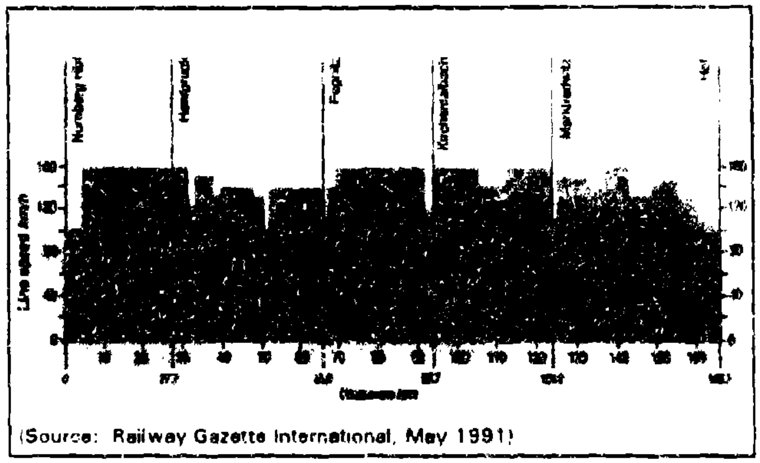


Figure C.19: Comparison of Speed for Conventional and Tilting Equipment on Nurnberg-Hof Line

C.2.2.7.1 The VT-610 Tilt-Body DMU

Figure C.20 illustrates the two-car VT-610 DMU ordered by DB. Note that one car carries two diesel engine-generator sets and the other car only one. Each diesel-generator set powers one body-hung dc traction motor, each of which in turn drives the inside axle on one of three trucks through a cardan shaft and axle-mounted right-angle drive. **Table C.6** summarizes the major attributes of the VT-610.

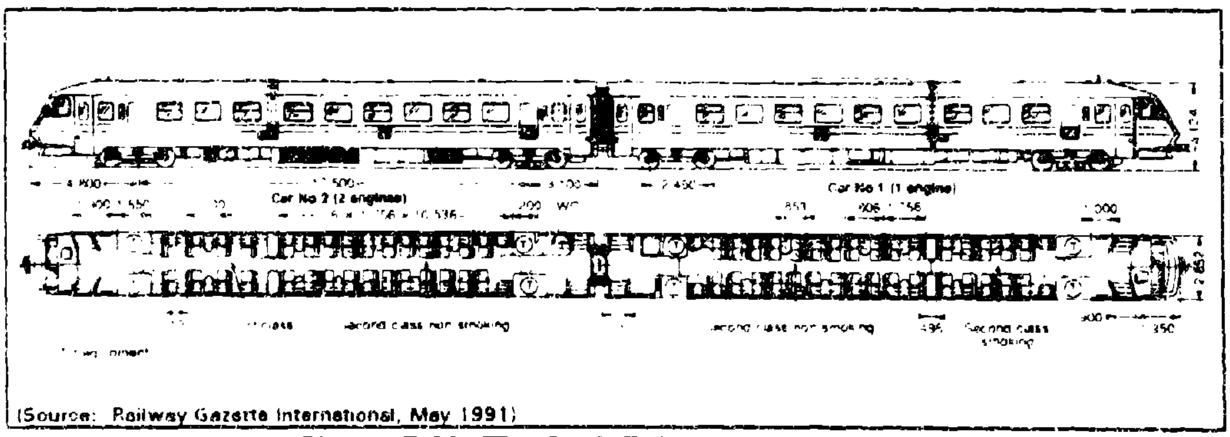


Figure C.20: The DB VT-610 Active-Tilt DMU

TABLE C.6
ATTRIBUTES OF VT 610 ACTIVE-TILT DMU

Vehicle Dimensions	Overall length (two-car set)	51.7 m (168'6.5")
	Length of single car	25.4 m (82'9.6")
	Width	2.85 m (9'4")
	Height above rail	4.12 m (13'5")
	Distance between truck centers	17.5 m (57'1")
	Truck Wheelbase	2.45 m (8')
Vehicle Structure	Self-supporting tubular structure welded from aluminium extruded profiled sections with body sides reinforced with fibre glass; structure conforms to UIC 505-1 profile	
Weight (2 car set)	103.2 tonnes (113.5 tons)	
Maximum Axle Load	13.4 tonnes (14.75 tons)	
Maximum Speed	160 km/h (100 mph)	
Seating Capacity	16 first class, 114 second class, 6 folding	
Propulsion	Three 12 cylinder MTU diesels, 485 kW each, direct-drive electric generators, each powering body-mounted dc traction motor driving inside axle of adjacent truck through a cardan shaft and axle-mounted gearbox	
Truck Design	Same as for ETR-450; new wheel diameter 0.89 m (2'11")	
Braking	KE disc airbrakes, one disc/driven axle, two/ trailer axles; electrodynamic braking and emergency EM rail brake	
Tilt Control/Actuation	As for ETR-450, but without sequential tilt command feature, and with reduced maximum tilt rate (3°/sec versus 6°/sec for ETR-450); spacing of hydraulic actuators modified to accomodate wider DB carbody.	
Maximum Tilt Angle	8 degrees	
Tilting Center Height	1.64 m (5'4")	
Minimum Curve	125 m (407') radius	
Revenue Consist	Up to four two car traction units, depending on traffic demand.	

C.2.3 THE ABB X2000[d]

C.2.3.1 Overview

The genesis of the X2000 Sprintor technology began almost two decades ago, with the realization on the part of senior management at Swedish State Railways (SJ) that on-going passenger traffic losses to competition from air and automobiles would be inevitable with improved railway performance, but that even under the most favorable conditions, rail traffic volumes could not begin to justify the massive investments in dedicated high-speed infrastructure. From this realization came an initial specification for a light-weight, low-axle-load (15 tonne/16.5 ton) electric multiple-unit trainset that would be completely compatible with the existing track and electrification of SJ but that would also incorporate an active tilting mechanism and radial steering trucks for both powered and unpowered vehicles, to increase the achievable speed in curves by 30%, and would be capable of at least 125 mph operation.

These requirements were dictated in part by market conditions, and in part by the large number of relatively tight-radius curves (as small as 291m [950 ft]) and overall track standards of the SJ lines. These tracks are built with continuous-welded UIC-50 (101 lb/yd) rail on concrete monoblock ties with elastic clip fasteners. The ties are a mix of older 190 kg (420 lb) units and newer 255 kg (560 lb) units, at 66 cm (26 in) centers. In curves of less than 500m (1650 ft) radius, the tie spacing is reduced to 61 cm (24 in). SJ electrification follows German practice with overhead catenary supplying single-phase 15kV 16 2/3 Hz power.

This original concept specification prompted ASEA (later to merge with Brown Boveri to become ABB Traction) to undertake an intensive and costly R&D program using its purpose-built X15 test train. The ABB investigations examined a range of truck and suspension designs, various types of passive and active body tilting and tilt-control mechanisms, and improved traction, braking, and control subsystems.

Despite the success of this extensive research and development program, when SJ issued a formal request for tenders in 1983, the combination of requirements, especially the very low axle load, active body tilting and steerable trucks on EMU vehicles, proved to be too demanding even for ASEA: no responsive bids were received.

However, the fundamental requirement for the technology was unchanged, and the urgency of the problem of traffic loss was increasing. Domestic air travel was growing at 10% to 15% annually, and Per-Arne Dahlin, SJ's project director for the high-speed train, stated that

> "*...introduction of the high-speed train is a question of survival...If we* (SJ) *don't act now, we will probably be out of the* (passenger) *business....*"

SJ was forced to re-think its specification. It dropped the requirement that the equipment be an EMU, accepting instead a non-tilting power car that would not carry passengers, plus four coaches and an unpowered driving trailer, and relaxed the allowable axle load requirement to 16 tonnes (17.6 tons). The revised specification called for 51 first-class seats and 241 second-class seats, plus a buffet section with 11 table seats. This has since been modified to increase the number of first-class seats to 102 and reduce the number of second-class seats to 152. Wheelchair access and tie-down space has also been added.

The revised specification was issued in 1984, and the contract for development of the technology and supply of 20 trainsets, with an option for an additional 32 trainsets, was awarded to ABB Traction in August 1986. Their experience with the X15 allowed ABB to move directly to creation of a pre-production trainset. Initial test runs were begun in August 1989, and the first production trainset was delivered to SJ in July 1990. This unit entered revenue service on September 4, running between Stockholm and Goteborg, a distance of about 430 km (270 miles). The second production trainset was delivered in December 1990, and with additional units following at the rate of one every two months. Deliveries will continue through 1994 for the firm order.

As additional units become available, SJ is adding Sprintor service between Stockholm and Malmo (624 km/390 miles) and Stockholm and Sundsvall (416 km/260 miles), as well as between Goteborg and Malmo (328 km/205 miles). These services will form a triangle linking the three largest cities in Sweden, with a catchment that includes about 6.8 million people, or almost 80% of the population of the country. There are also prospects for run-through links to Copenhagen, Denmark and possibly connections from there to the rest of the growing European high-speed network.

Although capable of 250 km/h (156 mph),[18] the X2000 was initially limited to 160 km/h (100 mph), pending completion of a $35 million program of infrastructure upgrading. The latter work included modification to the ATC signalling and control system with addition of new wayside indicators to ensure that the X2000 has adequate distance in which to brake from the higher-speed, and an interlocking between the CTC and vehicle detection coils that are being added to at-grade crossings with automatic gate protection. The signals which govern rail access to the crossing will not show green unless the detection coils verify that the road is not occupied. The automatic crossings are being fitted with full-width barriers, and the actuation circuitry modified to provide constant warning time at 200 km/h (125 mph) train speed. The number of at-grade crossings is being reduced from 300 to 100 on the Stockholm-Goteborg line. SJ has not announced plans to increase operating speed above 200 km/h (125 mph), although an X2000 trainset reached 250 km/h on a DB high-speed line during running trials in the summer of 1991. Other changes include adjustment of the transition geometry in some curves to reduce jerk rates and take full advantage to the tilt-body features of the X2000, and a comprehensive program of track maintenance to ensure full compliance with nominal geometric and defect standards for 125 mph operation.

Table C.7 summarizes existing trip times for the X2000 routes, the best time achievable with the X2000 under the 160 km/h (100 mph) restriction, and the time for 200 km/h (125 mph) operation.

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[18] Railway Gazette International, October 1991.

TABLE C.7
TRIP TIME IMPROVEMENTS WITH THE X2000

ROUTE	CURRENT TRIP TIME	X2000 AT 160 KM/H (100 MPH)	X2000 AT 200 KM/H (125 MPH)
Stockholm - Goteborg	3h49 (Best); 4h05 (Avg)	3h 35	2h 59
Stockholm - Malmo	5h 40	N/A	4h 16
Stockholm - Sundsvall	4h 30	N/A	3h 29
Goteborg - Malmo	3h 40	N/A	3h 29

C.2.3.2 The X2000 Equipment Specification

The X2000 trainset as delivered to SJ, shown in **Figure C.21**, is made up of a 3,260 kW electric locomotive, four tilt-bodies coaches, and an unpowered tilting driving trailer that is fitted with 47 second-class seats. **Figure C.22** illustrates the X2000 layout. Note that because this is basically an electric locomotive with coaches, the consist configuration can be readily altered to accommodate different levels of demand or different market requirements. Additional coaches could be added, and the driving trailer replaced with a second locomotive. It is also possible to operate two trainsets in multiple, although the dynamic behavior of the SJ catenary would limit speed to 180 km/h (113 mph).

Table C.8 summarizes the physical characteristics of the X2000. The most interesting features of this technology are radial steering trucks for both the locomotive and unpowered cars, the active tilting mechanism in the coaches and driving trailer, and the use of ac asynchronous traction motors with GTO thyristor control. The body structure is stainless steel, and the passenger cabin walls and floor are isolated from the body structure by rubber bushings, thereby reducing vibrations and damping out shocks arising from higher-speed operation on the existing tracks. The X2000 trainset is 141 m (460 feet) long, the total loaded weight is 343 tonnes (377 tons), and the train can normally stop in 1.76 km (1.1 miles) from 200 km/h (125 mph).

There are two controlling limits in determining allowable speed through curves:

- o The lateral and vertical forces imposed on the rail by the train during curving (and the consequent risk of rail rollover and/or vehicle overturning as well as the levels of wear - and thus costs - on wheel flanges and rails), and

- o The level of lateral acceleration felt by passengers during curving.

(Source: Rail Engineering International - Edition 1991, No. 3)

Figure C.21: The X2000 Active-Tilt Trainset

The former consideration has both safety and cost implications, while the latter has a direct bearing on perceived comfort level and thus, on service marketability and revenue.

Given the large number of short-radius curves on the lines to be served by the X2000, and the requirement that these lines continue to carry freight and conventional passenger traffic, ABB incorporated two very important features - radial steering trucks, which reduce track forces, and an active body-tilting mechanism, which reduces lateral acceleration perceived by passengers.

C.2.3.3 X2000 Trucks

The X2000 incorporates radial steering trucks. These trucks reduce track forces by permitting the (normally rectangular) configuration of the axles and truck frame to alter to a parallelogram shape, with the short side on the inside of the curve and the long side on the outside. This process, illustrated in **Figure C.23**, is caused by the differences in creep force generated at the contact points between the wheel d the rails on the inside and outside of the curves, so that the steering aspect of the truck is a ssive, rather than active, response to entry into a curve.

TABLE C.8
PHYSICAL CHARACTERISTICS OF X2000 TECHNOLOGY

Characteristic	Locomotive	Coaches/Driving Trailer
Length	17.2 m (56')	24.5m (80')/22.1m (72')
Width	3.07 m (10')	3.07m (10')
Height	3.83 m (12'6")	3.83m (12'6")
Weight	64 tonnes (70.4 tons)	54 tonnes (59.4 tons)
Truck Wheelbase	2.91m (9.5')	2.91m (9.5')
Wheel Diameter	1.1m (3 6')	0.88m (2'10")
Track Gauge	Standard (1.435m/4'8 1/2")	Standard (1.435m/4'8 1/2")
Propulsion Type	815 kW 3-phase AC asynchronous motors with GTO thyristor convertor	N/A
Supply Voltage	Overhead Catenary, 15kV, 162/3 Hz Single Phase	N/A
Continuous Power Rating	3260 kW	N/A
Maxium Tractive Effort	160 kN	N/A
Braking	Electric regenerative, plus two disc brakes and brake block units per axle	Two disc brakes/axle, plus magnetic rail brakes for emergency use
Tilt Control and Actuation	Accelerometer on leading/trailing trucks in consist, plus speed sensor	Differential transformer on each tilting truck; hydraulic actuators
Tilt Performance	N/A	Max tilt angle 8°; max tilt rate 4°/sec; max lateral acceleration .194*g* at truck, 0.08*g in cabin*
Tilt Center Height	N/A	1.6m (5'3")
Truck	Radial self-steering, rubber chevron primary, air spring secondary, both axles driven, quill drive	Radial self-steering, rubber chevron primary, air spring secondary
Vehicle Structure	Welded stainless steel; frame/stringer structure, corrugated sheet shell	Welded stainless steel; frame/stringer structure, corrugated sheet shell
Speed	Design 200 km/h (125 mph); has achieved 250 km/h	

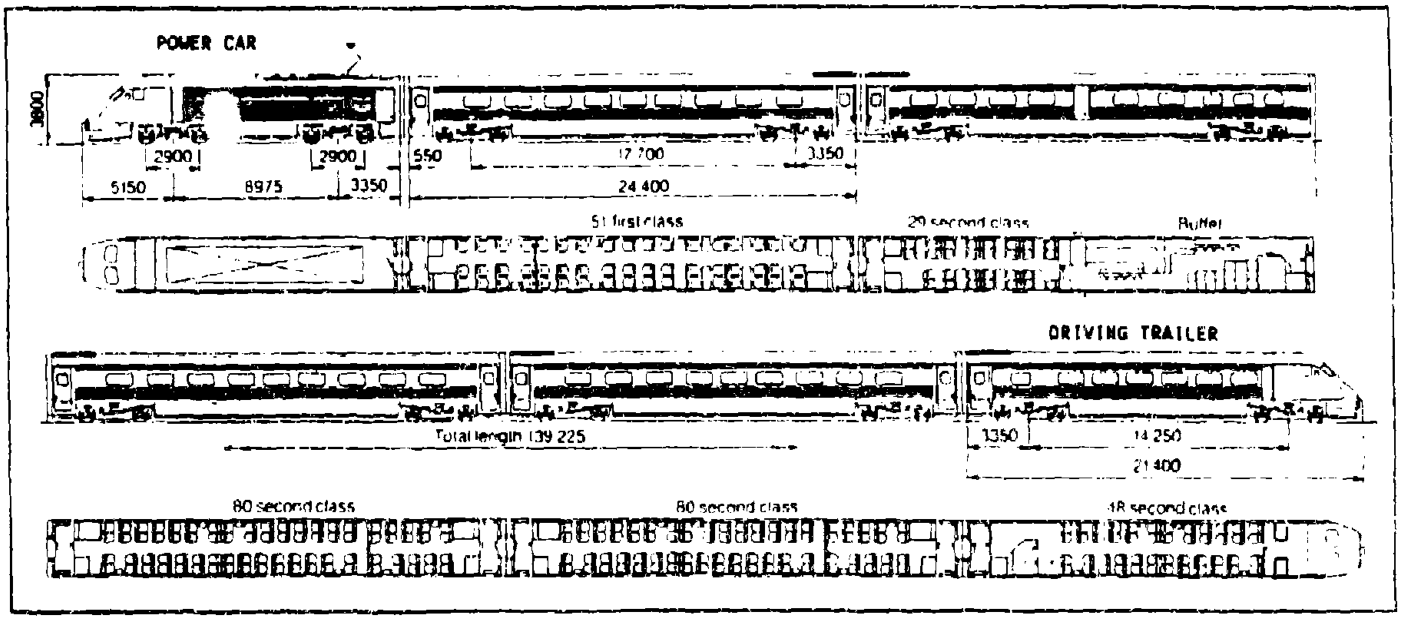


Figure C.22: Layout of X2000 Active-Tilt Trainset

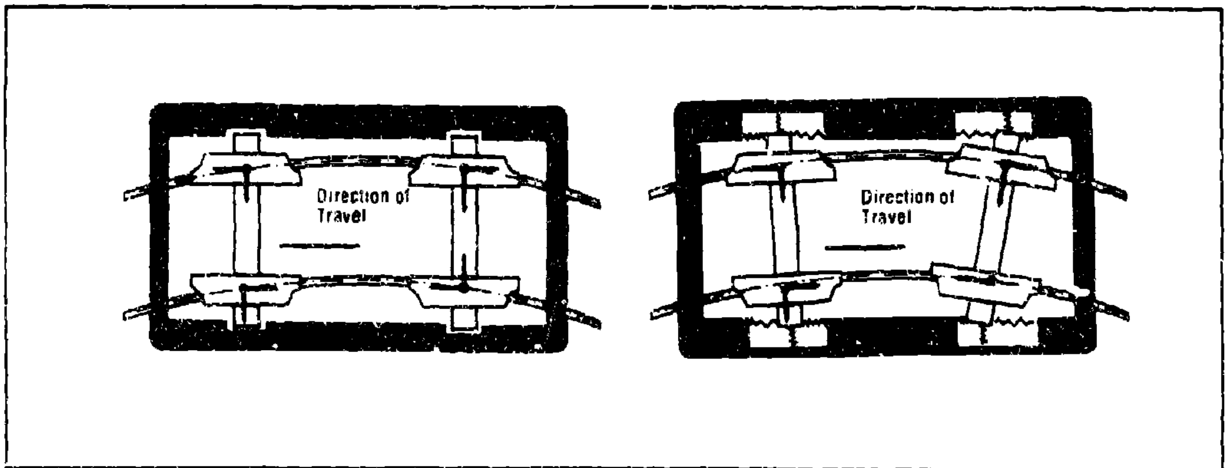


Figure C.23: Parallelogram Effect With Radial Steering Truck

Conventional trucks used on most freight and passenger equipment in the U.S. are rigid, which forces the wheels to move through a curve at some angle of attack to the rail, rather than parallel to it as on tangent track. This causes higher contact forces, increases wheel and rail wear, and imposes greater lateral loads on the track structure. However, conventional trucks are much cheaper to purchase and maintain than more complex steerable trucks. Since systematic costs are not always considered in railroad decision making, and interchange operation means that most cars spend the greater part of their lives off the owning railroad, the incentive to invest in superior technology is limited.

In the ABB truck fitted to the X2000 locomotive, shown in **Figure C.24**, the individual axles are attached to the truck frame by deformable rubber chevrons. This soft primary suspension allows the interaction between wheelset conicity and the creep forces at the wheel-rail contact points to alter the truck/wheelset geometry, permitting up to 40% higher speed without any increase in the level of imposed lateral force or wheel/rail wear. This basic truck design has been in fleet service on ABB-built regional commuter trains for over a decade, with a reportedly excellent reliability and safety record and significant reductions in both wheel and track maintenance requirements.

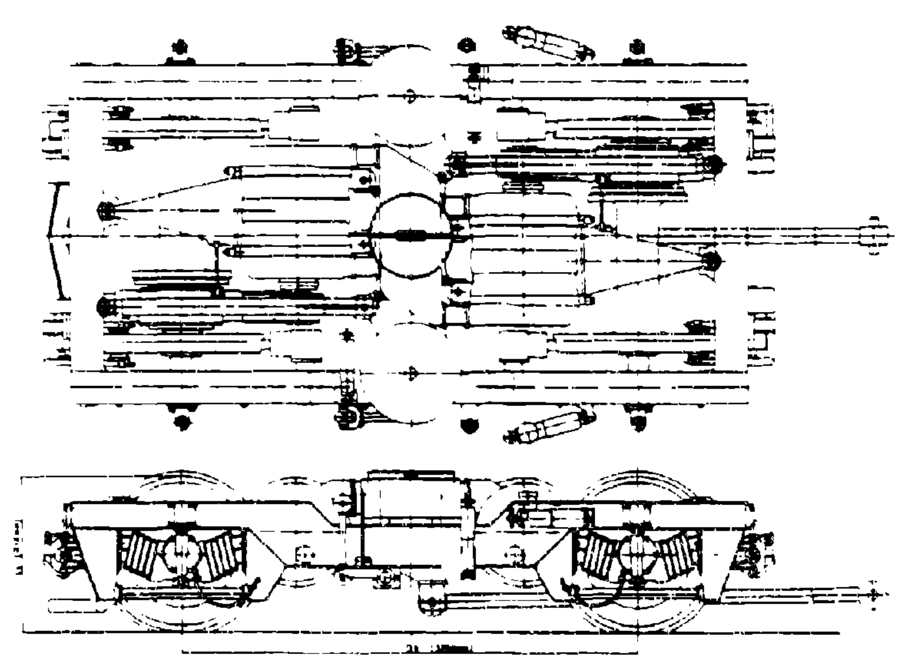
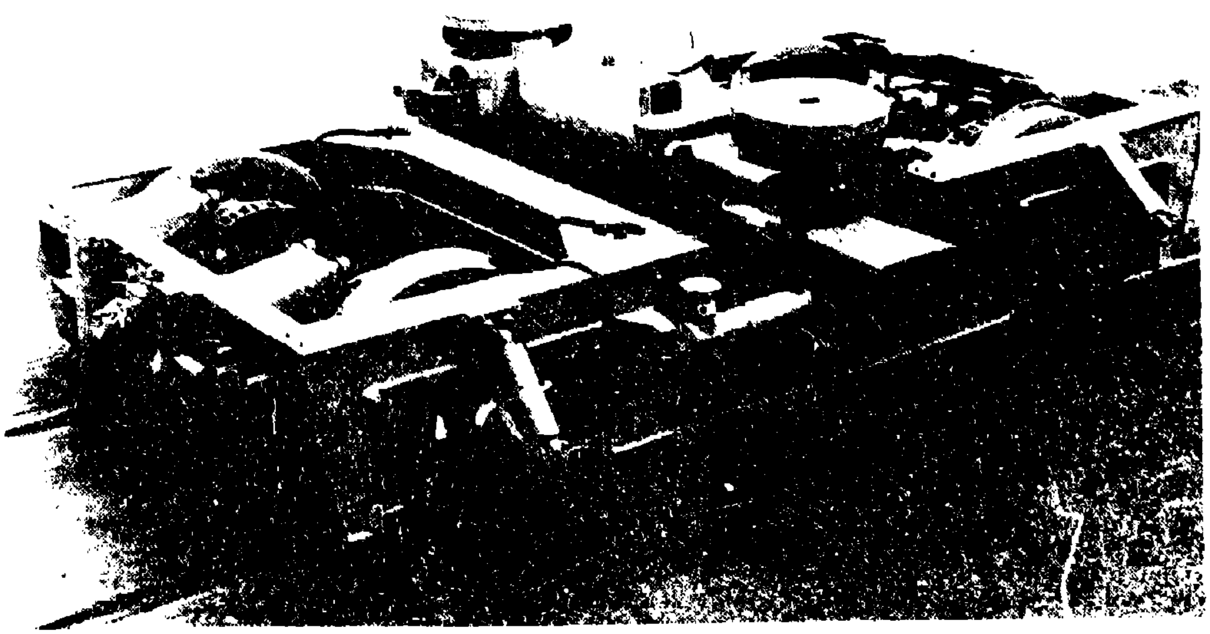


Figure C.24: X2000 Radial-Steering Powered Truck

Stability is achieved through primary hydraulic dampers fitted between the axle boxes and the truck frame, while secondary dampers between the bolster beam and the truck frame control the secondary suspension. Stability of the entire vehicle is ensured by hydraulic yaw dampers which link the vehicle body and the bolster beam to ensure good dynamic behavior at design speed on conventional (European) track.

The bolster beam is connected to the truck frame by a rubber-bushed center pivot. The locomotive body is connected laterally to the truck by two sets of yoke-mounted coil springs on each side. The yokes are connected to the bolster beam by traction rods, which transmit traction forces from the powered trucks to the locomotive body. Each powered truck is fitted with four disc brakes, and each wheel also has a tread brake. Two of the four disc brakes on each truck are equipped to act as parking brakes. To reduce unsprung mass, and thus, control both lateral and especially vertical track forces, each traction motor and flexible transmission is fully suspended from the truck frame, above the primary suspension. Power is transmitted to the axles by means of a flexible quill drive.

The unpowered truck used for the coaches and driving trailer, shown in **Figure C.25**, differs in several ways from the powered version described above. To accommodate the active tilting requirement, the unpowered truck is fitted with two bolster beams, as illustrated in **Figure C.26**. The lower (fixed) beam is attached to the truck frame by flat rubber sandwich bearings, which accommodate any rotational movement between the truck frame and the lower bolster beam, which is linked to the coach body by traction rods. The upper beam is connected to the lower beam by four pendulum links on each truck. The tilt motion takes place between the two bolster beams.



(Source: International Railway Journal, April 1990)

Figure C.25: X2000 Unpowered Radial-Steering Truck With Tilting Bolster

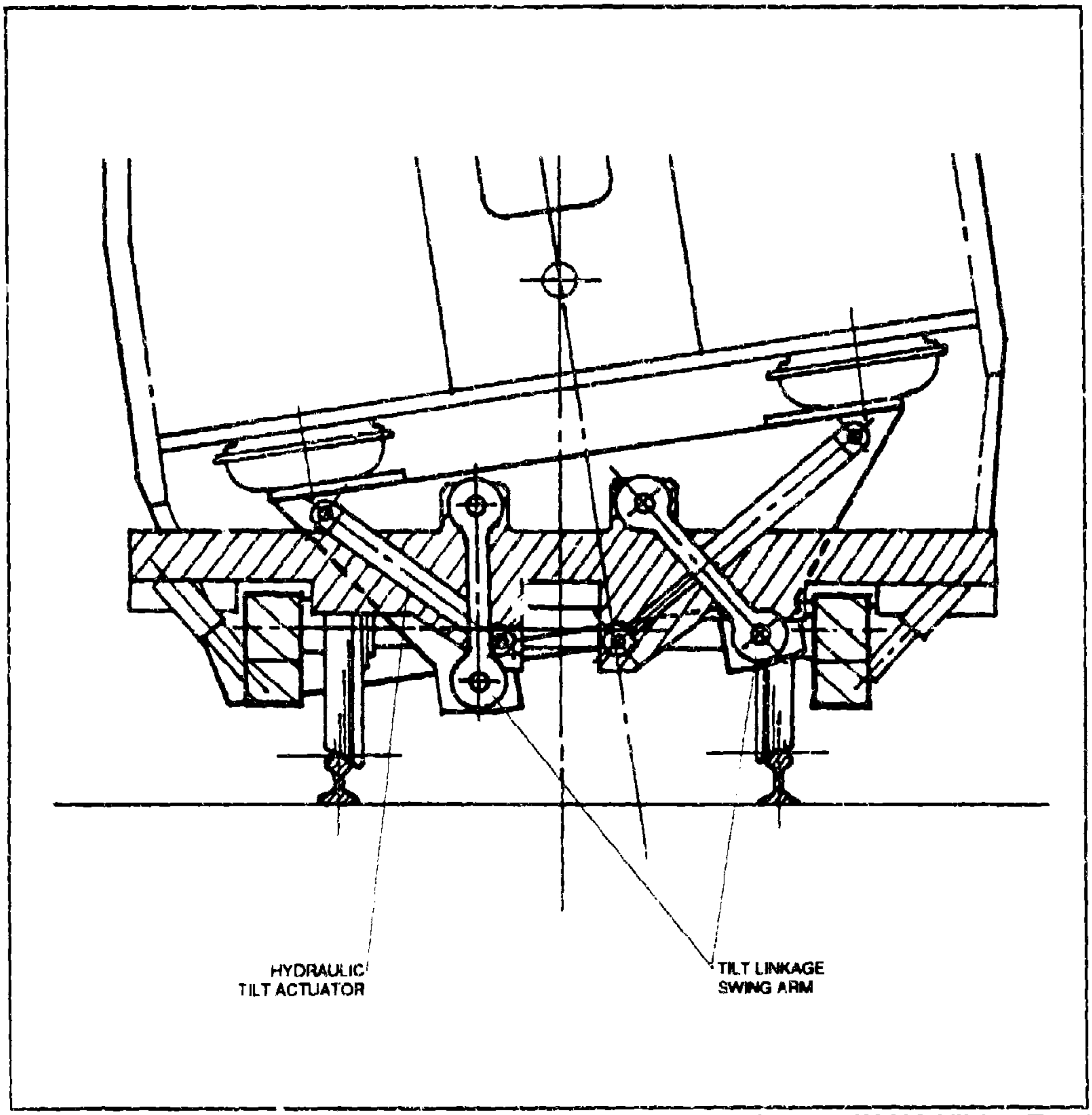


Figure C.26: Configuration of Bolster Beams and Interconnections on X2000 Tilting Truck

The secondary suspension on the coaches consists of air springs between the coach body and the upper bolster beam, with supplemental rubber suspension in the event of an air-spring failure. Each truck is equipped with torsion bars and four yaw dampers linking the upper bolster beam and the coach body.

C.2.3.4 Tilt Control and Actuation Subsystem

To permit negotiation of small-radius curves at high-speed without loss of passenger ride comfort, the X2000 is equipped with an active body-tilting mechanism. Tilting is accomplished by two hydraulic actuators mounted between the upper and lower bolster beams described above.

Figure C.26 shows a cross-section of the vehicle, truck, and tilt mechanism in the untilted position, while **Figure C.27** shows the same cross-section with full body tilt.

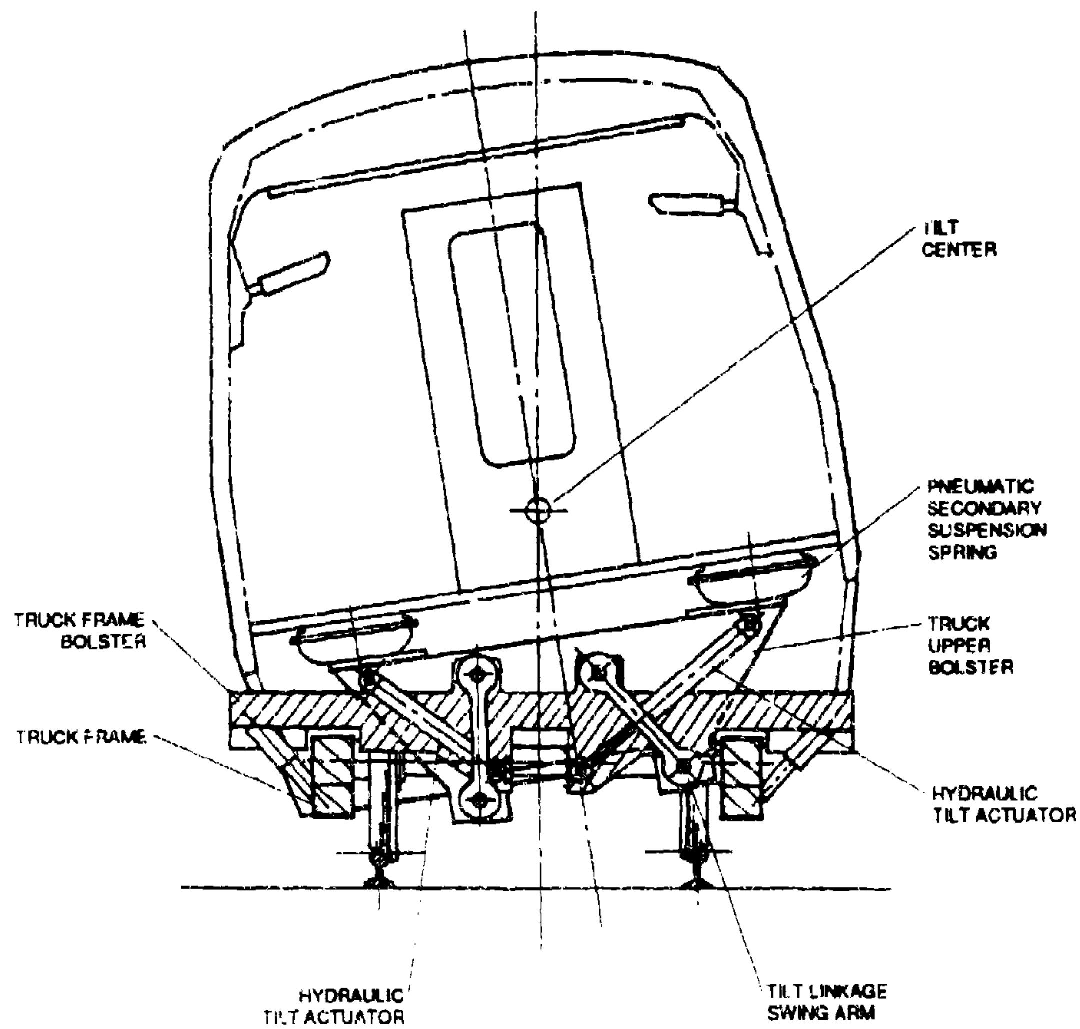


Figure C.27: X2000 Coach Body in Tilted Position

The tilting process is activated when an accelerometer mounted on the frame of the leading coach truck senses lateral acceleration in excess of a predetermined threshold value. The accelerometers generate a signal to the tilting mechanism that is proportional to the lateral acceleration experienced by the leading truck initially as it enters the transition spiral linking tangent track to a curve. This lateral acceleration is approximately proportional to that felt by passengers. The tilt control schematic for the X2000 is shown in **Figure C.28**.

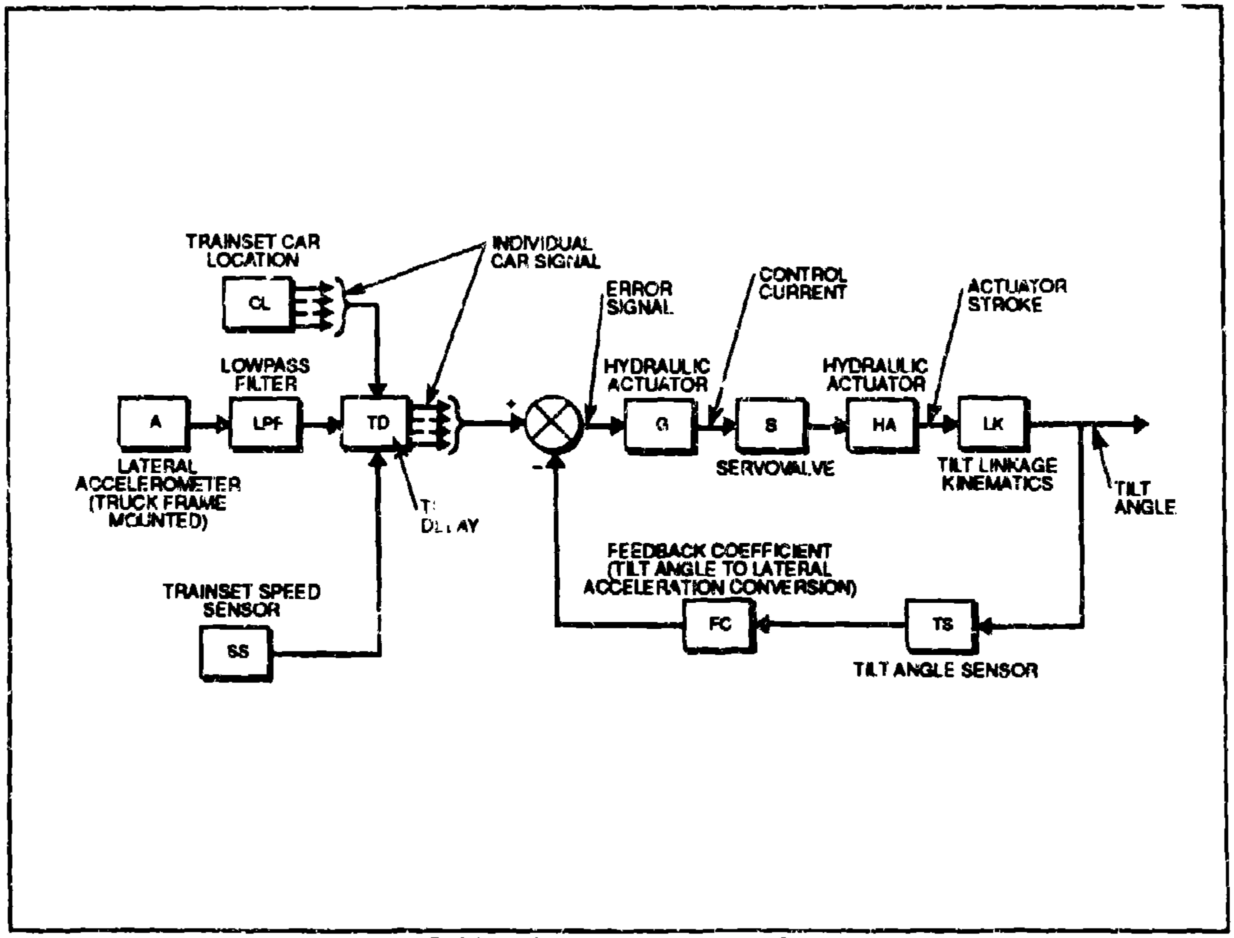


Figure C.28: X2000 Tilt Control Schematic

The signal from the accelerometers is low-pass filtered to ensure that tilting is not initiated in response to track irregularities. The potential delay in the onset of tilting caused by filtering is handled in part by controlling tilt initiation from the leading truck of the power car (or the driving trailer, depending on direction of operation), and in part by detection of the transition spiral. The actual degree of tilt is determined by means of differential transformers linked through microcomputer-based control and fault-monitoring systems. The control system is redundant and fail-safe in design to ensure operational reliability and safety.

The tilting mechanism is essentially similar to the unit developed and tested in the X15-3 high-speed test train. The hydraulic tilting mechanism gives a maximum tilt angle of eight degrees allowing for the effect of differential suspension compression; the effective maximum tilt of the coach body relative to the track surface is 6.5 degrees. The mechanism operates at a maximum tilting rate of four degrees per second. This compensates for about 70% of the centrifugal force, allowing about a 30% increase in curving speed for the same perceived level of lateral acceleration, as shown in Figure C.29.

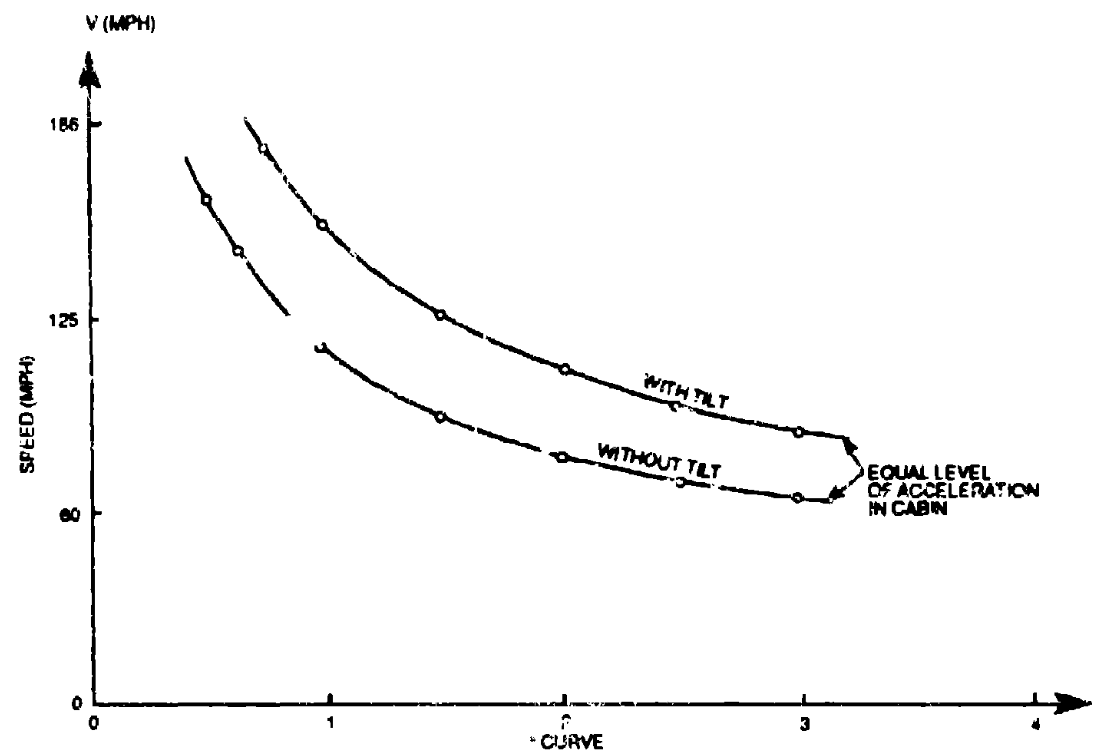


Figure C.29: Effect of Body Tilting on Passenger Compartment Lateral Acceleration

The tilt mechanism itself is composed of a body-mounted pump and filter unit in each coach, and a servo-mechanism mounted above the primary suspension of each truck. The actuating cylinders are also part of the sprung mass, as discussed above.

C.2.3.5 Propulsion and Braking

The electric propulsion subsystem of the X2000 is made up of the roof-mounted current-collection pantograph and the main transformer, and two identical traction power equipment modules, one serving each powered truck. Each module consists of a line converter, a dc link, an inverter, a feed-back chopper which returns the commutation energy to the dc link and the traction-control elements of the on-board microprocessor-based control system. There are four 815 kW ac asynchronous traction motors, one connected to each axle of each powered truck, powered by microprocessor-controlled GTO thyristors. The propulsion system is designed to provide full regenerative braking capability; this, together with the feedback chopper for commutation energy, is expected to save between 5 and 10% of nominal energy consumption.

The pantograph collects single-phase power at 15 kV, 16 2/3 Hz from the catenary. This power is passed to the oil-cooled 3.8 MVA main transformer, which is mounted below the locomotive body and has four separate secondary windings feeding the line converters. The traction module converts the OCS power into variable voltage, variable frequency (VVVF) three-phase power ranging from 0 to 1870 V at frequencies of 0 to 120 Hz. Each module contain 14 GTO thyristors and 14 diodes in the main circuit. Trigger pulses transmitted through fiber-optic cables, which are unaffected by electromagnetic interference, are used to fire the GTO thyristors. Each line converter is made up of two self-commutating GTO thyristor bridges; each pair of converters feed a separate dc link. The associated power factor is very close to one, which yields the minimum line current and the lowest possible energy losses in the catenary. The inverter operates on pulse-width modulation, and controls both motor voltage and frequency simultaneously up to the base speed of the traction motors. Above that speed, voltage is held constant (at 1870 V) and only the frequency rises, giving decreasing motor torque. A feedback chopper returns commutation energy to the dc link.

The 2.4 kV dc links, which normally operate in parallel, can be separated by switches to isolate one of the two powered bogies. This gives additional redundancy to the propulsion system and enhances overall reliability. The capacitor banks in the dc links present a low impedence to the line converters, and are protected against voltage peaks from phase breaks or catenary arcing. The four-pole, three-phase asynchronous motors have a squirrel-cage aluminum rotor and are force-ventilated. The stator is cast steel with vacuum-impregnated shaped windings. Test results have shown the motors to be very robust and almost insensitive to dirt and vibration. Auxiliary circuits are fed from an 840V winding on the main transformer, while a 996 V winding supplies heating power. A static auxiliary power inverter with an output of 360 kW supplies three-phase power at 380 V, 50 Hz for other on-train equipment (this is a standard European industrial-motor power supply).

C.2.3.6 Other Features

The vehicle frames and bodies are stainless steel, but are designed to meet the UIC 651 standards for vehicle strength (200 tonne [220 ton] buff load, versus 364 tonnes [400 tons] under existing U.S. standards). Although heavier in absolute terms than aluminum or a light alloy structure, stainless steel provides better stiffness per unit weight. ABB determined that use of stainless steel optimized vehicle weight and stiffness, allowing construction of 24.5 m (80 ft) coaches with a relatively high (11 Hz) natural frequency. The aerodynamic nose panels of the locomotive and driving trailer are formed from glass fiber reinforced plastic. The cab areas are protected by high-strength deformable steel cages to protect the operator in the event of collision with another rail vehicle, or (more likely) with a truck or automobile at a level crossing.

Given the dominance of aerodynamic drag as the source of running resistance at the design speed of this equipment, it is not surprising that considerable care has been taken to avoid parasitic drag. Exterior fittings, such as handles, are recessed and windows, doors and steps are flush-mounted. Under-floor equipment is enclosed in a corrugated belly-pan that not only improves trainset aerodynamics but also protects equipment from dirt, snow and moisture, and enhances heat dissipation. Retractable air intakes are at roof level, and the pantographs have a roof fairing to reduce drag.

Stainless steel construction also provides advantages during production. Once welding is completed, equipment and other steels fittings can be installed. For example, underfloor wiring and piping can be mounted before the floor structure is turned over for installation of the plywood floor and erection of the vehicle walls. All interior fittings are isolated from the steel structure by rubber mountings. In combination with normal insulation, this reduces interior noise levels to 60 dB(A) in the coaches and 68 dB(A) in the cab areas. SJ has imposed a requirement that exterior noise for the X2000 at 200 km/h (125 mph) not exceed those for conventional locomotive-hauled trains at 130 km/h (81 mph), i.e., about 85 dB(A) on well-maintained track.

Coach interiors have been designed to appeal to business travellers, with full upholstery, adjustible seats, reading lights, at-seat headphones with taped music and radio programming, an effective P.A. system and, in first class, computer work stations, and three conference compartments equipped with telephones and computer facilities.

The X2000 depends on an on-board computerized control system called TRACS for data collection, processing and analysis. This system integrates control of traction and braking with control of other on-board functions. The control structure is made up a number of decentralized computers linked to a master unit in the locomotive. Data sensors, data transmission, the "master" and "slave" computers and various process control units are connected through a single common parallel bus. Each coach is equipped with a "slave" computer that controls tilt, braking including anti-skid protection, and door operation. There are repeater screens in each driving cab to transmit driver commands and verify receipt. Traction control is a major function of this TRACS system. The firing of the GTO devices is computer-controlled, with the rate of rise in power supply drawn by one trainset limited to 300 kW/second. Power draw is automatically restricted if the catenary voltage falls by 20% (i.e., to 12 kV). Motor and coolant temperatures and an array of line and component currents are automatically monitored against designated limits.

C.2.4 JAPAN RAILWAYS MODIFIED SERIES 381 EMU, SERIES 2000 DMU, AND 250X*

C.2.4.1 Overview

Beginning in the late 1970s, Japan National Railways (JNR) began to address the inability of the passive-tilt Series 381 EMU equipment to meet its requirement for improved operating speed and trip times and acceptable levels of passenger comfort on the very sinuous narrow-gauge lines in the central mountains of Japan, and indeed on the narrow-gauge network generally. The recognition of this problem led to development of a potentially very powerful tilt control strategy and to the only pneumatic active-tilt mechanism currently in use or under development.

The initial stage in the development process involved the design, fabrication and testing of a tilt control and actuation mechanism that could be retrofitted to a Series 381 trainset. After considerable investigation and component testing through 1983, a practical pneumatically actuated active tilt subsystem was developed and installed in one series 381 trainset, as shown schematically in **Figure C.30**.

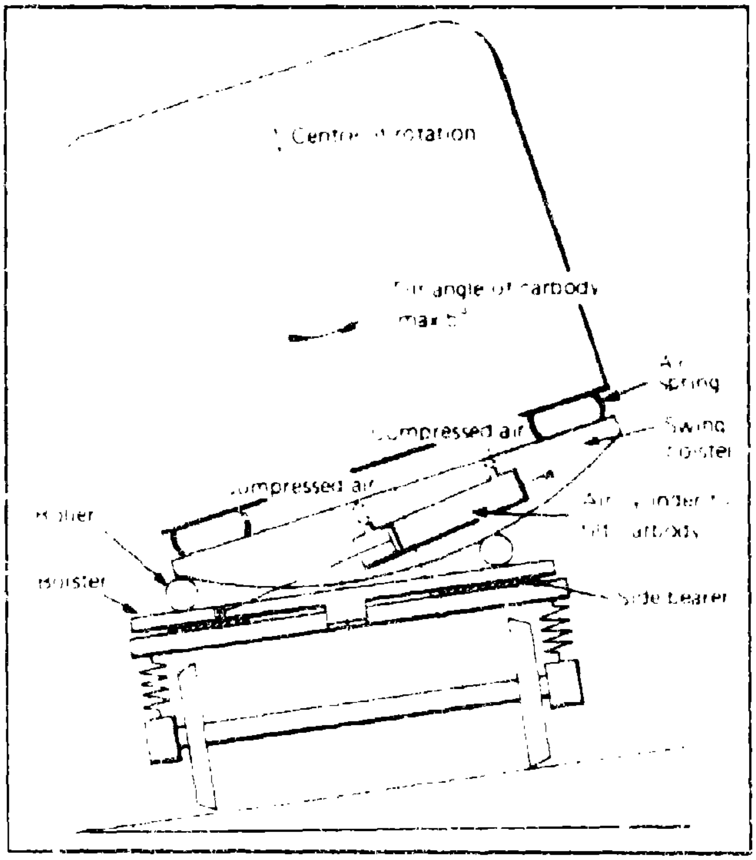
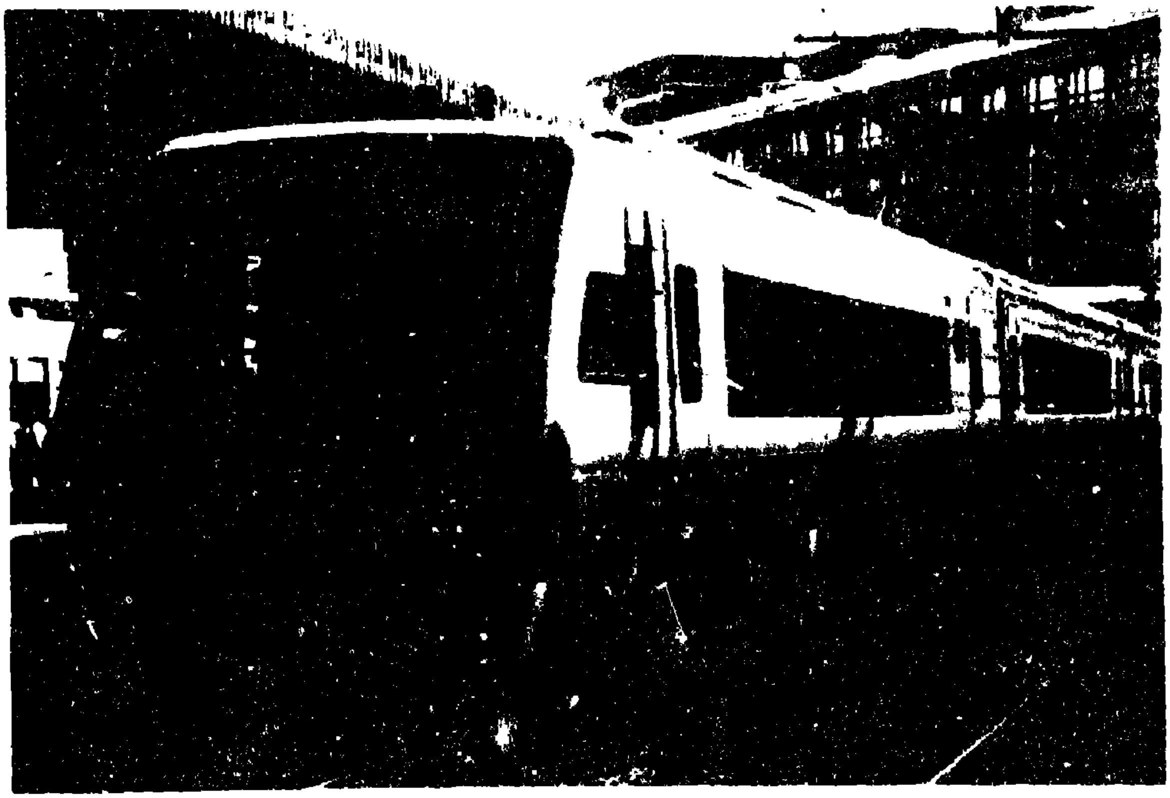


Figure C.30: Schematic Cross-Section of Car Equipped With Pneumatic Active-Tilt System

Tests with the retrofitted active tilting system and with other technical improvements to trucks and brakes began in 1983 and extended through 1985. The results were quite satisfactory, and demonstrated that trainsets so equipped could achieve the relatively modest original objectives for speed improvement, to 130 km/h (81mph).

The retrofit tilt controller limits the carbody roll rate to about five degrees per second and the roll acceleration to about 15 degrees/sec^2 to achieve acceptable ride quality, and incorporates a look-ahead feature to ensure tilt actuation in advance of curve entry. These roll motion limits reflect the apparent dependency of "tilt nausea" on the carbody roll motion in combination with reversals of roll direction as the vehicle enters and exits transition curves.

The success of the tests with the retrofitted Series 381 EMU equipment encouraged JNR and more recently, JR-Shikoku, one of the successor companies following the breakup of JNR, to pursue the design and construction of diesel-electric active-tilt equipment, designated the TSE (Trans-Shikoku-Experimental) Series 2000 diesel motor unit (DMU), shown in **Figure C.31**.

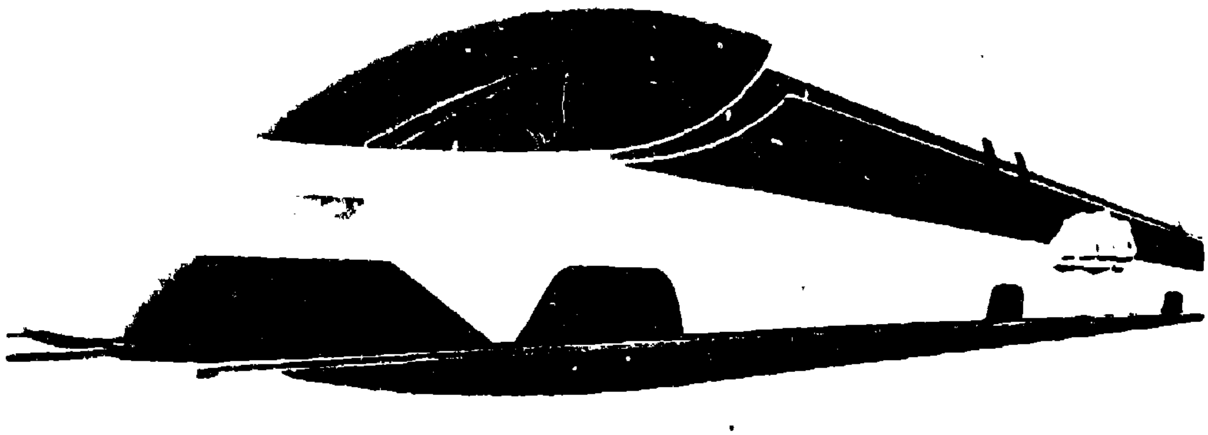


(Source: Railway Technology International 1991)

Figure C.31: The JR-Shikoku TSE-2000 Active-Tilt DMU

This vehicle, constructed by Fuji Heavy Industries, incorporated a tilt control and actuation subsystem derived from that developed for the Series 381 retrofit. Construction of the TSE 2000 prototype began in 1987, with testing completed by early 1989. In March 1989, the first TSE 2000 three car express trainset entered revenue service on Shikoku Island, running on the narrow-gauge lines between Takamatsu and Matsuyama (200 km/125 miles) and Takamatsu and Kochi (160 km/100 miles), with connection to the Shinkansen system at Okayama on Honshu Island. There are now 38 active-tilt diesel multiple unit (DMU) trainsets in operation in the JR-Shikoku fleet.

In the spring of 1991, the Japanese Railway Technical Research Institute (RTRI) announced a project to develop a new trainset, designated the 250X, for high-speed (250 km/h:156 mph) operation on narrow-gauge track. The objective of this challenging assignment is to allow the six passenger railways of the JR Group to extend the benefits of Shinkansen-quality operation to be accessible only on the ubiquitous narrow-gauge network. RTRI have clearly stated that active body tilting will be an essential feature of the 250X; a concept drawing is shown in **Figure C.32**.



(Source: Railway Gazette International, June 1991)

Figure C.32: Concept Drawing of the 250X High-Speed Narrow-Gauge Trainset

The 250X conceptual design incorporates a number of innovative features besides active-tilting, most notably power-steered, independent wheels driven by individual hub motors. The concept is discussed in greater detail below.

C.2.4.2 TSE 2000 Technical Specifications

Table C.9 summarizes the major technical specifications of the TSE 2000 active-tilt DMU equipment.

.2.4.3 Tilt Control and Actuation Subsystem

The TSE-2000 incorporates a unique tilt control and actuation subsystem based on the use of wayside location transponders and an on-board data file containing all curve location and track geometry data for a specific route, as shown in the schematic in **Figure C.33**. **Figure C.34** illustrates how this system operates and the relationship between track geometry and the tilt angle of the TSE trainset.

Table C.9
Characteristics of the TSE-2000 Active-Tilt DMU

Vehicle Dimensions	End Car 1	Intermediate Car	End Car 2
Length	20.8m (67.8')	20.8m (67.8')	20.8m (67.8')
- Width	2.84m (9.3')	2.84m (9.3')	2.84m (9.3')
- Height	3.39m (11')	3.39m (11')	3.39m (11')
Weight			
Empty (metric/U.S.)	37.7 (41.5)	36.9 (40.6)	37.7 (41.5)
- Loaded "	40.5 (44.6)	40.2 (44.2)	40.3 (44.3)
- Axle load	10.1 (11.1)	10.1 (11.1)	10.1 (11.1)
Seating	46	54	43
Performance			
Maximum Speed		120 km/h (75 mph)	
Sustainable speed, 2.5% gradient		95 km/h (59 mph)	
Curve Limits	Curve radius (m/ft)		Speed (km/h;mph)
	More than 600m (1956')		Standard + 30 (19)
	400m to 600m (1304'-1956')		Standard + 25 (15.6)
	Less Than 400m (1304')		Standard + 20 (12.5)
Body Construction	Welded stainless steel		
Propulsion	Two contra-rotating turbo-charged 246 kW diesel engines, driving the inboard axle of each truck through a torque converter, an electromagnetically activated automatic gearbox and a telescopic cardan drive shaft to a right-angle final drive		
Tilt Control and Actuation	Controller using on-board route geometry file, wayside location transponders and on-board tilt and speed sensors; actuation by pneumatic cylinder		
Tilt System Performance			
Tilt Angle	5 degrees maximum		
Tilting Rate	5 degrees/second maximum; 15 degrees/sec² tilt angle acceleration		
Uncompensated Lateral Acceleration	1.64 m/sec², (0.16724g) at truck; 0.78 m/sec² (0.08*g*) in cabin		
Tilting Center Height	2.275 m (7.42')		
Brakes	Electro-pneumatic disc		

As the trainset approaches a run-on transition, determination of requirements for tilt onset will occur in one of two ways. In some locations, there will be a wayside transponder that forms part of the automatic train signalling (ATS) system used by JR-Shikoku. These transponders have been configured to also provide accurate absolute train location data. Alternately, between such transponders, the on-board computer will calculate train location based on speed/distance signals from sensors mounted on the first and fourth axles of the lead vehicle. This relative measure of location is updated by the absolute location signal from the next transponder, so that there is no propagation of location error between signal blocks.

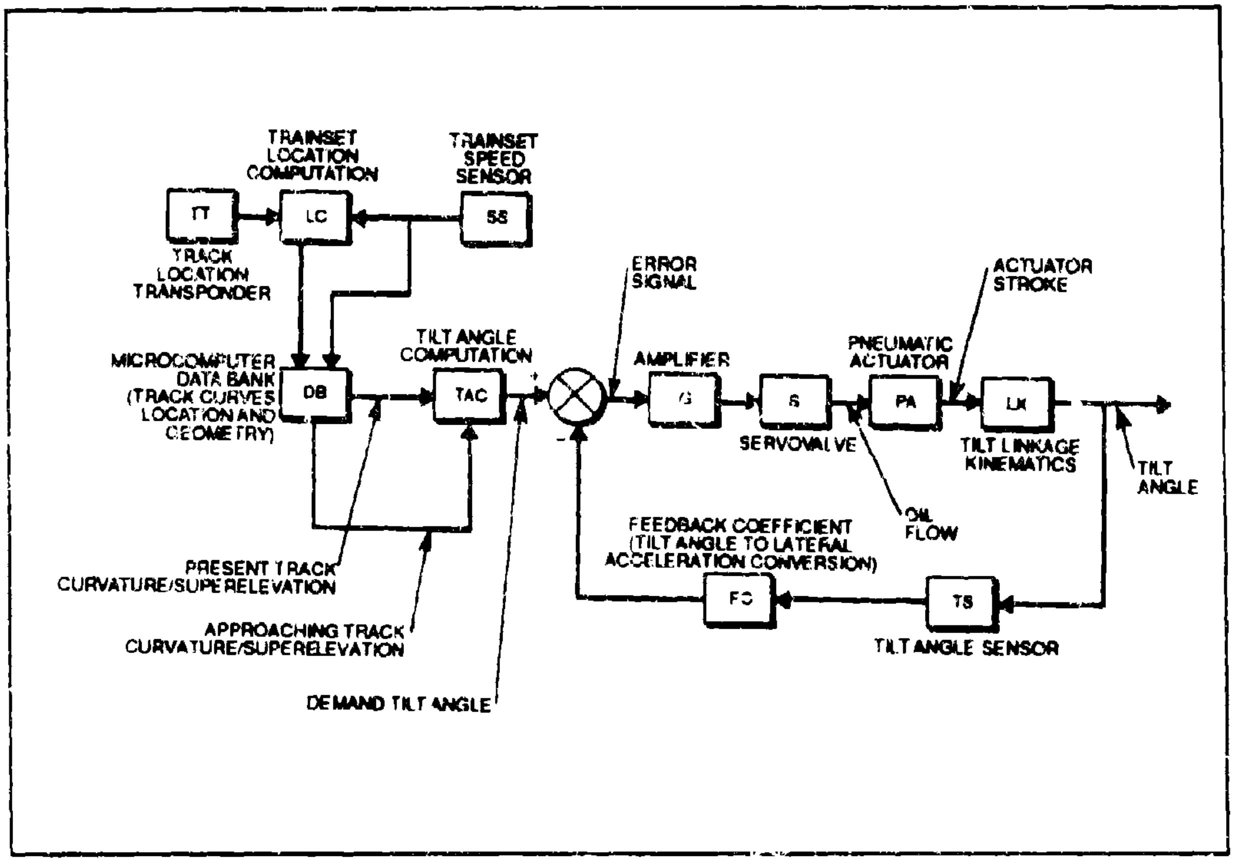


Figure C.33: Tilt Control System Schematic for TSE-2000 DMU

The on-board command controller then matches the train location with the appropriate curve location and geometry data from the route data file downloaded from a master command controller prior to departure. This allows the command controller to determine the required timing of tilt onset in advance of entry into the transition curve, and to send the necessary command to each of the on-board tilt controllers (one per car) with enough lead time (equivalent to 50m [163 ft] - the "A" spot in **Figure C.34**) to ensure that tilt actuation takes place sufficiently far in advance of changes in curve geometry. The command signal also tells the tilt controller the complete sequence of tilt magnitude and tilting rate required to match the curve entry and exit sequences.

The tilt controller operates the tilt actuation mechanism through an electro-pneumatic servo-valve based on the tilt requirements established by the command controller.

The tilt actuation mechanism, shown in **Figure C.35**, consists of an air cylinder mounted diagonally between the truck bolster and the truck pendulum beam which supports the pneumatic secondary suspension springs. The tilting bolster is fabricated with circular sections on its lowersurface; these circular sections rest on rollers mounted on the truck frame. The power for the tilt actuation mechanism is provided by an on-board air compressor.

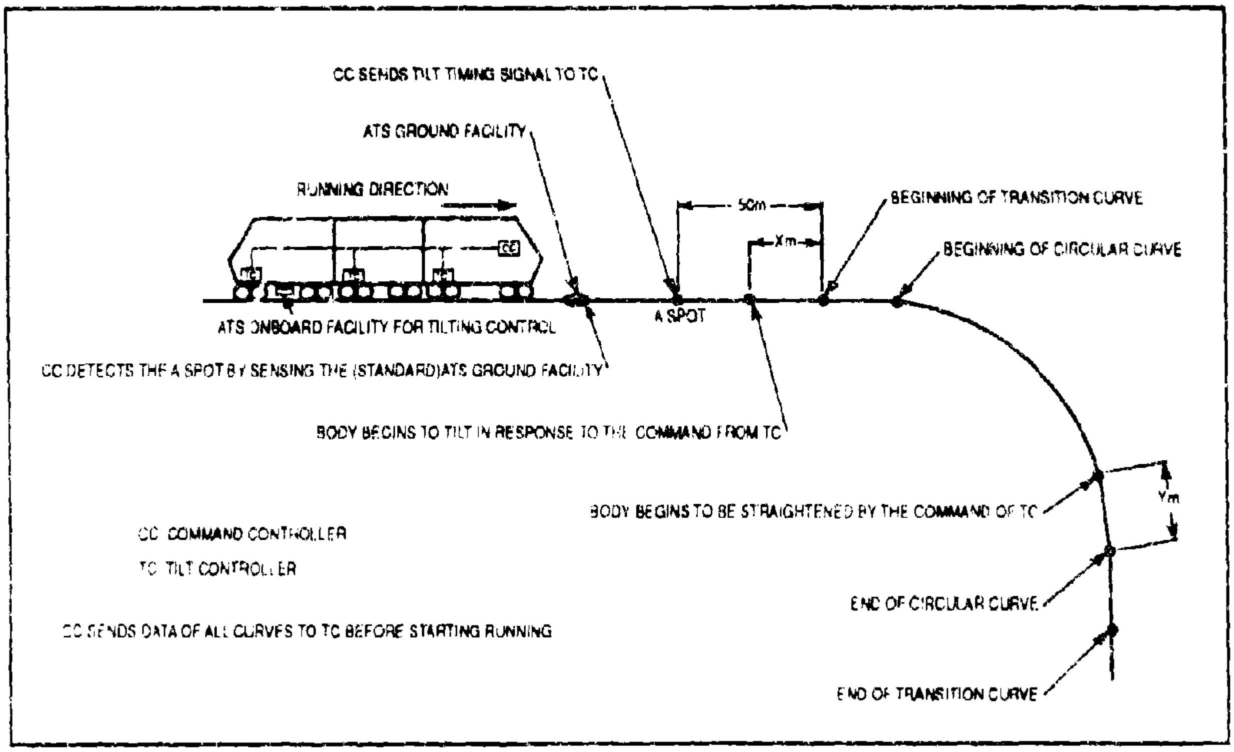


Figure C.34: Operation of TSE-2000 Tilt Control System (Top) and Relationship Between Curve Geometry and Carbody Tilt Angle (Bottom)

C.2.4.4 The 250X Conceptual Design

As noted in the Overview to this section, in the Spring of 1991, the RTRI announced that it is developing an advanced high-speed trainset for operation on the narrow-gauge network. A concept drawing is shown in **Figure C.32**. This development project reflects the political and financial reality that most of the cities served by the widespread narrow-gauge network operated by the six-passenger railways of the JR Group will not receive Shinkansen services. The only practical alternative is a very significant improvement in comfort and performance on the narrow-gauge lines.

The initial design concept released by RTRI incorporates a number of advanced features, most notably short, very light-weight, articulated carbodies fabricated from light alloys, and/or advanced materials such as fiber-reinforced plastics (**Figure C.36**), single-axle trucks with independent wheels driven by individual hub motors (**Figure C.37**), and "intelligent" on-board monitoring and control systems.

It will be interesting to monitor the progress of this project over the next few years, especially insofar as many of the advanced concepts will be equally applicable to standard-gauge equipment, and thus could be of direct value for applications in the U.S.

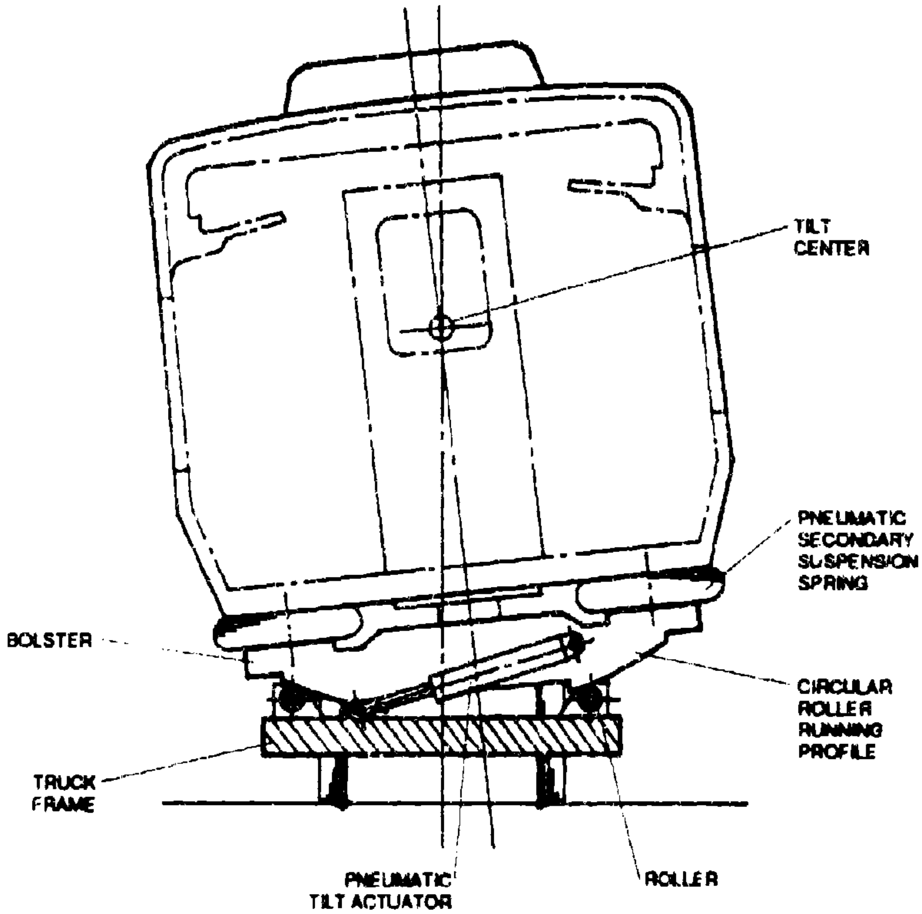
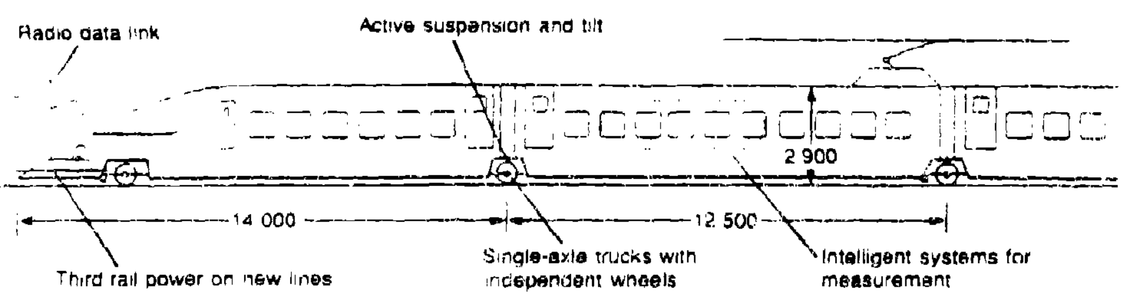
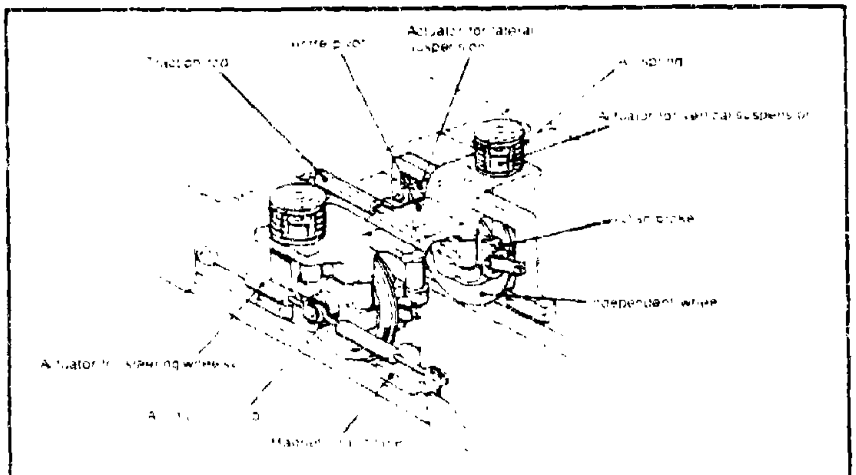


Figure C.35: Tilt Mechanism for TSE-2000 Trainset



(Source: Railway Gazette International, June 1991)

Figure C.36: 250X Design Concept



(Source: Railway Gazette International, June 1991)

Figure C.37: 250X Truck Concept

C.3 PASSIVE-TILT TECHNOLOGIES

C.3.1 THE TALGO *PENDULAR*

C.3.1.1 Background

The Talgo *Pendular* passive-tilt coaches, illustrated in **Figure C.38**, originated as an evolutionary development of conventional broad-gauge equipment produced by Patentes Talgo S.A. of Madrid, Spain. In response to a 1974 requirement on the part of Spanish National Railways (RENFE) for improved service on its domestic and especially its international run-through services to France, Talgo S.A. initiated development of the *Pendular* rolling stock.

The development program proceeded smoothly, with testing of the first prototype beginning in 1976. This test progam included running at a speed of 200 km/h (125 mph). The initial order for the production equipment was placed by RENFE in 1977, and the consists entered service in 1980. **Table C.10** summarizes major milestones in *Pendular* development and operating history.

In addition to the passive-tilt system based on differential compression of roof-level airbags that also form the secondary suspension springs, most of the Talgo fleet is equipped with unique single-axle, radial-steering trucks with independent wheels and provision for an automated change of gauge to accommodate both the RENFE broad-gauge lines and the standard-gauge track of France, Italy, and Switzerland.

Although DB has so far failed to follow-up on its earlier interest in the Talgo equipment for selected international daylight services, there is apparently an initiative to create an operating company to provide sleeper services on the Berlin-Munich, Berlin-Zurich and Zurich-Vienna

Figure C.38: The Talgo *Pendular* Passive-Tilting Trainset

routes, using Talgo equipment. These services would require about six long trains of 26 to 28 coaches each. Lead time for the equipment would be about two years from receipt of a firm order. Negotiations to create the operating company were still underway in June 1991.

Of particular interest in the present context is the test experience and quantititive assessment of performance on the Northeast Corridor. A Talgo trainset with Pendular passive tilting coaches was included as part of the on-going CONEG/Amtrak high-speed equipment demonstration project during the spring and fall of 1988. Measurements of carbody lateral acceleration in curves at speeds up to eight inches of cant deficiency showed that the passive tilt system, although not eliminating the steady-state lateral acceleration completely, certainly kept the level below 0.1 g and offered a safe and excellent ride much superior to the non-tilting coaches tested concurrently. In fact, the Talgo coach was superior on short and rough-entry curves, with peak-to-peak lateral "jolt" less sensitive to cant deficiency than the active-tilt LRC trainset also included in the demonstration. A conclusion of the tests was that the TALGO trainset could operate at higher cant deficiency speeds without compromising passenger comfort and derailment safety limits, thus offering the potential to reduce trip time on existing track considerably.

C.3.1.2 Technical Specification

Table C.11 summarizes the major technical specifications of the Talgo *Pendular* rolling stock.

Table C.10: Major Milestones in Talgo *Pendular* Development and Operation

1974:	Start of development program for passive tilt Talgo Pendular equipment based on earlier non-tilting design
1976:	Testing of prototype Talgo passive-tilt trainset at speeds up to 200 km/h (125 mph)
1977:	First production Talgo *Pendular* trainsets ordered by RENFE
1980:	Talgo Pendular trainsets enter revenue service in Spain; top speed 180 km/h (112.5 mph)
1986:	RENFE decides all new Talgo trainsets to be *Pendular* passive-tilting type
1987:	Talgo trainset tested on U.S. Northeast Corridor by Amtrak/CONEG
1988:	Talgo Pendular 200 car designed for 200 km/h (125 mph) ordered by RENFE
1988:	Talgo trainset tested by DB on conventional and high-speed lines; maximum speed achieved was 288 km/h (180 mph)
1989:	428 Talgo Pendular cars in RENFE fleet
1989:	Talgo Pendular service extended into Switzerland and Italy; frequency to French destination increased
1989:	Development of 250 km/h (156 mph) version of Talgo *Pendular* announced
1989:	DB announces plan to acquire Talgo Pendular equipment for selected international services, but acquisition delayed due to events in Eastern Europe
1991:	Prototype Talgo Pendular trainset for 250 km/h operation begins testing

C.3.1.3 Tilt Subsystem

The passive-tilting mechanism used in the Talgo *Pendular* is illustrated in **Figure C.39**. The principal of operation of the system is quite simple: by locating the air springs that make up the secondary suspension on top of tall support pedestals, in pockets at roof level **(Figure C.40)**, and pivoting the carbody so the effective tilt center is *above the roof of the car*, differential compression of the airbags allows the natural pendulum action of the carbody to occur.

The actual height of the tilt center varies between 2.8 and 3.4 m (9 ft 2 in to 11 ft 1 in), depending on the car type and loading. The nominal car floor height is just 0.63m (25 in) above the rail head, a consequence of the carbodies being supported between the shared single-axle trucks.

Table C.11: Major Specifications of the Passive-Tilt Talgo *Pendular*

Vehicle Dimensions	
- Length	13.1m (42.7')
- Width	2.95m (9.6')
- Height	3.33m (10.8')
Weight	
-Intermediate car	13.2 tonnes (14.5 tons, average)
-End car	16.6 to 19.1 tonnes (18.2 to 21 tons)
Vehicle Construction	Light-weight self-supporting semimonocoque structure of welded aluminium alloy extrusions and rolled sheet
Maximum Axle Load	11.8 tonnes (13 tons); 5.9 tonnes (6.5 tons) per independent wheel
Operational Speed	180 km/h (112.5 mph) maximum for original (1977 order) fleet; 200 km/h (125 mph) for later additions; 250 km/h design speed for latest version currently under development; service speed limited to 140-160 km/h (87-100 mph) on some routes.
Seating	Varies depending on interior configuration; RENFE Ist class has 1 X 2 seating (26 seats per car) with 2 X 2 in second class (36 seats per car)
Propulsion	Cars only; most conventional (European) diesel or electric locomotives could be used but special 4000 hp (3000 kw) low-profile non-tilting diesel hydraulic locomotives are used on most RENFE services.
Trainset Makeup	Typically single locomotive, Talgo service car, 10 intermediate Talgo cars, and a Talgo end trailer car but shorter or longer consists can be made up depending on traffic.
Truck	Single-axle design supporting two articulated cars, with independently suspended and independently rotating wheels on radially-steered half-axles; design permit automated adjustment of gauge for run-though services between Spain (broad gauge) and France, Italy and Switzerland (standard gauge).
Brakes	Wheel-mounted hydraulic disc brakes with pneumatic anti-skid control
Tilt	Passive, to 3.5° or 5°, depending on version

The tilting mechanism is set up to permit a maximum tilt angle of 3.5° (original order) or 5° (later orders). Tilting rate is not regulated, being determined by the carbody inertia, tilt-center location and transition curve geometry. However, the mechanism only functions when train speed exceeds 70 km/h (44 mph) on track with a radius of curvature of 1,500 (4890 ft) or less. This lockout is achieved by shutting the air spring level adjustment valve for each pair of airbags.

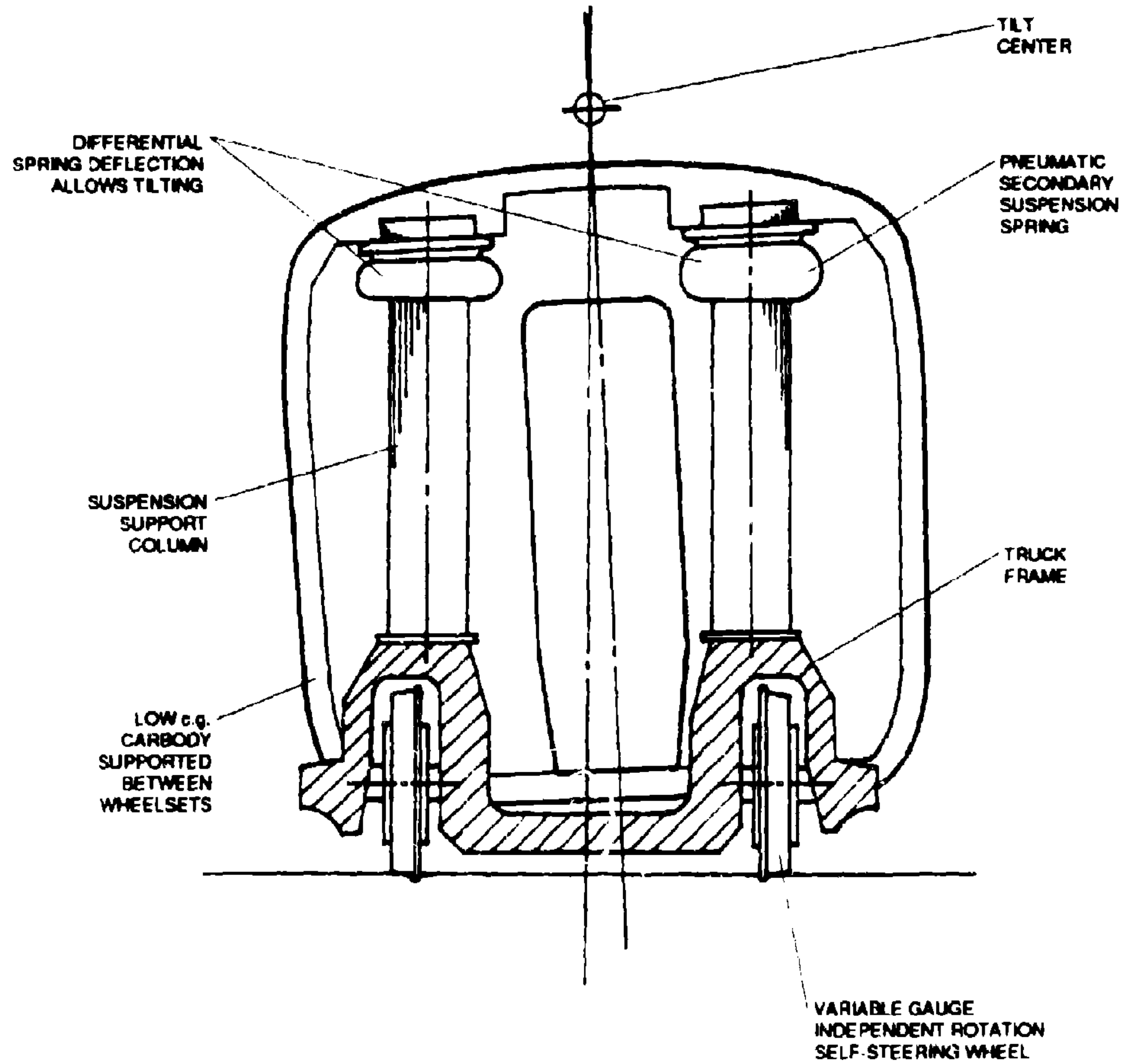


Figure C.39: Talgo *Pendular* Passive Tilt Mechanism

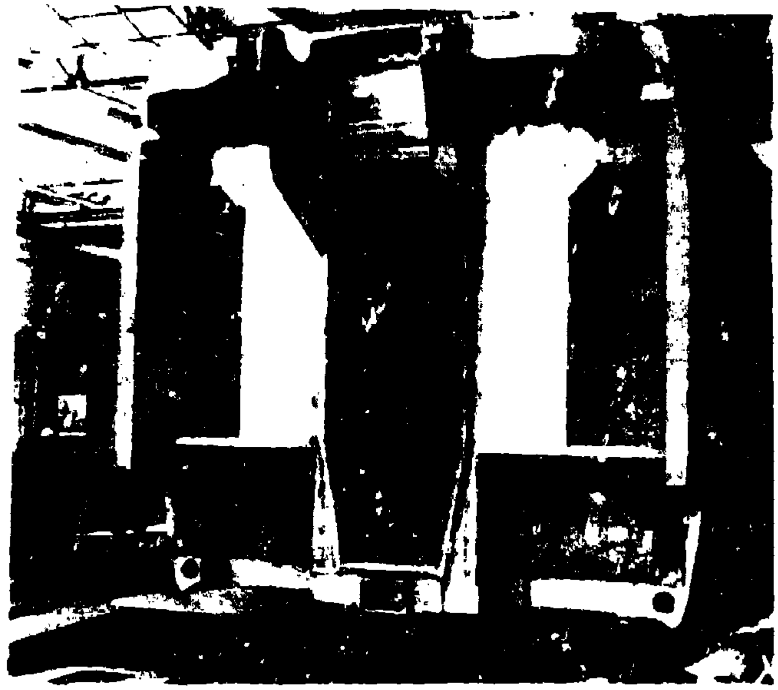
In RENFE service, the maximum allowable uncompensated lateral acceleration measured at the truck is held to 1.75m/sec^2, or 0.178*g*, with 0.09*g* accepted in the cabin. During steady-state curving (with zero uncompensated acceleration in the cabin), the maximum uncompensated lateral acceleration at the truck is 0.85m/sec^2, or 0.087*g*.

C.3.1.4 Trucks

The Talgo *Pendular* truck shown in **Figure C.41**, incorporates a number of interesting features in addition to those specifically required for passive body tilting. The two-wheel between-car articulation trucks feature independently suspended and rotating wheels on half axles; these wheels are radially steered by a linkage between the bearing housing of each wheel and the two adjacent carbodies.

The primary suspension for each wheel is made up of a large single coil spring connected to the adjacent carbodies by linkages which control the vertical and lateral displacement of the axle bearing housing. The secondary suspension is provided by a rolling diaphragm type of soft air spring. As noted above, the secondary suspension air springs are supported on tall pedestal columns so as to be located close to the carbody roof. Braking is provided by wheel-mounted disc airbrakes with anti-skid control.

The other unique feature of the Talgo truck, albeit one of no particular relevance to U.S. applications, is its automated gauge-change capability, essential for run-through services between Spain and the rest of Europe.



(Source: Railway Gazette International, December 1989)

Figure C.40: Talgo Carbody End Structure; Note Roof-Level Pockets for Airbags

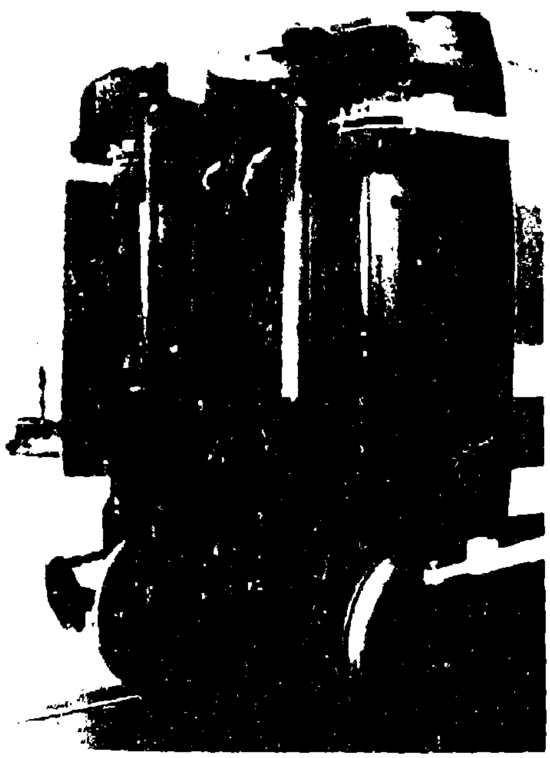


Figure C.41: Talgo Articulation Truck

C.3.1.5 Other Features

The Talgo *Pendular* design incorporates unusually short carbodies (13.1 m [42 ft 8.5 in] length as compared with a bit over 25 m [82 ft] for the ETR 450 and X2000, 20.8 m [68 ft] for the JR Series 381 and 17.5 m [57 ft] for the VT-610). This facilitates achievement of low axle loads even with the two-wheel articulation truck.

C.3.2 THE SIG NEIKO PASSIVE-TILT SUBSYSTEM[I]

C.3.2.1 Overview

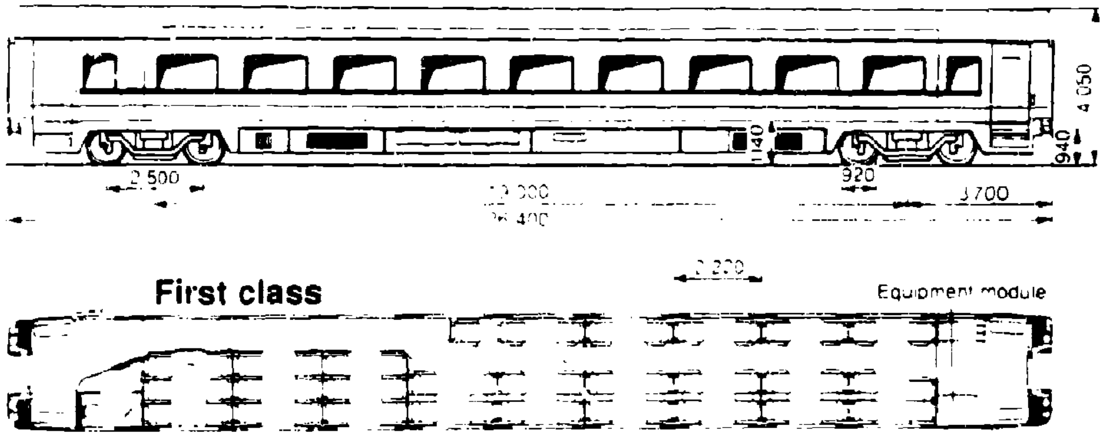
As befits a national railway that serves one of the most mountainous countries in Europe, Swiss National Railways (SBB) have had a relatively longstanding interest in techniques to improve ride quality, speed and/or safety on highly curved track. In the early 1970s, SBB built and tested three prototype active-tilting high-speed passenger coaches. The active tilting system provided a six-degree maximum tilt angle, with a tilt center near seat level and a control system based on a gyroscope reference signal.

Subsequently, the Mark III passenger coaches purchased by SBB incorporated tapered carbodies for tilting clearance, and the initial production was equipped with active-tilt trucks. However, once in revenue service, the active-tilt feature was found to be too complicated and costly to maintain relative to the perceived improvement in passenger ride quality and the actual increase in average speed, and the feature was abandoned.

In 1987, SBB announced a sweeping and ambitious strategic development plan to shift its rolling stock and services into the next century. Termed the Bahn 2000 project, this initiative includes large-scale equipment renewals and service improvements, most notably introduction of high-speed (220 km/h; 137 mph) intercity passenger services. As part of this program, which was approved by Swiss voters in late 1987, SBB plans to acquire 230 km/h (144 mph) electric locomotives with radial steering, and several hundred light-weight low-axle-load passenger coaches, dubbed the IC 2000. **Figure C.42** illustrates a possible layout for the IC 2000 first-class coach.

Given the inherent alignment constraints faced by SBB and the emphasis on equipment renewal in the Bahn 2000 program, design features that improve curve negotiation while maintaining or improving passenger safety, perceived comfort, and life-cycle costs clearly will have considerable appeal.

The Swiss Industrial Association (SIG) has undertaken development of a new truck design to address just this aspect of the Bahn 2000 procurement. The SIG product differs from the tilt

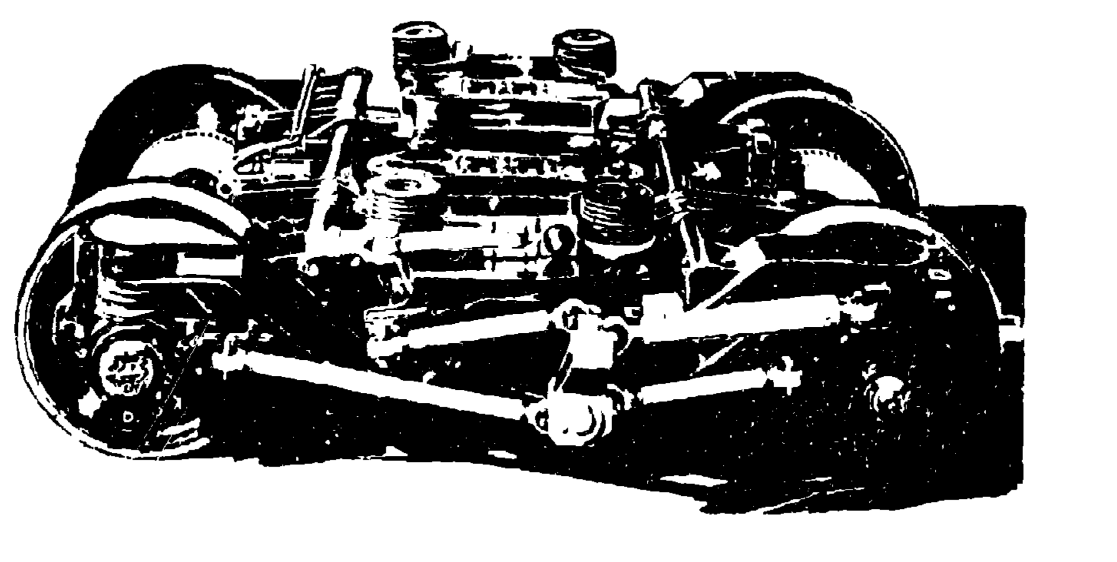


(Source: Railway Gazette International, December 1987)

Figure C.42: IC 2000 Coach Layout

technologies described in other sections of the Appendix primarily in beirg a subsystem (truck) rather than a complete vehicle (e.g., the LRC car) or trainset (e.g., the ETR-450 EMU).

The SIG truck, shown in **Figure C.43**, incorporates both a limited passive-tilt mechanism, termed "Neiko" from the German term for "roll compensator" and forced radial steering, labelled "Navigator."



(Source: Swiss Railway Revue)

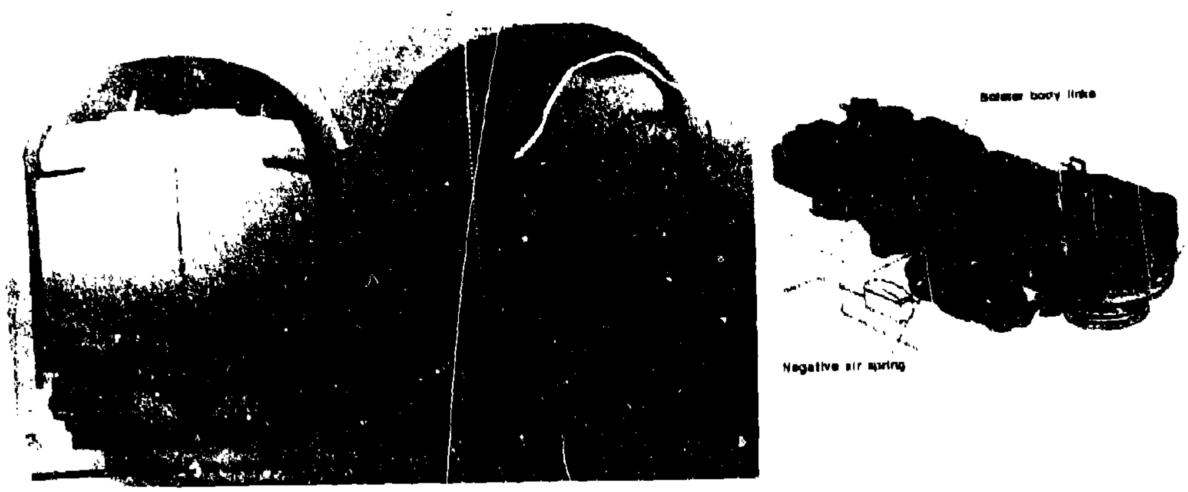
Figure C.43: The SIG Truck with Neiko and Navigator

The SIG truck with Neiko and Navigator will be extensively tested on three modified SBB Mk IV coaches, with the possibility that it will be adopted as the standard truck for the planned IC 2000 coach procurement. Trials began in late June 1991.

A subsystem-level product like the SIG truck, while lacking some of the instinctive appeal that comes with a streamlined bodyshell, has other advantages from an American viewpoint, most notably the potential to form part of a major retrofit package for existing Amtrak equipment (although clearly there could be technical and institutional issues to be resolved before feasibility is established).

C.3.2.2 Tilt Mechanism

The SIG tilting mechanism (or roll compensator) is shown in **Figure C.44**. This mechanism is based on four inclined links that control the roll angle of the bolster beam relative to the frame of the truck. These links create an effective roll center for the vehicle that lies above its center of gravity, thereby permitting pendular tilting during curve negotiation. The links are configured so that the vehicle has freedom to move vertically on its air-spring secondary suspension, but the bolster beam is forced to move in the opposite direction to the truck frame whenever there is lateral displacement due to curving.



(Source: Railway Gazette International, January 1991)

Figure C.44: Neiko Tilt Mechanism

This arrangement provides a maximum effective tilt angle of about three degrees: 1.2 degrees of inward tilt due to the action of the inclined links, and up to an additional two degrees from elimination of differential suspension compression.

This aspect of the SIG truck is not new: essentially the same sort of arrangement, referred to as the "swing-hanger" truck, was the state-of-the-art in passenger truck design in the U.S., about 1950. This design, which was marketed by a number of manufacturers including General Steel Industries, provided a relatively soft lateral ride by virtue of the pendular secondary suspension. Some versions of the design incorporated a swing link that caused the carbody to bank slightly inward on curves. The effect was that of a passive-tilt system with a very limited tilt angle. While these cars did provide a superior ride under their design conditions, they were not well suited for operation at elevated cant deficiencies, as the pendular motion resulted in a lateral shift of the c.g., and an increase in weight transfer to the outside (high) rail. A few cars equipped with these trucks are still operated by Amtrak as part of its Heritage Fleet.

The innovative feature of Neiko is its inclusion of a lateral, bi-directional air spring mounted in the center of the truck frame. This air spring acts on a vertical member projecting downward from the underside of the bolster beam. Whenever the bolster begins to move laterally outward in response to curve entry, this spring acts to reinforce the outward motion. The spring helps to overcome the inertia of the links and carbody, so as to reduce the lag between curve entry and carbody response that is characteristic of passive-tilting mechanisms. This feature provides much better tracking of curve entry and exit that is possible with an unassisted passive mechanism, while avoiding the complexity and cost associated with active-tilt systems.

However, Neiko does have limitations. The relatively small effective tilt angle means that full compensation for lateral acceleration due to cant deficiency will typically not be achievable, especially at relatively high speed. On the other hand, three degrees of tilt will normally allow a 10% to 20% increase in curving speed at the same level of perceived passenger comfort.

SIG asserts that during earlier running trials with Neiko-equipped coaches, the number of passengers stating that they were dissatisfied with curving peformance decreased from 29% to 14% for standees, and from 8% to 3% for seated riders.[19] The results of the SBB trials may offer greater insight into the potential of this feature.

C.3.2.3 Forced Radial Steering

Figure C.45 shows the Navigator forced radial steering mechanism installed on the SIG truck. This feature is designed to improve the balance between the tight control of truck geometry required to minimize wheelset yaw instability for safe high-speed running, and the flexibility

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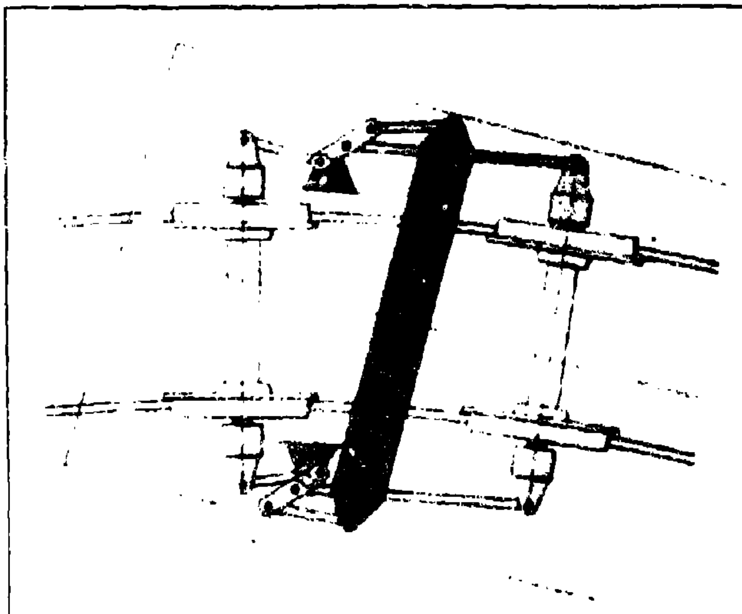
[19] "Neiko and Navigator n[illegible]v feature in IC 2000 bogies," Railway Gazette International, January 1991.

needed to produce a low angle of attack for the leading wheel flange during curving.

Although cross-braced trucks are capable of providing some benefits at freight speeds, designs in this class are prone to yaw instability at higher-speed, leading to truck hunting and increased operational risks.



(Source: Railway Gazette International, January 1991)

Figure C.45: The SIG Forced Radial Steering Mechanism

The Navigator design uses the rotational movement of the carbody relative to the truck to force the axles into an approximately radial position, by means of the links shown in **Figure C.45**. The Navigator prototype first appeared in 1986, and an improved version reached 283 km/h (177 mph) during tests on DB during 1988.

SIG claims that the Navigator feature reduces lateral forces by about 35%, and that this, in combination with a lower flange angle of attack, will result in a five-fold increase in wheel life. If true, the potential cost savings would be significant, as would the reduction in risk exposure due to excessive track forces.

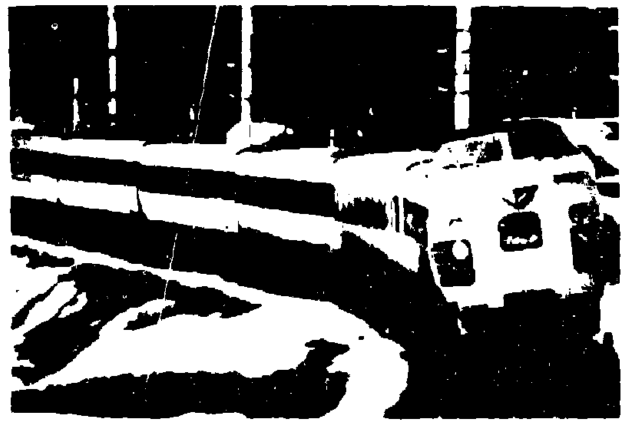
C.3.3 JAPAN RAILWAYS SERIES 381 EMU[b]

C.3.3.1 Background

The JR Series 381 passive-tilt EMU, shown in **Figure C.46**, was developed in an attempt to maintain and improve the competitive position of the then-JNR narrow-gauge services on lines serving the central mountains of Honshu.

The equipment was based on the JNR Kuhoma Series 591 passive-tilt prototype EMU, which also featured articulated trucks, and the Series 391 gas-turbine-powered passive-tilt prototype. The Series 381 trainsets were introduced into fleet service in 1973 in an attempt to increase maximum speed (to 130 km/h from 120 km/h) and especially to raise allowable speed in curves by 20 to 25 km/h (i.e., to 85 km/h from 65 km/h, for a 300m radius curve) on the Nagoya-Nagano line.

The revenue service experience of JNR, and its successor company JR-Shikoku, with this equipment on severely curving narrow gauge track through mountainous terrain is very informative with respect to ride comfort considerations in general, and to tilt motion sickness in particular.



(Source: Japanese Railway Engineering, December 1986)

Figure C.46: The Series 381 Passive-Tilt EMU

Neither of the original objectives were realized. Maximum speed could not be increased due to braking and signal deficiencies, and persistent and widespread tilt motion sickness led JNR first to issue motion-sickness medication to all passengers, and to restrict curving speeds to the levels used for conventional equipment. Ultimately, JNR was forced to develop a retrofit package incorporating a pneumatic servo-mechanism to provide a form of active tilt control[20].

The passive-tilting trainset operating experience of the JNR and its successors is quite extensive; only that of RENFE in Spain is similar. In 1984, there were 277 passive-tilting cars in the Series 381 EMU fleet. This equipment is still in revenue service but with the advent of the TSE-2000 active-tilt DMU and active-tilt retrofits for the Series 381, it represents a dead end in terms of U.S. applicability.

C.3.3.2 Technical Specifications

Table C.12 summarizes the major technical specifications of the Series 381 passive-tilting EMU.

C.3.3.3 Tilt Subsystem

The Japanese Railways Series 381 passive-tilting EMUs employ a unique mechanism to achieve carbody roll rotation, as shown in **Figure C.47**. These trainsets incorporate truck-mounted rollers which support a cross-beam mounted on the carbody. Portions of the lower surface of the cross-beam, which rests on the rollers, takes the form of a constant-radius curve.

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[20] Op. Cit., Railway Gazette International, April 1985. See also "Speedup on JNR 1067mm Gauge Lines," Y. Yukawa, Japanese Railway Engineering, Vol. 24 No. 2, 1984.

Table C.12
Major Features of Series 381 Passive-Tilting EMU

Vehicle Construction	Aluminium
Seating	76 (Intermediate cars), 60 (End cab cars)
Maximum Speed	130 km/h (81 mph) design, 120 km/h (75 mph) in service
Propulsion	Power from overhead catenary, one pantograph per two-car traction unit; d.c. traction motors, four axles driven in each traction unit.
Revenue Consist	Multiples of two-car traction units with driver cab cars at each end
Truck	Conventional low-speed truck modified to accumodate tilt mechanism
Tilt Mechanism	Curved bolster on truck-mounted rollers
Tilt Performance maximum tilt angle	5 degrees; inertial tilt rate
Maximum Lateral Acceleration	0.85 m/sec^2 or 0.087*g* at truck with zero uncompensated lateral acceleration in cabin during steady-state curving; 1.65 m/sec^2 or 1.68*g* at truck with 0.08*g* uncompensated lateral acceleration in cabin
Tilting Center Height	2.30 m (7'6"); tilt center must be above carbody c.g. for responsive passive tilting

C.4 OTHER TILT DEVELOPMENTS

C.4.1 The EM 600 High-Speed EMU[21]

There is ongoing development of the 300 km/h (187 mph) advanced-concept EM 600 tilting carbody electric trainset, by Eurotren Monoviga SA of Madrid, in collaboration with Spanish National Railways. A prototype trainset designated EM 403 has undergone extensive testing. The EM 600 trainset design includes the option of active tilting based on mercury-level tilt sensor control of pneumatic valves to achieve differential displacement of the secondary suspension air springs so as to tilt the carbody. The trainset design incorporates very light weight articulated carbodies supported by trucks each carrying four independently suspended and rotating wheels.

The yaw angle of each wheel can be independently controlled to enhance curving performance and reduce wheel and rail wear and costs.

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[21] "EM 600 Eurotrain Technology in Development for RENFE," J.P. Silva, Paper C396/003, pp 147-154, Proceedings IMechE International Conference; Transit 2020: Planning, Financing, Design and Operation of Railways Worldwide, London, England, Oct.1990.

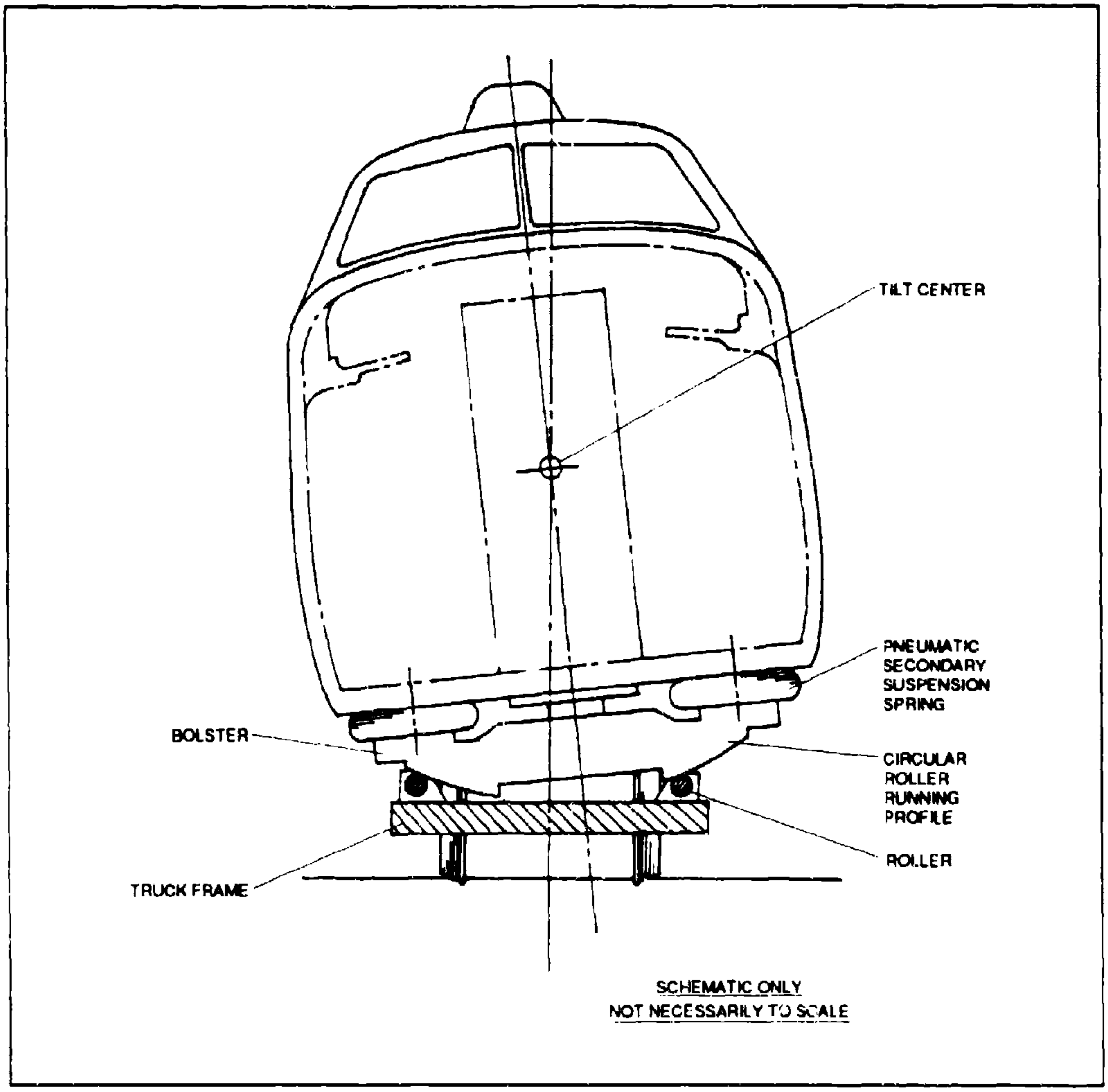


Figure C.47: The Series 381 Passive-Tilting Mechanism

C.4.2 EB Strommens Vaerksted Type B7 Coach and Class 70 EMU[22]

Beginning in the early 1980s, Norwegian State Railways (NSB) has explored the possible application of active- or passive-tilting on its aluminum-bodied Type 7 coaches and latterly on

[22] Information drawn from "Tilting Back in Favour," Railway Gazette International, May 1991; Jane's World Railways, Railway Systems/Norway, Jane's Information Group Limited, Surrey, U.K., 1990-91; "NSB set to introduce Class 70 EMUs," Railway Gazette International, May 1990; and "Norwegians Press On With Lightweight Rolling Stock," Railway Gazette International, June 1984.

the Class 70 EMU sets. Two prototype high-speed coaches built by EB Strommens were fitted with an experimental active-tilt system and underwent extensive testing in 1984-85, in part by NSB and also by British Rail on behalf of NSB.

The active-tilting system, illustrated in **Figure C.48** provided a seven degree maximum tilt angle and a relatively rapid tilt rate of seven degrees/sec to meet demanding transition curve conditions. Tilt control was by mean of two lateral accelerometers and a gyroscope mounted in the locomotive.

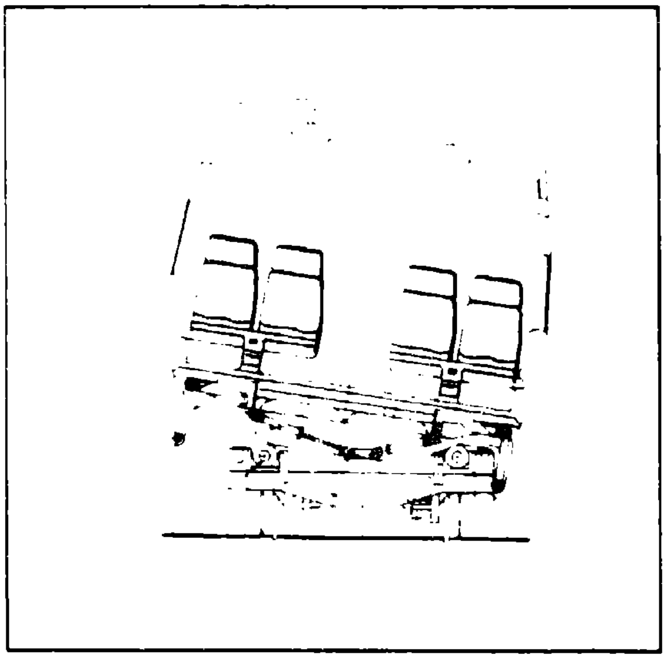
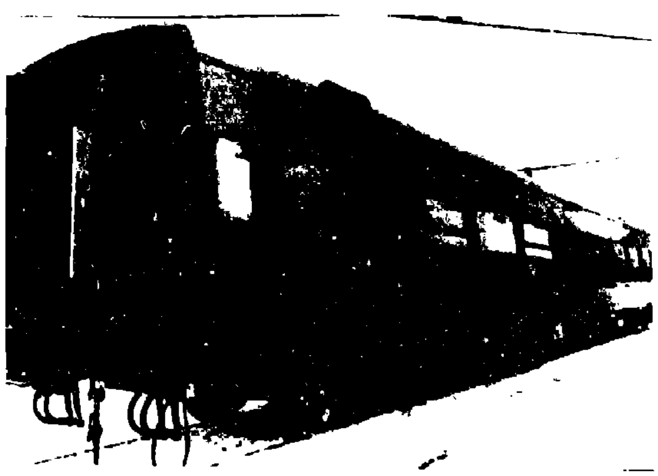


Figure C.48: Type 7 Tilt Mechanism

Interestingly, although not used with operational tilting, the production version of the B7 coach **(Figure C.49)** manufactured after March 1986 incorporates the tilt mechanism on its trucks (**Figure C.50**). NSB is reportedly reviewing the feasibility of tilt operation in view of the success of SJ with the X2000. All the T7 fleet has a cross-section profile that would permit tilt operation within the existing NSB clearance envelope.

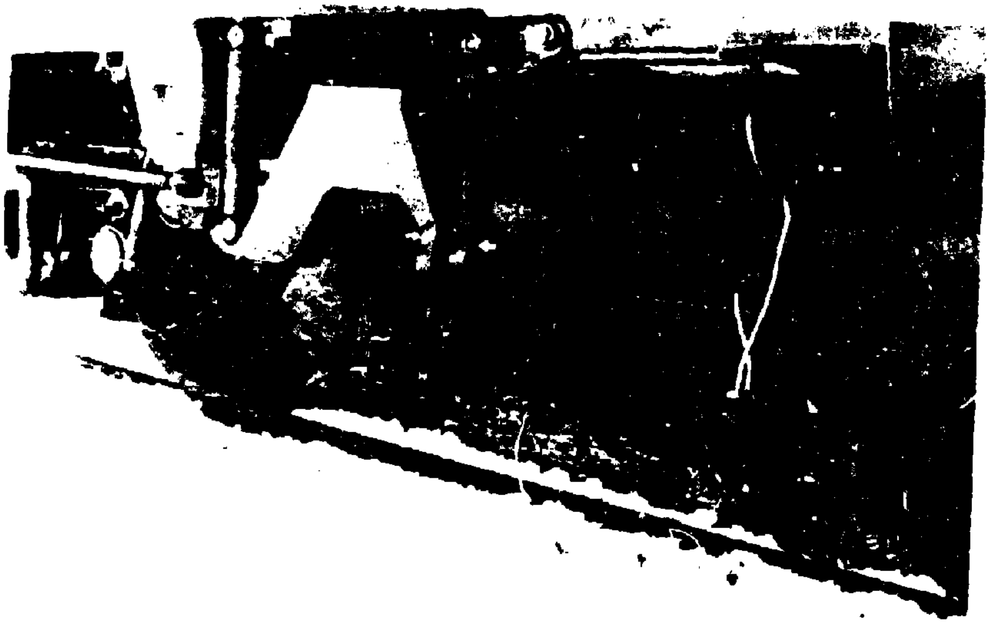
NSB is also taking delivery of nine 4-car Class 70 EMUs, again from EB Strommen, that are tilt-compatible; the body section will permit up to two degrees of passive tilt, although no tilting mechanism is included in the production version.



(Source: Railway Gazette International, May 1986)

Figure C.49: Type 7 Coach

NSB has also announced planning for the development of an advanced concept 200 km/h (125 mph) tilt-body trainset to replace intercity rolling stock in mainline operation by 2004. However, no details have become available.



(Source: Railway Gazette International, May 1986)

Figure C.50: Type 7 Truck With Tilt Mechanism

ENDNOTES FOR APPENDIX C

a. Information about the LRC was obtained from the following sources, plus any others cited as footnotes:

1. Jane's World Railways, Railway Systems/Canada, Jane's Information Group Limited, Surrey, U.K., 1980-81, 1982-83 and 1985-86.

2. Improved Passenger Equipment Evaluation Program - Train System Review Report, Volume 8, LRC (Canada), U.S. Department of Transportation, Federal Railway Administration Report No. 80/14.VIII, March 1979.

3. The LRC Coach Trucks and Suspension, W.H. Elmaraghy, J.A. Gaiser and H.J. Bexon, ASME Paper No. 79-RT-4, 1979 (paper presented at Joint ASME/IEEE Railroad Conference, Colorado Springs, Colorado, April 1979.

4. LRC and HST Comparison, J.D. Young, CIGGT Newsletter V.9, No. 1 August 1981.

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b. Information about the ETR-450 was obtained from the following sources, plus any others cited as footnoted in the text:

1. "Tilting Body Systems and the Fiat-Pendolino," P. Losa and A. Elia, pp 41-45, Railway Technology International, 1990.

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6. Improved Passenger Equipment Evaluation Program - Train System [illegible] Report, Volume 4, ETR-401, U.S. Department of Transportation, Federal Railway Administration Report No. 80/14.IV, March 1978.

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10. "Tilting Back in Favour," Railway Gazette International, May [illegible]1.

c. Information about the VT 610 was obtained from the following [illegible]s plus any others cited as footnotes in the text:

1. Jane's World Railways, Railway Systems/Germany (Federal Republic), Jane's Information Group Limited, Surrey, U.K., 1989-90 and 1990-91.

2. Tilting Trains for Bavaria, V. Kottenhahn, Proceedings of the Tilting Body Trains Seminar, London, England, Dec. 1989, The Institute of Mechanical Engineers, Railway Division, London.

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Information about the Class 4012 EMU was obtained from:

4. "The New Train-Set of the OBB with Carriage Body [illegible] [illegible]ail Technology International, 1991.

d. Information about the X2000 was obtained from the following sources, plus any other cited as footnotes in the text:

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1. Jane's World Railways, Railway Systems/Spain, Jane's Information Group Limited, Surrey, U.K., 1978, 1982-83, 1985-86, 1989-90 and 1990-91.

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1. "SBB Poised To Order Bahn 2000 Stock," pp 826-827, Railway Gazette International, December 1987.

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