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IMPROVED PASSENGER EQUIPMENT EVALUATION PROGRAM

TECHNOLOGY REVIEW

WHEELS, AXLES, COUPLERS

Unified Industries Incorporated
5400 Cherokee Avenue
Alexandria, Virginia 22312

March 79

~~MARCH~~ 1978

FINAL REPORT

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16. Abstract

The status of two foreign rail technologies is analyzed in this report. The two technologies are Wheels and Automatic Couplers.

The wheel development program for the French TGV is reviewed. It illustrates the application of classical wheel design to modern high-speed rail transport. The resilient wheel and its reduction of effective unsprung mass is studied; the SAB resilient wheel application on British Rail is reviewed. Lightweight axles are also discussed.

The review of automatic coupler technology is primarily concerned with the latest coupler designs from both the mechanical and electrical train-line concepts. The evolution and development of advanced couplers in the United States and Europe are covered.

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METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

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

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

Approximate Conversions from Metric Measures

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

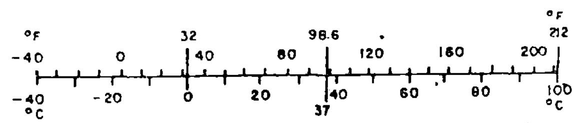

TABLE OF CONTENTS

Section No.		Page No.
1	WHEELSETS	1
	INTRODUCTION	1
	WHEELSETS FOR THE FRENCH TGV HIGH SPEED TRAIN	2
	THE TGV TRAINSET	2
	THE TGV WHEELSETS	3
	THE SCNF WHEEL	4
	THE TGV WHEEL	6
	THE TGV AXLE	13
	ACKNOWLEDGMENT	14
2	RESILIENT WHEELS	15
	INTRODUCTION	15
	SUMMARY OF THEORY	15
	THE SAB RESILIENT WHEEL	20
	THE SAB DESIGN	20
	BRITISH RAIL TESTS	22
	THE NORWEGIAN STUDY	26
	LATERAL FORCES	26
	TRACTION MOTOR BENEFIT	27
	SAFETY AND RIDE QUALITY	28
	CREEP EFFECTS	28
	TEMPERATURE EFFECTS DURING BRAKING AND SUSTAINED HIGH SPEED	28
	MAINTENANCE	29
	NOISE REDUCTION	29
	ACKNOWLEDGMENTS	29
3	LIGHTWEIGHT AXLES	30
	INTRODUCTION	30
	TORSIONAL FLEXIBILITY	30
	EFFECT ON TRUCK STABILITY	31
4	THE AUTOMATIC COUPLER	35
	INTRODUCTION	35
	DEFINITION OF AUTOMATIC COUPLING	35
	SCOPE OF REVIEW	35
	SELECTION OF COUPLER DESIGNS	36
5	DEVELOPMENT OF THE U.S. COUPLER	37
	EARLY TRAIN COUPLERS	37
	MODERN AMERICAN PASSENGER CAR COUPLERS	41
	DEVELOPMENT OF TRAIN LINE CONNECTIONS	42
	FULLY AUTOMATIC COUPLERS	45
6	THE NEW UIC (EUROPEAN) AUTOMATIC COUPLER	52
7	AUTOMATIC HIGH VOLTAGE COUPLER	57

LIST OF FIGURES

Figure No.		Page No.
1	OUTLINE FOR TGV TRAINSET	3
2	THE EARLIER SNCF WHEEL	4
3	THE MODERN SNCF WHEEL	5
4	EFFECT OF WHEEL SHAPE ON THERMAL DEFORMATION	7
5	TGV WHEEL PROFILE	7
6	SNCF TEST RESULTS FOR STEELS OF TABLE 1	10
7	TGV WHEELSET	13
8	TGV WHEEL-AXLE ASSEMBLY HUB OVERHANG	14
9	WHEEL/RAIL FORCE SIMULATION MODEL	16
10	TYPICAL FORCE/TIME HISTORY AT DIPPED RAIL JOINT	17
11	$\frac{P2-P0}{2\alpha V}$ VERSUS EQUIVALENT UNSPRUNG WEIGHT	19
12	THE SAB RESILIENT WHEEL	21
13	TRACK FORCE/SPEED CURVES MEASURED AT INSTRUMENTED RAIL JOINT	23
14	MAXIMUM STRESS AT FIRST BOLTHOLE OF FISHPLATE	24
15	PROBABILITY DENSITY DISTRIBUTION OF VERTICAL WHEEL ACCELERATION ON CONTINUOUS WELDED TRACK	25
16	VERTICAL WHEELSET ACCELERATIONS AT SWITCHES AND CROSSING	25
17	LATERAL WHEELSET ACCELERATIONS ON STRAIGHT TRACK-COMPARISONS OBTAINED WITH A BR CLASS 82/2 LOCOMOTIVE FITTED WITH SAB RESILIENT WHEELS AND PLAIN WHEELS	27
18	CREEP-TIME CURVE FOR A TYPICAL SAB RESILIENT WHEEL AT AN AMBIENT OF 18 DEGREES C (64.4 DEGREES F)	28
19	THE EFFECT OF WHEEL CONICITY ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	33
20	THE EFFECT OF PRIMARY LONGITUDINAL STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	33
21	THE EFFECT OF CREEP COEFFICIENT ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	33
22	THE EFFECT OF SECONDARY YAW STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	33
23	THE EFFECT OF PRIMARY LATERAL STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	34
24	THE EFFECT OF SECONDARY LATERAL STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	34
25	THE EFFECT OF PRIMARY LATERAL DAMPING ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	34
26	THE EFFECT OF SECONDARY LATERAL DAMPING ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	34
27	THE EFFECT OF WHEEL RADIUS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS	34
28	LINK AND PIN (1831)	37
29	MILLER DRAW HOOK (1863)	38
30	JANNEY COUPLER (1873)	38
31	JANNEY CLASPED HAND PRINCIPLE	39
32	JANNEY SWINGING KNUCKLE DESIGN	39
33	JANNEY TYPE D (1916)	39

LIST OF FIGURES (continued)

Figure No.		Page No.
34	WILLISON INDUSTRIAL DESIGN (1916)	40
35	WILLISON SPREAD CLAW PRINCIPLE	40
36	AAR STANDARD H TIGHTLOCK (1947)	41
37	ROBINSON WING TYPE PASSENGER CAR CONNECTOR FOR STEAM AND TWO AIR CONNECTORS	43
38	ROBINSON PIN AND FUNNEL PASSENGER CAR CONNECTOR	43
39	TRANSIT CAR COUPLER	46
40	ELECTRO-PNEUMATIC SWITCHING UNIT	47
41	ELECTRIC COUPLER BOX	47
42	SYMINGTON-WAYNE COUPLER OPERATION	48
43	OHIO BRASS TRANSIT CAR COUPLER	49
44	WABCO N-2-A RAPID TRANSIT AUTOMATIC CAR COUPLER	50
45	THE EUROCOUPLER CONCEPT	52
46	UIC AUTOMATIC COUPLER WITH CROSSBEAM SUSPENSION	53
47	UIC AUTOMATIC COUPLER WITH TELESCOPIC LEG SUSPENSION	54
48	UIC COUPLING RANGE	55
49	AUTOMATIC HIGH VOLTAGE COUPLER-GERMAN ET-403	58

LIST OF TABLES

Table No.		Page No.
1	CANDIDATE STEELS FOR TGV WHEEL	9
2	TGV & RTG TRAIN DATA	9
3	SNCF MATERIAL SPECIFICATION FOR TYPE R7 WHEEL	11
4	COUPLER MECHANICAL CHARACTERISTICS	44
5	COUPLER AIR LINE CHARACTERISTICS	44
6	U.S. RAILROAD APPLICATION OF AUTOMATIC COUPLERS IN NEC AND CONNECTING ROUTES	51

SECTION 1. WHEELSETS

INTRODUCTION

The development of wheelset technology in foreign countries is of interest with respect to two aspects: the achievement of higher speeds, and the reduction of unsprung weight. These topics are addressed in the interest of reporting technology relevant to high-speed passenger trains.

The railroad wheel is generally a simple device, both in its method of manufacture and in its implementation. It is manufactured by either casting or forging, both of which are basic industrial processes. In conventional practice, it is used as a "wheelset," a rigid two wheel and axle combination with the wheels pressed onto the axle. This arrangement prevents differential rotation between the wheels during curving, and provides guidance (depending upon the conicity of the wheels) to position the wheelset relative to the rails. The wheelset is also the main load bearing component of a car or locomotive, and as such its structural integrity is a primary consideration.

The natural simplicity and efficiency of the steel wheel/rail combination has helped to carry it forward into the regime of high-speed ground transport. It has been pointed out[1] that when the new high-speed train (Tres Grande Vitesse, or TGV) was first being thought about in France, much consideration was given to more sophisticated (i.e., magnetic levitation, air-cushion) vehicle/support systems. Research, however, proved that such a change was not economically justifiable, and today the 270 km/h TGV, currently being road tested, rolls on flanged steel wheels over conventional rails.

The continuing research that has made this possible centers on improving the strength and durability of wheels, axles, and bearings. Advanced analytic methods such as finite element structural and thermal analysis are being utilized to quantify the response of the wheel to known inputs. Fracture mechanics is being used to predict catastrophic failures from crack growth. Classified metallurgy continues to assess and improve the microstructural response of wheel materials to the environment imposed upon them.

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[1]Andre Portefaix, "The Interface Between Wheel and Rail," Revue Generale des Chemins de Fer, special issue (publication date unknown) entitled The TGV And The Paris--South-East Line, containing English translations of the main articles from the November and December 1976 issues.

WHEELSETS FOR THE FRENCH TGV HIGH-SPEED TRAIN

Portefaix[2] has noted that the very high-speed train (Tres Grande Vitesse, or TGV) was not preordained to be a traditional rail vehicle; it simply turned out to be the most practical approach to achieving high speed ground transportation. The reasons for this were several:

a. If the physical wheel-on-rail contact were eliminated (in favor of magnetic levitation, for example), then eliminated along with it would be the current methods of propulsion, braking, signaling, and electric power collection. Replacement technologies for all of these would be required.

b. A monorail type structure was indicated as the best support and guidance method for the levitation systems. This in turn creates great difficulties in switching to another track, because the monorail switch requires a large displacement of the running support beam. The moving parts of the traditional railroad switch need to be displaced only a few inches to allow the wheel flange to pass.

c. For the speeds considered (i.e., up to 186 mi/h), conventional rail is more energy efficient than the more sophisticated vehicle support systems. For example, the levitation power required for an air cushion train is 30 to 40 percent of the total power requirement, while the mechanical resistance (i.e., nonaerodynamic) of a high-speed conventional train is only a few percent of the total resistance.

d. Even before the energy crisis, 186 mi/h was deemed an economic upper limit on speed. Since the TGV-001, a gas turbine powered experimental prototype for the electric TGV, had shown that 186 mi/h was within the capacity of the wheel/rail combination, there was no justifiable reason to invent a new land transport method.

THE TGV TRAINSET

According to Revillon,[3] the French National Railways (SNCF) plans to place 87 TGV trainsets in operation at 168 mi/h on the new line from Paris to Lyon between 1981 and 1983. The rolling stock must meet very demanding requirements. Safety must be guaranteed at very high speeds along with economy and high performance. Research and testing has shown that a requirement for this is a moderate axle load (16 metric tons), to insure vehicle stability and maintenance of track geometry.

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[2]Portefaix, ibid.

[3]A. Revillon, "Wheelsets for SNCF High-Speed Trainsets," Proceedings of the Sixth International Wheelset Congress, Colorado Springs, Colorado, USA, October 1978.

Figure 1 shows an outline drawing of the TGV trainset. It consists of two electric locomotives, one at each end of the trainset, and an articulated section consisting of 8 trailers mounted on 9 trucks. The design permits:

a. Reduced aerodynamic drag through an improved aspect ratio.

b. Reduced mechanical drag and lower weight due to fewer trucks.

c. Better passenger comfort since no passengers are seated directly over the trucks.

d. More freedom of movement for the passengers by elimination of doors between coaches.

Three types of brake equipment are utilized on the TGV: conventional shoe-type brakes, rheostatic (dynamic) brakes, and disc brakes. In the interest of safety, no significant excursions into new technology were made in the braking equipment; only familiar, tested systems or their derivatives were used. All axles are equipped with electronic skid control.

The TGV is one of eight trainsets specifically studied under the Improved Passenger Equipment Evaulation Program (IPEEP) and is reported upon in a separate report.[4]

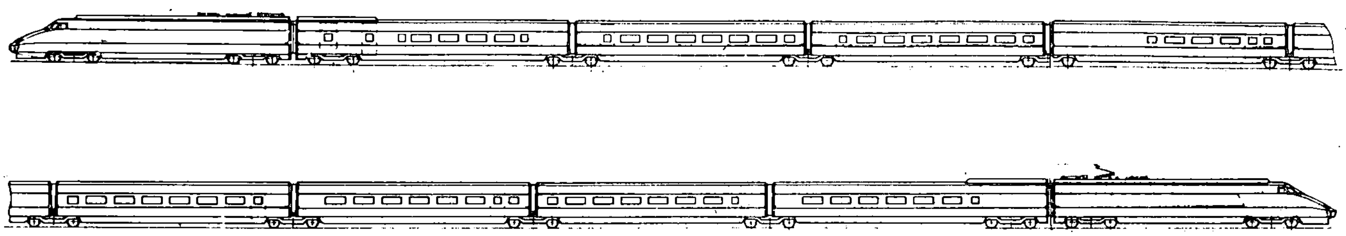

FIGURE 1. OUTLINE OF TGV TRAINSET

THE TGV WHEELSETS

The TGV wheel design is firmly rooted in the basic SNCF concept. The basic demands placed upon the TGV wheels are not significantly greater than those on other SNCF wheels, although the speed is higher, the axle loading is lower; as a result the Hertzian stresses in the running surface and the cyclic stresses in the web are lower than in conventional SNCF wheels.

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[4]IPEEP Train System Review Report, Volume 6 TGV-PSE (France).

THE SNCF WHEEL

The SNCF wheel of today, according to Portefaix,[5] is the result of more than ten years of research and development, and has evolved significantly from the design of earlier years, as shown in figure 2.

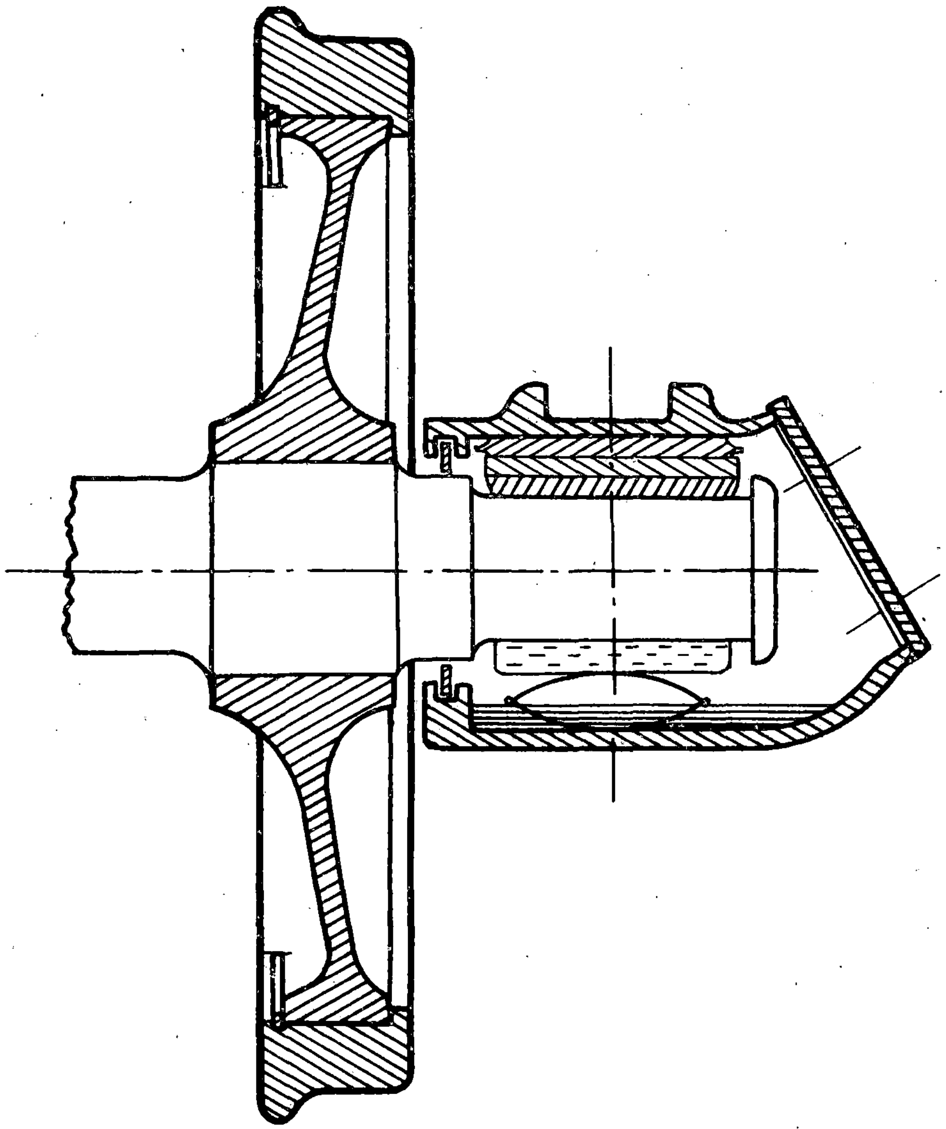

FIGURE 2. THE EARLIER SNCF WHEEL

[5]Portefaix, op. cit.

The modern SNCF wheel (shown in figure 3) is of monobloc construction for reasons of weight reduction, heat diffusion, and the elimination of the risk of loosening shrunk-on tires under traction conditions. The planes of symmetry of the hub, the web, and the rim merge in order to reduce the risk of buckling. The hub extends beyond the wheel seat for better resistance of the latter to fatigue and contact corrosion; the diameter of the wheel seat itself exceeds that of the body of the axle by about 15 percent in order to resist "shrinking-on" stresses.

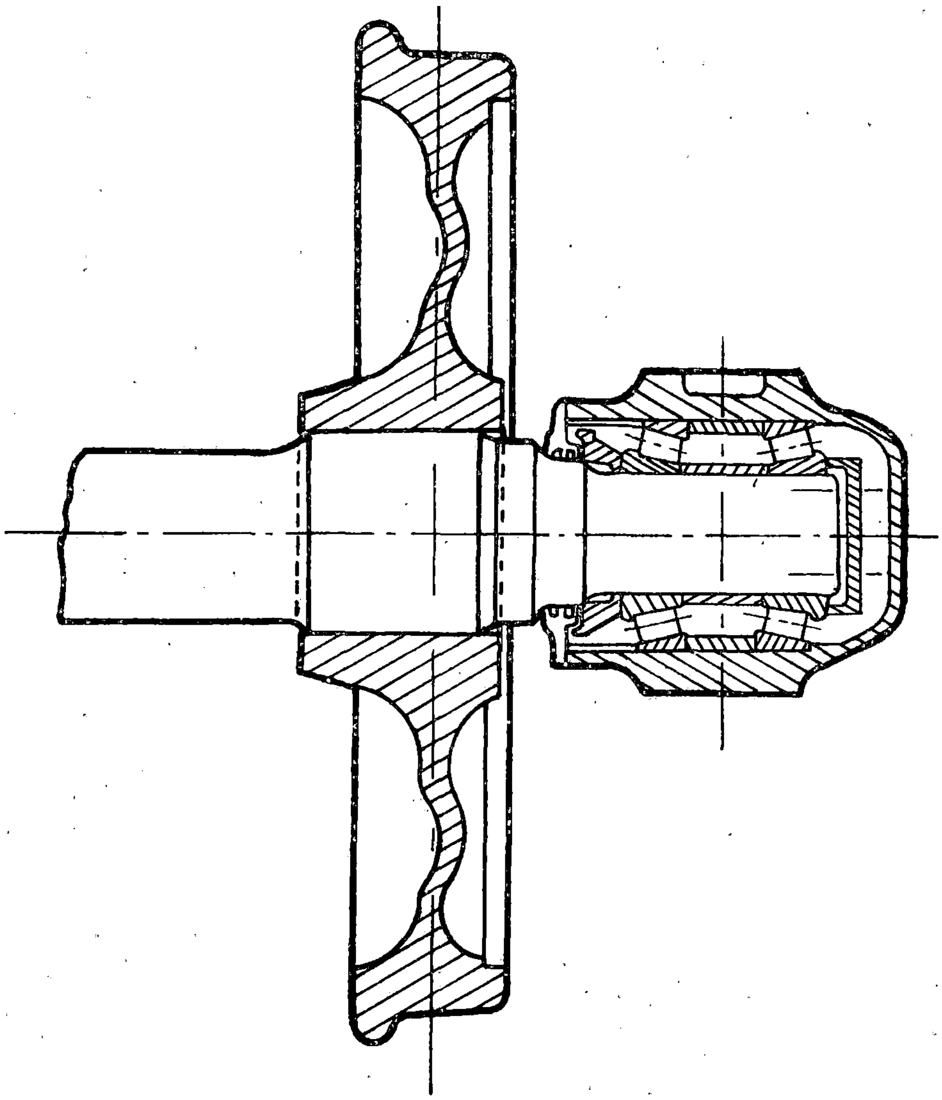

FIGURE 3. THE MODERN SCNF WHEEL

The wheel is made of non-alloy moderate carbon steel (0.45 to 0.55 percent) with 0.65 to 0.80 percent manganese. It is carefully manufactured, being forged, rolled, and finally machined on all surfaces. It is ultrasonically tested and surface heat treated (to a depth of 35mm) by soaking at 825°C and letting down to 500°C. The heat treatment increases the hardness (to 250 Brinell) reducing wear, and increasing rim tensile strength (780 to 860 MPa); it also prestresses the running surface in compression (150 to 180 MPa) reducing the risk of shelling due to Hertzian stresses and the risk of thermal cracks from braking.

THE TGV WHEEL

It is clear that the SNCF used the TGV research program as a vehicle for the development of a better wheel for general use. Hence, the wheel for the TGV is similar to the latest wheel designs for normal service.

The specific shape, however, and the grade of steel to be used were dictated, according to Revillon,[6] by the following criteria:

a. Speed-related dynamic stresses -- at 168 mi/h the TGV wheel rotates at 1,575 rpm, creating 1,250 g's of centripetal force at the rim; the danger of wheel cracking at the center must thus be dealt with. Conventional SNCF wheels are designed for 900 rpm. Mean vertical dynamic loading also increases, but along an approximately linear characteristic with speed instead of the quadratic variance of centrifugally related stresses.

b. The desirability of minimizing tread wear to preserve dynamic stability and passenger comfort and to avoid overly frequent reprofiling.

c. Possible internal stress problems and/or fatigue cracking due to increased unsprung mass (approximately 30 percent greater than for a conventional wheelset). This is due to the presence of twin-disc brakes on each axle and a portion of the mass of the suspension on drive axles. The problem is further aggravated by the speed increase.

d. The use of cast iron brake shoes for wheel surface cleaning in order to maintain adhesion (even on dirty rail) and to eliminate small surface defects.

The wheel center profile was determined through finite element computer analysis and laboratory testing. The slightly corrugated design was found to be better than the classic straight wheel center; the corrugations allow flexure to occur such that under thermal gradients due to braking an essentially radial displacement profile is maintained. This is shown in figure 4. If the wheel center is sufficiently thick, it also stands up well to dynamic lateral loading. Figure 5 shows the wheel finally adopted.

The problem of steel grade choice was different than for conventional vehicle wheels since frictional braking is only a small percentage of the

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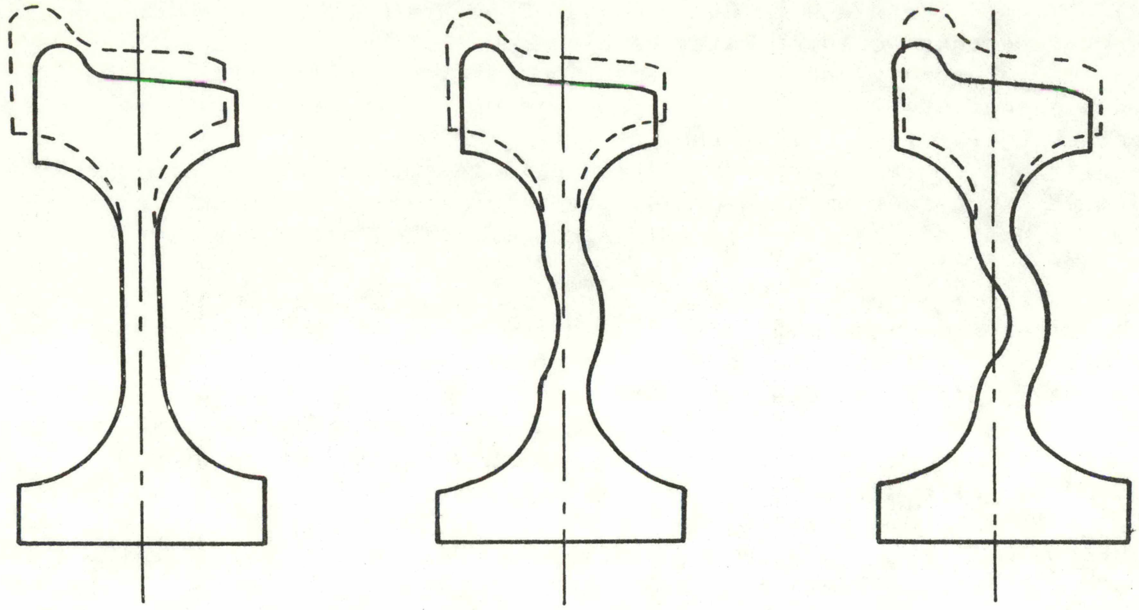
[6] Revillon, op. cit.



DEFORMATIONS SHOWN ARE FOR AN IDENTICAL THERMAL GRADIENT AND DIFFERENT WHEEL-CENTER PROFILES

FIGURE 4. EFFECT OF WHEEL SHAPE ON THERMAL DEFORMATION

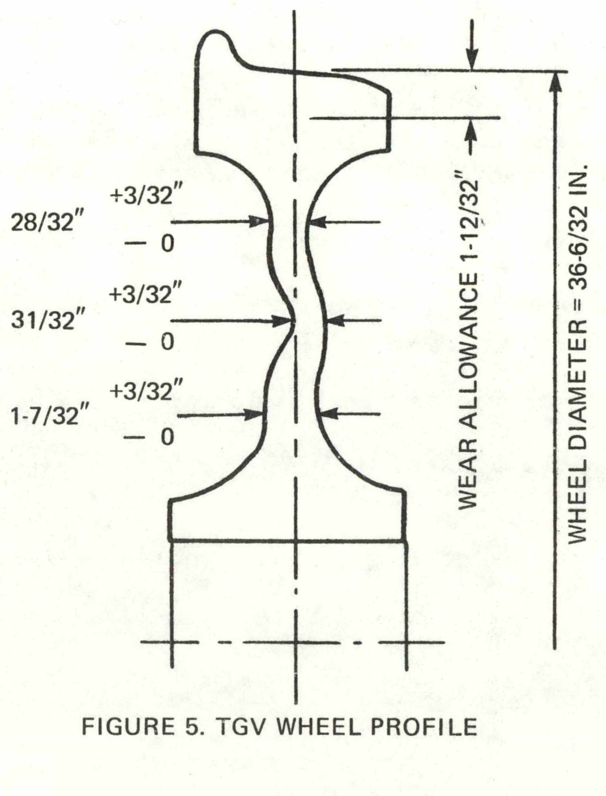


FIGURE 5. TGV WHEEL PROFILE

total. Above 124 mi/h the tread brakes are applied with very little force (5 kN); this is increased to 24 kN at 99 mi/h (124 mi/h for emergency braking). Heat cracking that would normally be expected with high carbon and manganese content is, therefore, not a problem. The one restriction here is that the brake shoes cannot protrude beyond the edge of the wheel; if they do, severe heat cracking is likely along the outer edge of wheel/rail contact. This is apparently due to the excess heat buildup in the outer edge of the wheel being alleviated by contact with the rail. Metallurgical problems due to the thermal effects of skidding are not a problem, since all axles have electronic skid control.

The steel grade was thus selected on the basis of ability to maintain tread profile. The International Union of Railways (UIC) studies have shown that a wheel/rail tensile strength ratio of 1.10:1, is economically desirable from the maintenance viewpoint. Also, a wheel harder than the rail was thought to improve adhesion. This led to a preliminary specification of 850 to 950 MPa tensile strength for the wheel steel.

Three steel grades, described in table 1, were field tested on the existing SNCF gas turbine trainsets (RTG), which have some similarity to the TGV. Table 2 shows comparative data for the RTG and TGV trainsets. It can be seen that the closest similarity is in the axle loading.

After 186,400 mi of RTG testing, the results of which are summarized in figure 6, the expected decrease of wheel wear as a result of increased steel hardness was found. However, it was also found that the harder grade of steel was less tolerant of dirty rail conditions than expected; this led to excessive slip/slide problems which in turn required frequent reprofiling of the wheels.

This latter situation should not be a problem on the TGV since all axles have electronic slip/slide control. However, it is, also, desirable to limit hardness somewhat in order that the cast iron brake shoes can perform their natural reprofiling action in removing surface defects; as discussed above, this reduced the risk of fatigue cracking and shelling of the running surface. The compromise choice thus went to steel grades R7 and R8; the specification for grade R7 is shown in table 3. Current research is directed toward use of niobium-vanadium steel to reduce thermal degradation problems while maintaining current hardness levels.

The issue of high unsprung mass was explored in the light of past SNCF experience; it was anticipated that fatigue cracks due to inclusions might develop in the body of the wheel rim. TGV trainset wheels are thus subjected to a special automatic ultrasonic inspection during manufacture which:

a. Inspects along 2 axes, using 3 MHz probes with monitored beam homogeneity.

b. Is calibrated using flat-bottomed holes, 2mm in diameter, as standard defects.

TABLE 1. CANDIDATE STEELS FOR TGV WHEEL

	C %	Mn %	Rm N/mm2	Rp N/mm2
Steel 1	0.47	0.67	840	650
Steel 2	0.50	0.71	860	690
Steel 3	0.56	0.76	950	740

TABLE 2. TGV AND RTG TRAIN DATA

		RTG	TGV
Maximum speed mi/h		99	168
Number of cars		5	10
Number of drive trucks		2	6
Number of bearing trucks		10	7
Wheel dia. (in)		33.86	36.22
Wear allowance (in)		1.58	1.38
Per-axle load (t)	D	16.4	17
	B	15.3	15
	D	one cast iron shoe plus hydraulic brake without skid preventer	two cast iron shoes plus hydraulic brake without skid preventer
Braking	B	one cast iron shoe plus two discs without skid preventer	two cast iron shoes plus two twin discs with skid preventer
D : Drive B : Bearing			

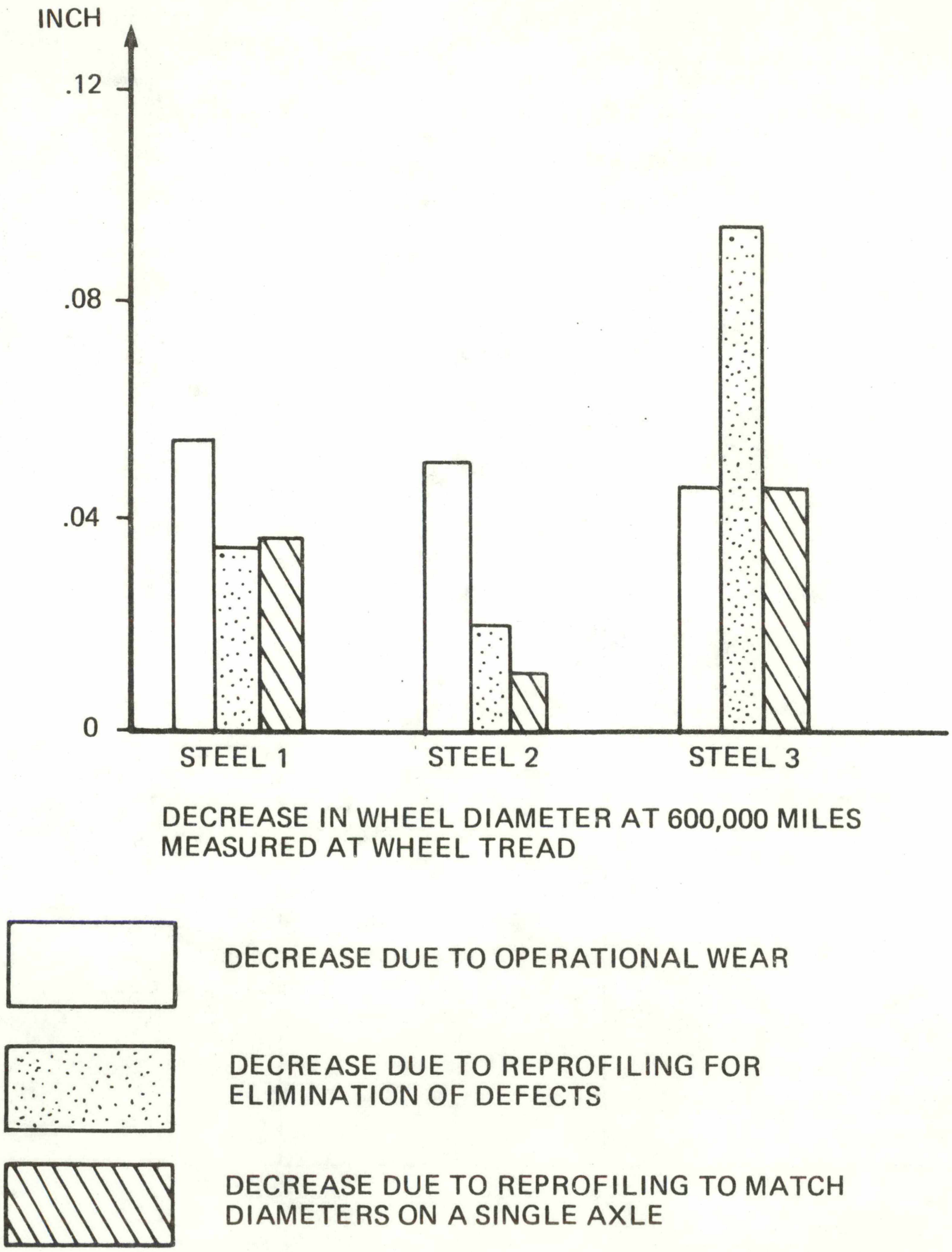


FIGURE 6. SNCF TEST RESULTS FOR STEELS OF TABLE 1

TABLE 3. SNCF MATERIAL SPECIFICATION
FOR TYPE R7 WHEEL

Chemical composition	C %	⩽ 0.52
	Mn %	⩽ 0.80
	Si %	⩽ 0.40
	P %	⩽ 0.04
	S %	⩽ 0.04
	Cr %	⩽ 0.30
	Ni %	⩽ 0.30
	Mo %	⩽ 0.05
	Cu %	⩽ 0.30
	V %	⩽ 0.05
	Cr + Mo + Ni + Cu	⩽ 0.60
Rim tensile test Specimen 15 mm from surface	Rm N/mm2	820 to 940
	A %	⩾ 14
Rim impact test U-notch	KU at +20°C in joules	⩾ 15

Revillon[7] indicates that manufacture of the TGV wheels has created no special problems.

[7]Revillon, op. cit.

THE TGV AXLE

The TGV axle with its two twin brake discs is shown in figure 7. It is designed by standard SNCF procedure,[8] updated to accommodate the 220 kg weight of the twin-discs. Wheels and discs are assembled onto the axle by means of cold press fitting, also the SNCF standard method.

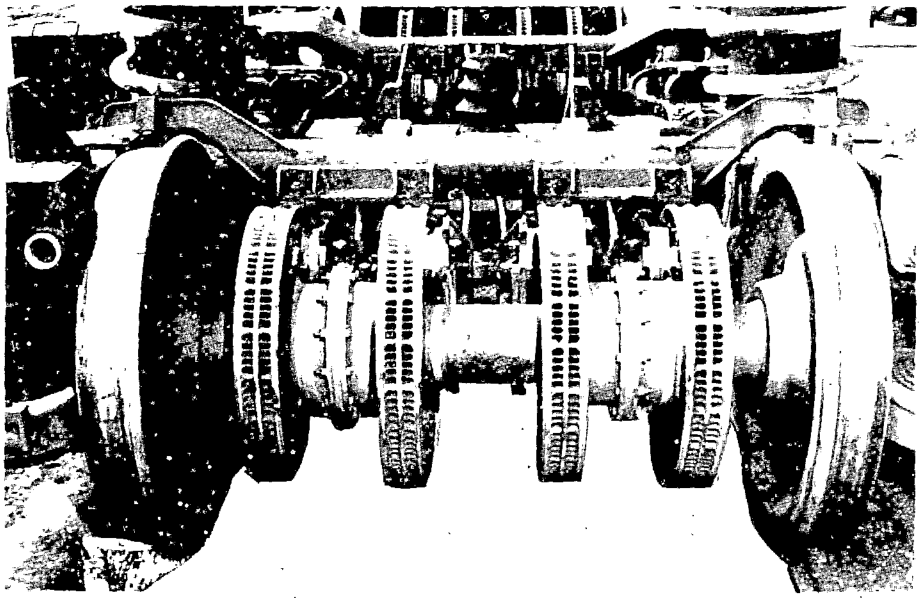


FIGURE 7. TGV WHEELSET

Cold press fitting is preferred over hot by SNCF for several reasons:

a. Fatigue behaviour is improved by work hardening of the fitting surface and formation of a slightly rounded edge in the bore entrance.

b. The operation is simple to perform because the parts are cold; for the same reason it is possible to record a graph of process conditions which permits better process control.

c. A biodegradable grease can be used to make the union; when it has decomposed the fit is additionally firm.

d. Due to the relative ease of the fitting operation as discussed above, a slight projection of the wheel hub relative to the axle journal (figure 8) is achievable; this enchances fatigue behavior.

[8]Revillon, op. cit.

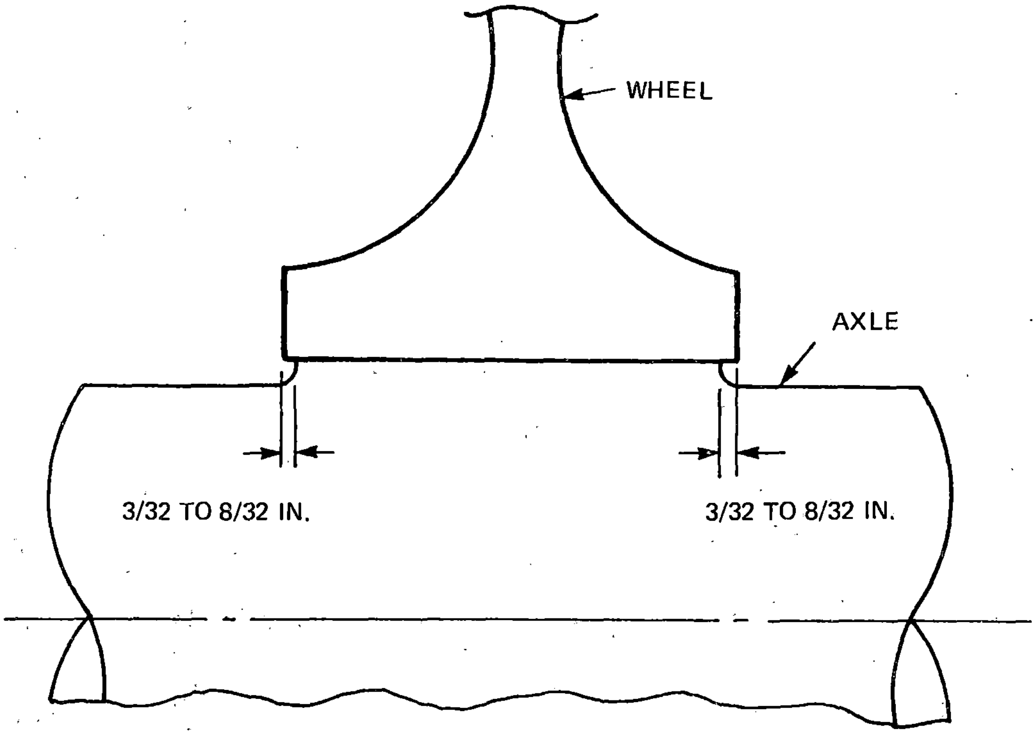


FIGURE 8. TGV WHEEL-AXLE ASSEMBLY HUB OVERHANG

Current TGV axle research involves examination of mechanical and thermal surface treatments and the use of alloy steels such as 20 NCD 6 (0.20 percent C - 1.25 percent Ni) to improve fatigue behavior.

ACKNOWLEDGEMENT

The discussion of the TGV wheelset is derived from Portefaix[9] and Revillon[10]. Mr. Portefaix, now retired, was previously Deputy Manager, Rolling Stock Directorate of the SNCF; his paper was obtained directly from the SNCF during an IPEEP visit there. Mr. Revillon was a delegate of the SNCF to the 6th International Wheelset Conference at Colorado Springs, Colorado, USA, during October 1978; an IPEEP delegation also attended that conference.

[9] Portefaix, op. cit.

[10] Revillon, op. cit.

SECTION 2. RESILIENT WHEELS

INTRODUCTION

In the current worldwide trend of improving intercity rail passenger transportation systems, the most fundamental issue is that of reducing journey time. Some of this can be accomplished by improved train scheduling. Additional help can come from higher tractive effort and stronger braking systems, reducing the acceleration/deceleration time associated with stops. The greatest journey time reduction, however, is achieved by the obvious method -- run the train faster.

As Radford[11] observes, operating modern high-powered locomotives and lightweight passenger equipment at high speeds is technically feasible from the equipment standpoint, but a safe and comfortable ride requires that the track be maintained to a correspondingly high standard of surface level and alignment. This maintenance problem is compounded by the high speed itself; as the wheels pass over irregularities on the rails, the resulting track forces increase with speed.

SUMMARY OF THEORY

When Canadian National Rail (CNR) was investigating[12] the requirements of operating higher-speed passenger service on their system, it was found that little quantitative information was available in North America on the effects of the axle load and unsprung mass of railway locomotives and cars by dynamic wheel/rail forces. It was known, however, that British Rail (BR) had done a theoretical and experimental study on the subject and that a mathematical model, shown in figure 9, had been developed for computer simulation of the phenomena. CNR, therefore, contracted with British Rail to operate the model for several existing and proposed locomotives and cars on typical CNR track.

The BR model assumes the rail to be continuously supported on an elastic foundation with the weight of the tires uniformly distributed and added to that of the rail. The stiffness of the foundation is derived from measuring values of track modulus and the damping factor is considered to be a fixed proportion of critical damping.

For the CNR project, the irregularity in the rail surface was modeled as a series of symmetrical joints (see figure 9) with dip angle $2\alpha = 0.2$ radians, the model considers a single wheelset passing over the joints with the proportion of truck and body weight carried by one wheelset and with primary and

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[11] R.W. Radford, "Wheel/Rail Vertical Forces in High-Speed Railway Operation," Transactions of the American Society of Mechanical Engineers, Journal of Engineering for Industry; ASME Paper Number 77-RT-1, presented at ASME-IEEE Railroad Conference, Washington, D.C., March 30 - April 1, 1977.

[12] Ibid.

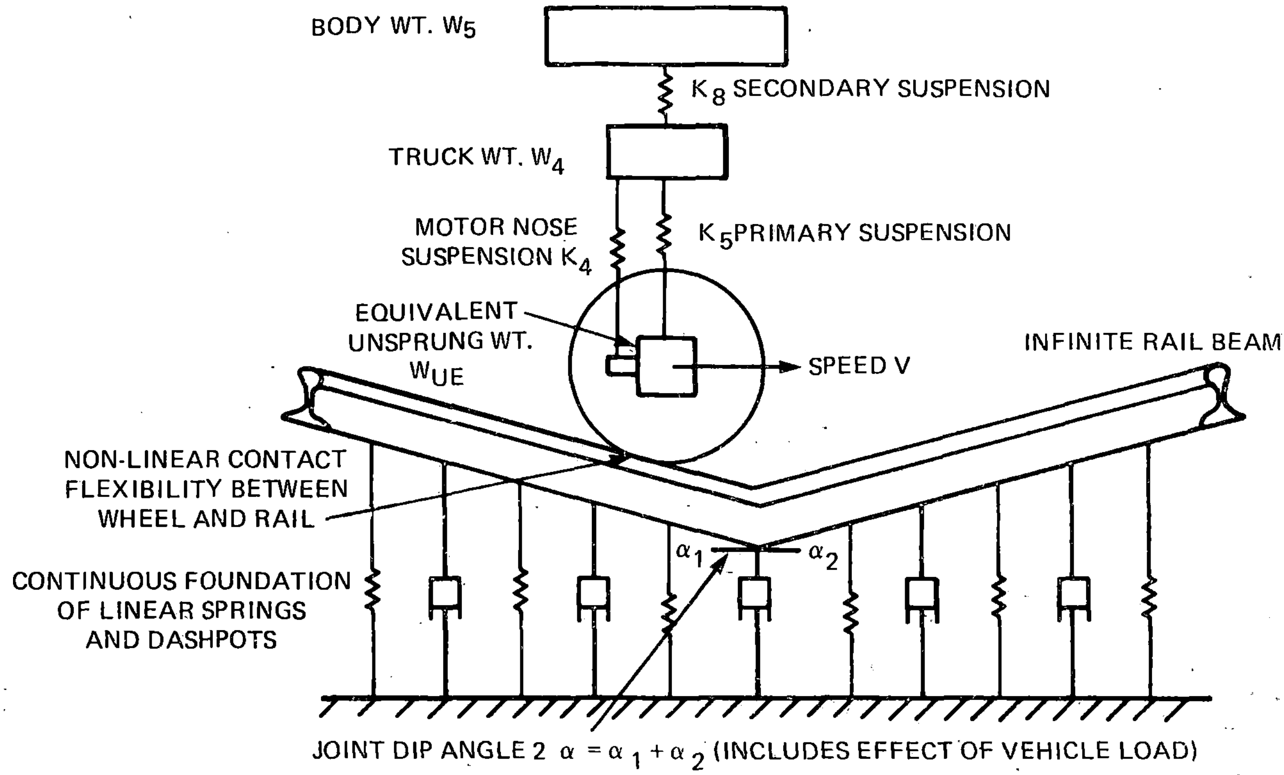

FIGURE 9. WHEEL/RAIL FORCE SIMULATION MODEL

secondary suspension appropriate for one wheelset. The modeled unsprung mass includes also the rotational inertia effects of the traction motor and of its armature through the gearing.

A graph of typical output from the model is shown in figure 10. It can be seen as the wheel passes over the dipped joint, two principal vertical force peaks are generated between the wheel and the rail. While these force peaks have the appearance of a transient phenomenon, they are in reality steady state since the vehicle is passing over successive dipped joints, each the same as the other and evenly spaced. The phenomenon would be quasi-steady state if vehicle velocity were varying.

The first force peak, P1, occurs very quickly after impact, within a few inches, and is of high frequency (500-1,000 Hz). It corresponds to the localized oscillation of the unsprung wheel mass and the track mass connected by the elasticity between them. The increment of P1 above the static wheel load P0 is thought to be responsible for localized damage to the rail ends (rail batter), but is not transmitted into the ballast and does not cause significant track deflection.

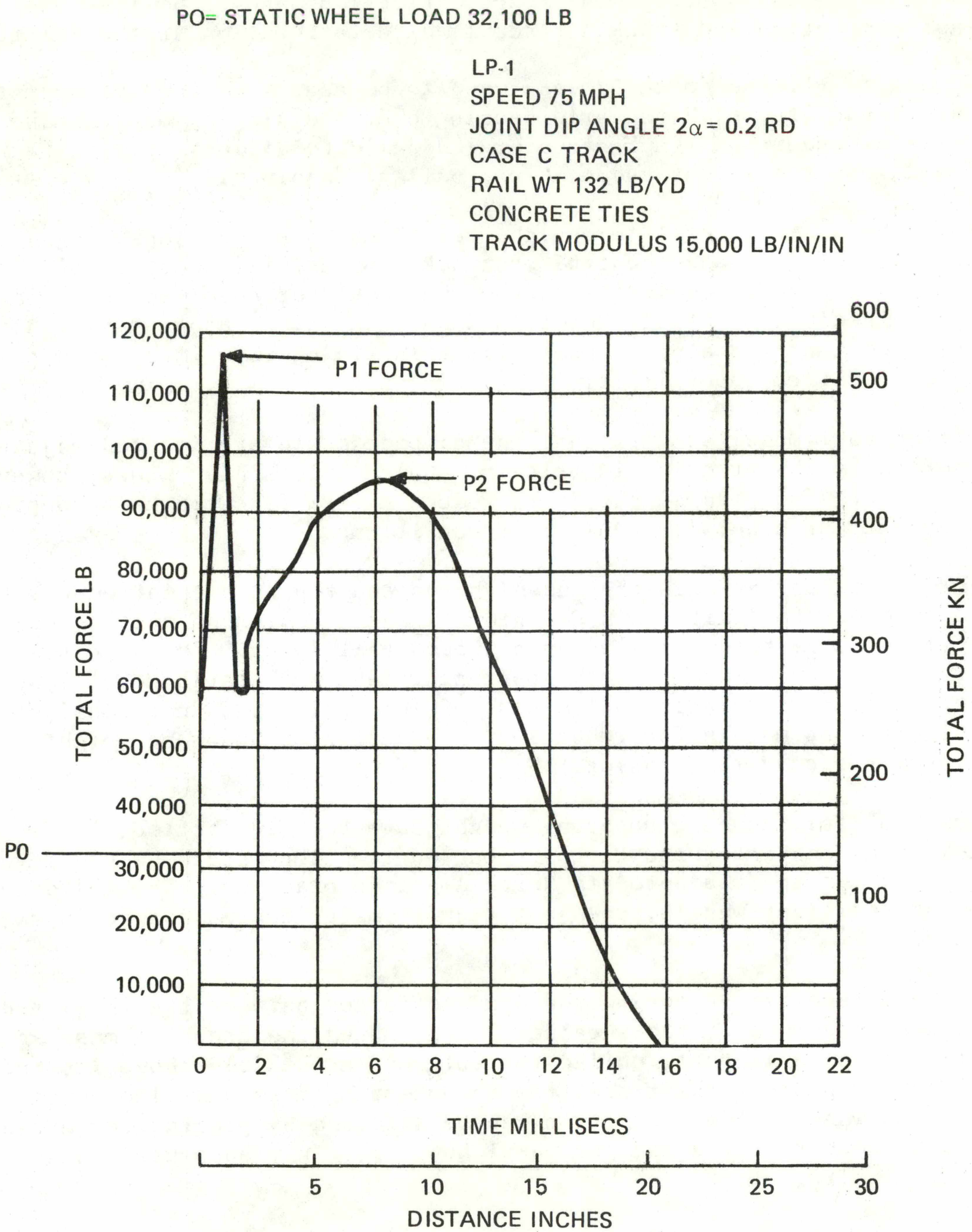

FIGURE 10. TYPICAL FORCE/TIME HISTORY AT DIPPED RAIL JOINT (CALCULATED FROM BR MODEL)

The second force peak, P2, is of greater duration and lower frequency than P1. It occurs at 20-100 Hz within approximately 10 inches of the joint. Its increment above the static wheel load, as is the approximate case with P1, is proportional to the product of vehicle speed and joint dip angle. P2 is transmitted into the ballast, causing track deflection, and is believed to cause ballast compaction and track surface level deterioration at the joint.

In contrast to P1, the force P2 was found to be heavily dependent upon the unsprung mass associated with the axle passing over the dipped joint. Like P1, P2 is also dependent on track structural dynamic conditions (i.e., track weight and modulus, tie weight and spacing, ballast damping).

Figure 11 shows, in general terms, the relationship between unsprung weight, the P2 force peak and the type of track. By utilizing curves of the form of those in figure 11, calculated from the model for specific track data, the proper combinations of static wheel load and equivalent unsprung weight can be found such that for a given speed limit the force P2 will be less than or equal to a specified upper limit.

The British Rail model assumes that the dipped joints are located adjacent to each other as is the practice in Britain. Calculations have shown, however, that the effect of staggered joints is to reduce the P2 force peaks by approximately 15 percent for otherwise equivalent conditions.

CNR's[13] conclusion has been that as higher speed trains are introduced to its system, in order to limit track deterioration the equipment must be designed to limit the P2 force created at a dipped rail joint. To do this, it is necessary to reduce either or both unsprung weight and static wheel load. Static wheel load, particularly in the case of locomotives, is not considered to have a significant margin for reduction. Reduction of unsprung weight is thus considered the only real option.

Three methods for reducing unsprung weight have been identified.[14] Two of these, truck frame suspended and carbody mounted traction motors, would involve major changes in design if adapted to North American practice. The third method, use of resilient wheels, does not require major design change in adjacent equipment.

One of the paremeters affecting the dynamic forces between the wheel and rail is the unsprung mass of the wheelset. What makes the unsprung mass so important in this area is that, unlike the bulk of the vehicle above it, there is no mechanism to protect the track from the unsprung mass and vice versa. With a resilient wheel, some protection can be afforded by placing rubber suspension pads between the tire and the wheel hub. The only absolutely unsprung mass is then the tire.

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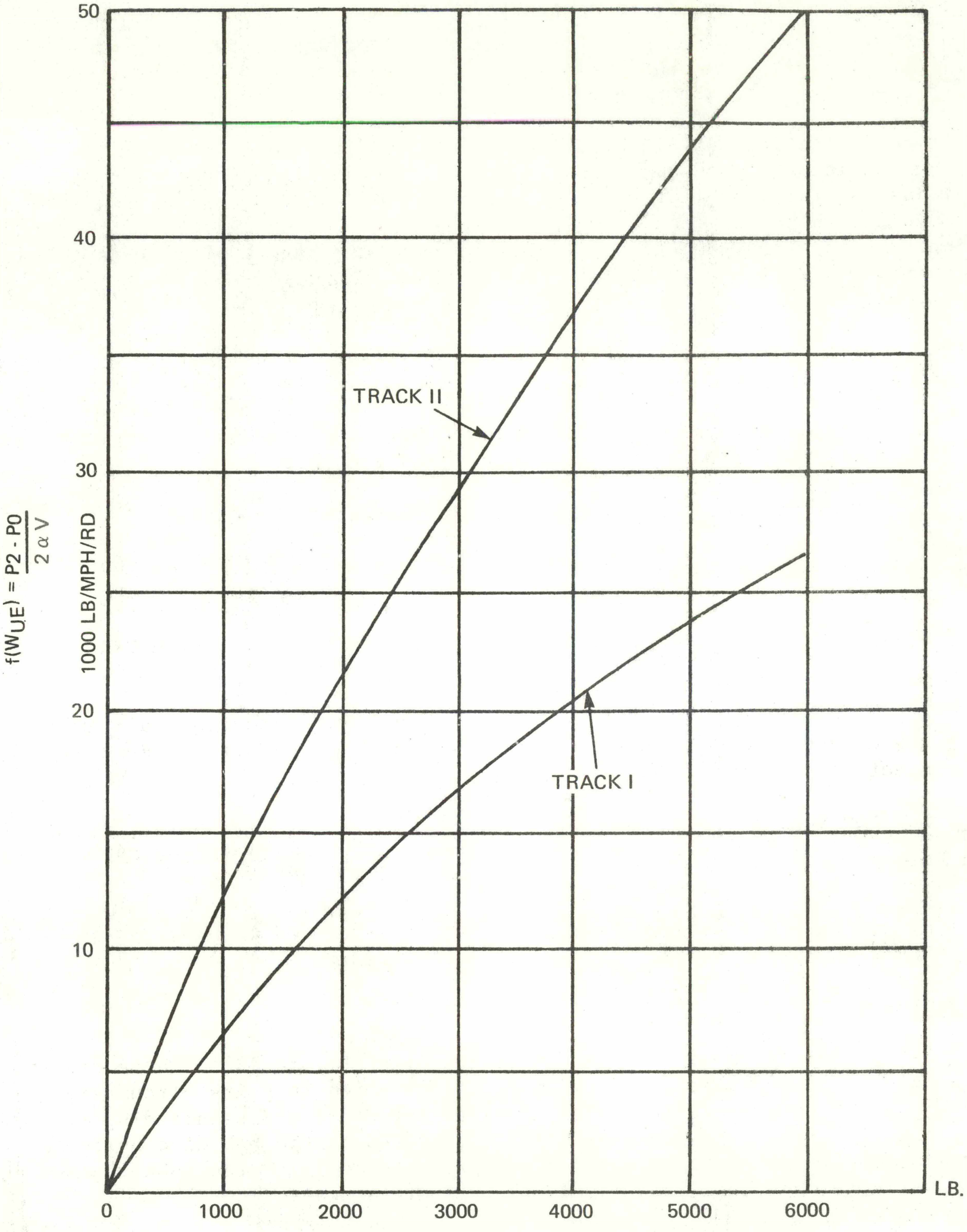
[13]Radford, ibid.

[14]Radford, ibid.



W_{UE} = EQUIVALENT UNSPRUNG WEIGHT PER WHEEL

TRACK I: WOOD TIES

TRACK II: SAME RAIL AS TRACK I, CONCRETE TIES WITH WIDER SPACING

FIGURE 11. $\frac{P2 - P0}{2 \alpha V}$ VERSUS EQUIVALENT UNSPRUNG WEIGHT (FROM BR MODEL)

THE SAB RESILIENT WHEEL

The Swedish firm SAB has been producing resilient wheels for more than 40 years. Covell[15] discusses the application of these wheels to both British Rail and Swedish Rail locomotives.

British Rail's interest in resilient wheels was identical with that of CNR. They anticipated increased track damage from the higher speed (99 mi/h and up) passenger trains that would be operating over the Anglo-Scottish West Coast mainline which was then being electrified. With a background of 10,000 resilient wheels already applied to mainline vehicles, SAB supplied this type wheel to British Rail for installation on its Class 86/2, 80-ton electric locomotives.

SAB[16] claims that the following benefits accrue from substitution of their resilient wheels in place of conventional solid wheels:

a. Considerable reduction of dynamic wheel/rail forces at vertical track irregularities; the CNR analysis also predicted this.

b. Major reduction of lateral dynamic forces at the axle.

c. Significant reduction of wheel/rail noise (implicit from CNR study).

d. Longer periods between wheel reprofilings (implicit from CNR study).

e. Reduced stress levels on wheels, axles, axle boxes, and all axle-borne equipment due to reduced inertia forces (also implicit from CNR study; wheel contact stress was addressed directly).

f. For vehicles with axle-hung traction motors or gear boxes, reduced torque fluctuations from rail top irregularities (also implicit from CNR study).

g. Increased tractive effort available and reduced wheel-slip tendency, due to the torsional flexibility of the wheels.

THE SAB DESIGN

The SAB resilient wheel consists of seven parts (figure 12). These are the monobloc tire (3) suspended on rubber pads (6) arranged in pairs circumferentially which are compressed in the axial plane between the pressure disc (2) and the hub disc (1). The pressure disc is fastened to the hub disc by bolts (4), and studs, to give the required axial precompression of the rubber pads which distribute the wheel load evenly throughout the wheel.

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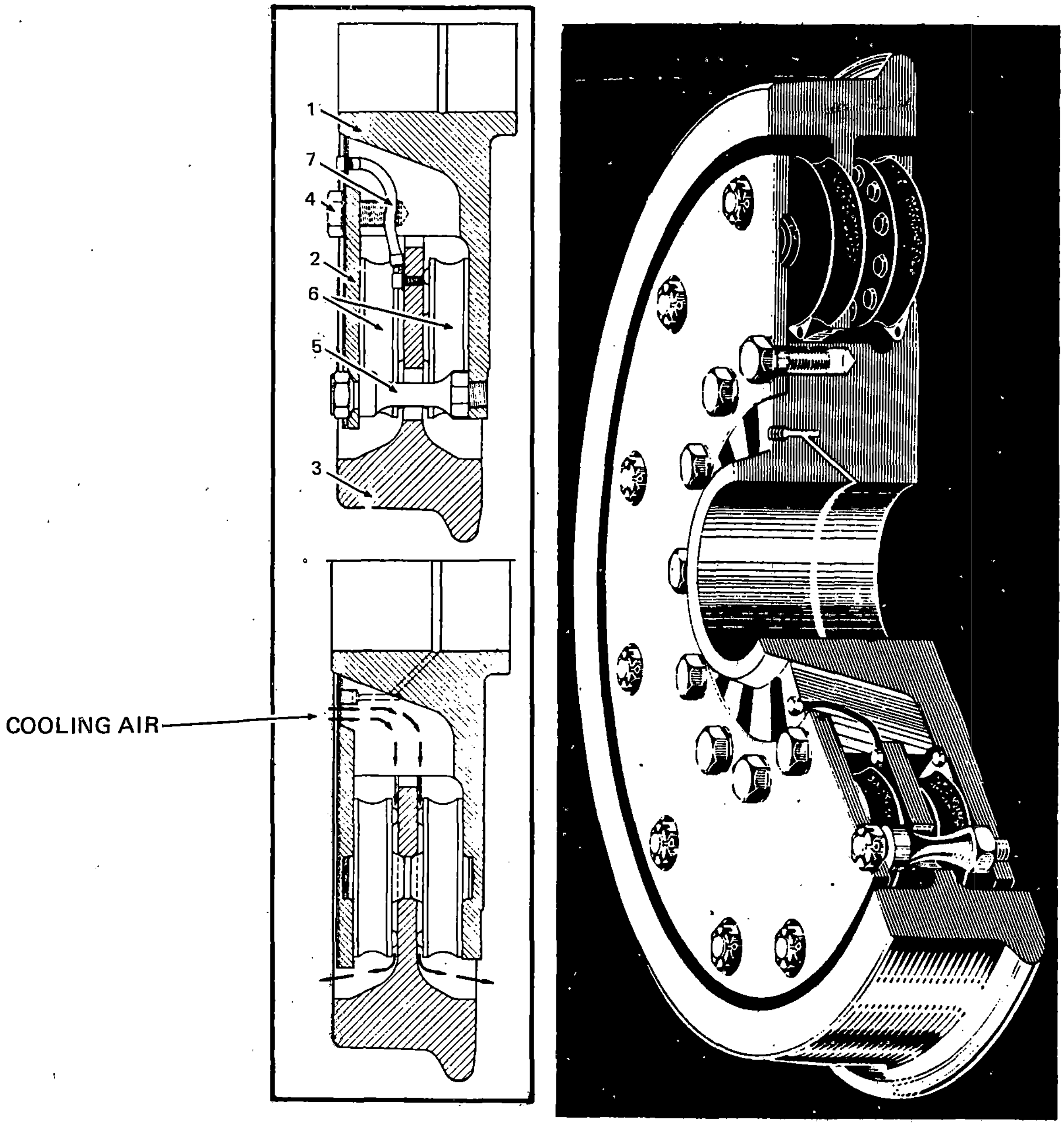
[15] Bo G. Cavell, "Resilient Wheels of SAB Design Applied to Mainline Locomotives of High Power," Rail Engineering International, January 1974.

[16] Ibid.



1. INNER OR HUB DISC
2. OUTER DISC
3. TIRE, INTEGRAL WITH MIDDLE DISC
4. OUTER DISC SECURING BOLT
5. SPACER BOLT, SECURING OUTER DISC TO HUB DISC
6. PAIRS OF RUBBER BLOCKS COMPRESSED BETWEEN OUTER, MIDDLE AND HUB DISCS
7. ELECTRICAL CONNECTION BETWEEN TIRE AND HUB

FIGURE 12. THE SAB RESILIENT WHEEL

By varying the hardness of the rubber and the thickness, diameter, and number of the rubber pads, the characteristics of the wheel can be adjusted within certain limits to meet the requirements of individual applications. Tired wheels can be used instead of the monobloc type. Oil injection grooves, to facilitate removal of the wheel from the axle, and electrical connections are supplied to customer specifications. Wheels can be supplied to provide full disc or full on-tread braking. The wheel illustrated in figure 12 is a typical high-speed design giving high torsional stiffness and for which full disc braking can also be provided. The rubber elements are subjected to shear stresses by radial and tangential forces and to compressive strains by axial or horizontal forces. To help dissipate the heat generated by these deflections in high-speed service and by full on-tread braking, the wheel hub is provided with spokes designed to act as fan blades and circulate coooling air around the rubber pads.

BRITISH RAIL TESTS

Before embarking on a major retrofit project of installing resilient wheels on 58 of its Class 86/2 locomotives, British Rail (BR) conducted a test program[17] on two of these locomotives. The following aspects of vehicle performance were monitored:

a. Track forces generated at discrete and random vertical irregularities.

b. Traction motor environment.

c. The ride quality of the locomotive in all three planes.

d. Safety.

e. Creep deflections of the rubber pads within the wheels and their effect on vehicle performance.

f. The effects on the rubber pads of temperatures generated by tread braking and normal running.

The two locomotives were identical except for their secondary suspensions and traction motor gear ratios. One was a standard Class 86; the other had a flexicoil secondary suspension and a reduced traction motor gear ratio.

Track force measurements were made at a test rail-joint installed in the high-speed London to Glasgow mainline. The instrumentation consisted of 12 load-sensing base-plates installed on the two ties immediately preceding the test joint and the four just after it; rail stresses were recorded from 12 strain gage locations. Track force/speed curves were determined for each locomotive with and without resilient wheels; and example is shown in figure 13.

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[17]Cavell, ibid.

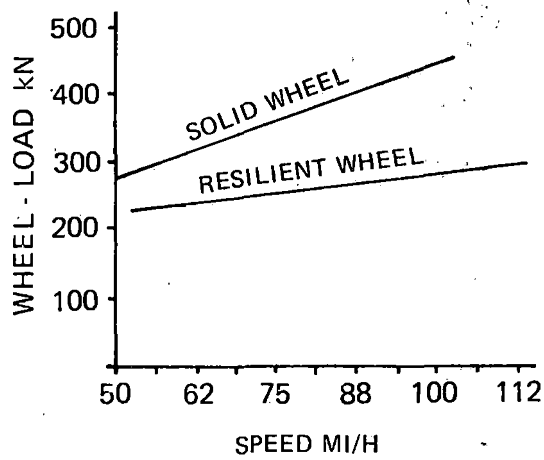

FIGURE 13. TRACK FORCE/SPEED CURVES MEASURED AT INSTRUMENTED RAIL JOINT-BR CLASS 86/2 LOCOMOTIVE

The dynamic component of the force exerted by the resilient wheels was found to be approximately linear with speed and 50 percent less than that developed with solid wheels at all speeds. From figure 14 it can be seen that the forces generated by the SAB wheels at 99 mi/h are equivalent to those induced by conventional ones at 53 mi/h at the same joint; this was confirmed by wheelset accelerometer measurements as the locomotive passed over the test joint. These forces correspond to the P2 force peaks observed in the CNR study.

The strain gages on the rail indicate that the impact forces at the end of the rail being run onto are reduced by about 10 percent. These correspond to the P1 force of the CNR study, and the low reduction is consistent with the finding of that study that P1 forces are not much affected by changes in unsprung mass.

The use of the term "unsprung mass" here is in the theoretical context of the CNR study; that is, the addition of the resilient elements to the wheel causes the tire to be the only unsprung component. In reality, of course, this is not completely true since the rubber pads create not only a very stiff resilient connection, but also a nonlinear one.

The track force measurements also indicated that the maximun stress range at the first bolthole connecting adjacent tracks was reduced by as much as 40 percent (figure 14). The result of the force reduction is to reduce the probability of fracture failure of the rail end adjacent to the boltholes of the run-onto rail.

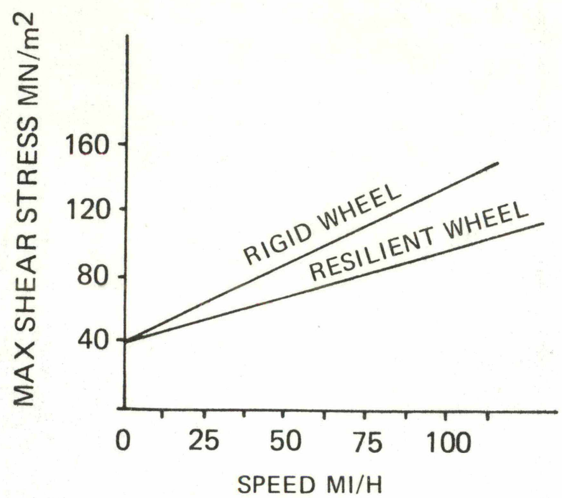

FIGURE 14. MAXIMUM STRESS AT FIRST BOLTHOLE OF A FISHPLATE

As can be seen from figure 9, if there were perfect rail alignment and an even distribution of track elasticity, there would be no impact forces generated as the wheel passed over the joint. Imperfections do exist, however, and not only at rail joints; the loads experienced by the adjacent ties in response to the impact forces lead to failure of the supporting ballast, and this in turn causes the rail surface irregularity to become worse. The impact forces then become more intense, and the process repeats itself until either rail failure occurs or the track becomes too rough to run on.

Figure 15 shows a probability density analysis of wheelset vertical acceleration covering the 0 - 100 Hz frequency range. It was recorded over a section of continuously welded track at 99 mi/h for both rigid and resilient wheels. Comparing the rms acceleration levels in the two cases shows a 45 percent reduction in favor of the resilient wheels.

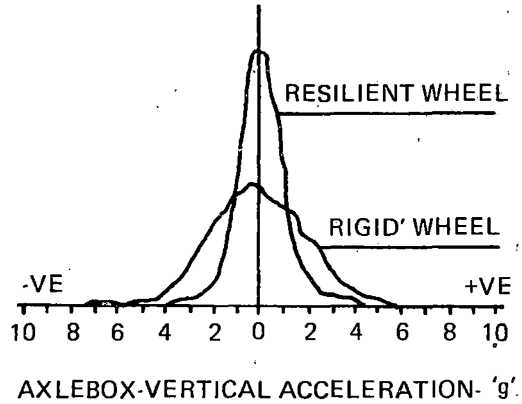

FIGURE 15. PROBABILITY DENSITY DISTRIBUTION OF VERTICAL WHEEL ACCELERATION ON CONTINUOUS WELDED TRACK

Figure 16 shows comparative acceleration versus speed information for rigid and resilient wheels during the traversing of switches and crossings. Again, there is a 45 percent reduction in favor of the resilient wheel.

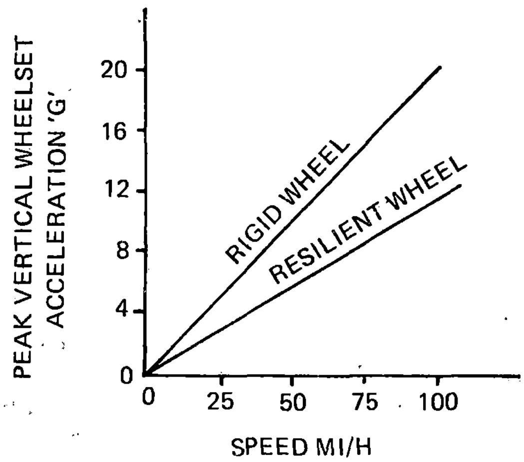

FIGURE 16. VERTICAL WHEELSET ACCELERATIONS AT SWITCHES AND CROSSING

Comparing the conclusions from figures 15 and 16 with the approximate 50 percent reduction of vertical acceleration in favor of resilient wheels on the instrumented test rail-joint, it is apparent that for equivalent track and roadbed structures resilient wheels will cause about the same percentage reduction of P2 type vertical wheel/rail force for any track surface condition.

THE NORWEGIAN STUDY

The Norwegian National Railways (NSB) are also investigating[18] the use of SAB resilient wheels. The NSB rights-of-way possess a high percentage of curved track with the result that any significant average speed increase will be dependent on increasing allowable speeds through curves.

In evaluating various locomotive and car combinations traveling on curves, they have come upon the situation in which a prospective locomotive could operate faster than others through the curves without generating excessive lateral forces on the track, but is limited by the excessive dynamic vertical forces it generates. NSB utilized a formula developed by British Rail to estimate the dipped joint vertical dynamic forces. They also noted the additional quasi-static vertical wheel load on the outer rail during curving. In the CNR analysis, this would be seen as an increase of static wheel load (P0) on top of which is superimposed the dynamic increment. NSB concluded that the dynamic vertical force reductions brought about by use of the SAB resilient wheel would allow a 5 percent speed increase.

It is not clear precisely how NSB arrived at the 5 percent number. Assuming that they had available to them the British Rail test results discussed above, however, they could utilize the 45 to 50 percent reduction factor on dynamic vertical forces that the SAB resilient wheel showed there. They could then utilize the known P0 load and calculated (from the BR formula) dynamic vertical load increment for the current speed as a basis for calculating a speed increase with resilient wheels which would leave the maximum vertical force on the outer rail unchanged.

Whatever the case, it is clear that NSB is seeking to utilize the same dynamic vertical force reduction capability of the SAB wheel for curve speed increase that CNR and BR would use for track damage control on any high-speed line.

LATERAL FORCES

As Radford[19] notes, when the wheelset traverses irregularities in alignment of the rail, wheel/rail lateral forces occur and their magnitude depends upon the unsprung mass associated with the wheelset. These forces are additive to the quasi-static curving forces felt by the wheelset and thus affect the ratio of lateral to vertical wheelset forces (L/V ratio).

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[18]Magne Glomnes, Norwegian National Railways, "Higher Cruising Speed, An NSB Research and Development Project," publication unknown.

[19]Radford, op. cit.

A decrease in the effective unsprung mass of the wheelset can reduce the L/V ratio on rough curved track, and the corresponding probability of derailment during curve negotiation due to the L/V limit being exceeded. This may also have factored into the Norwegian conclusion that the SAB resilient wheel would allow a 5 percent speed increase on curves.

Cavell[20] notes that the additional lateral plane flexibility introduced by the SAB wheel could potentially cause truck instability or hunting. British Rail handled this problem by providing a stiff hydraulic yaw damper in the secondary suspension of the class 86 locomotive; it was mounted in stiff rubber mountings and was proved sufficient to maintain stability up to 124 mi/h under high wheel conicity conditions. Comparative lateral wheelset accelerations on straight track are shown in figure 17.

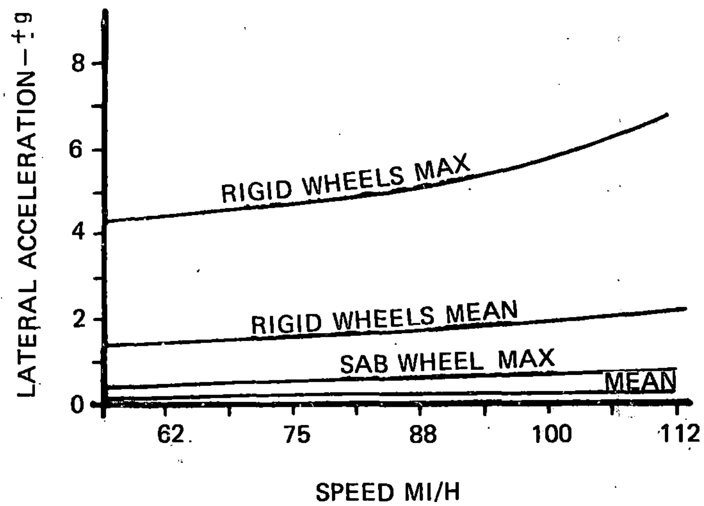

FIGURE 17. LATERAL WHEELSET ACCELERATIONS ON STRAIGHT TRACK-COMPARISONS OBTAINED WITH A BR CLASS 82/2 LOCOMOTIVE FITTED WITH SAB RESILIENT WHEELS AND PLAIN WHEELS

Cavell notes further that lateral flexibility has been introduced with axlebox springs in the past, but must be up to 10 times greater than when introduced via resilient wheels to achieve similar results.

TRACTION MOTOR BENEFIT

During the BR tests of resilient wheels, accelerometers were installed on the axle-hung traction motor frames. In the 0-100Hz frequency range, maximum vertical accelerations were reduced from 20 to 12g. Also, according to Cavell, torque fluctuations at the motor armature can be reduced up to 30 percent with resilient wheels.

[20] Cavell, op. cit.

SAFETY AND RIDE QUALITY

Ride quality of the BR Class 86 locomotive was found to be unaffected by the SAB wheel even after 17,000 miles of service. The hydraulic yaw damper prevented truck hunting even though the locomotive has flexicoil secondary suspension.

CREEP EFFECTS

Figure 18 shows the results of static wheel deflection measurements taken over 24 hours at ambient temperature. Another 12 hours produced no further creep (or cold flow). Similar measurements taken after a period of intensive passenger service, where the wheels reached temperatures 86 degrees F above ambient, showed a 21 percent temporary set after a 24-hour stand. It has been uniformly observed that such temporary sets smooth out well before the locomotive reaches high speed.

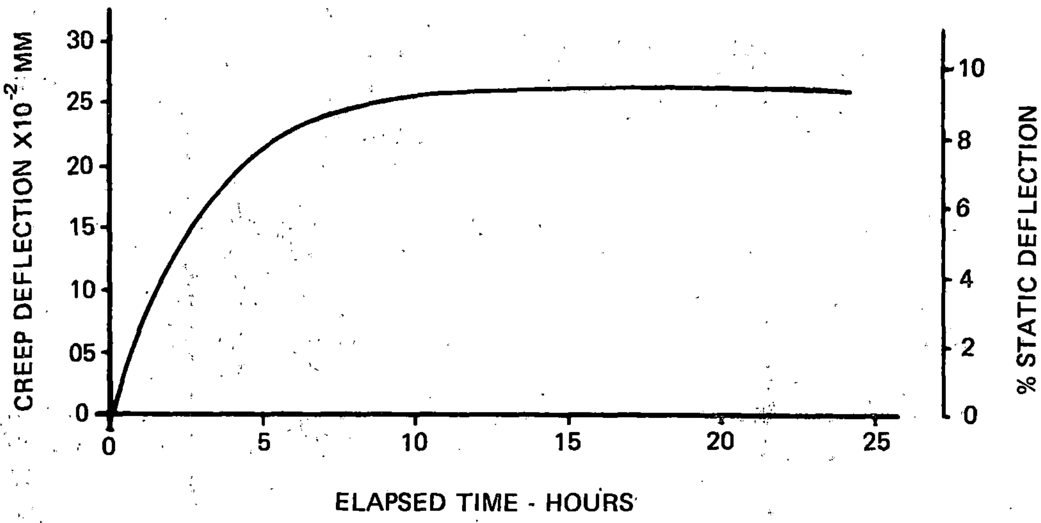

FIGURE 18. CREEP-TIME CURVE FOR A TYPICAL SAB RESILIENT WHEEL AT AN AMBIENT OF 18 DEGREES C (64.4 DEGREES F)

TEMPERATURE EFFECTS DURING BRAKING AND SUSTAINED HIGH SPEED

The Class 86 locomotive has full dynamic braking; however, in the event of malfunction, full tread braking is also available. The SAB test wheels were therefore designed for the latter condition. During full tread braking tests the rubber pads never exceeded 153 degrees F in temperature; this is well within the critical temperature of the rubber.

Rubber temperatures were observed for 3 days on one locomotive in high-speed service. The pads quickly reached an equilibrium temperature of 86 to 104 degrees F above ambient during running. Another 68 degrees F increase was occasionally seen during heavy braking. Examination of the pads after 170,000 miles of service showed no thermal damage. Bench and other service tests have shown that even emergency braking does not overheat the pads.

The low rise of temperature during high speed operation is indicative of the mechanical efficiency of the test wheels; they dissipate only 1.4 kW each due to hysteresis at 99 mi/h

MAINTENANCE

SAB resilient wheels have been produced for more than 40 years. In main-line service, monobloc tire and rubber spring lifetimes of 621,400 miles are considered normal, with long periods occurring between tire reprofilings.

A severe service example is provided in their application to Swedish State Railways (SJ) Class DM3 1D+D1 7,300 kW electric locomotives which are in iron ore service over track where the ballast is frequently frozen solid. Under arctic conditions, conventional wheels tend to slip and become out-of-round. Combined with the stiff track, this resulted in wheel reprofilings after 25,000-62,000 miles service and sometimes as early as 12,500 miles. New tires were required every 186,000 miles and locomotive overhaul was fixed at 280,000 miles.

Application of resilient wheels, combined with some traction motor modifications, increased the tire reprofiling period to 124,000 miles and the tire life to 466,000 miles. The overhaul period was also increased to 466,000 miles at which time both the tires and rubber springs are replaced.

NOISE REDUCTION

The SJ took noise measurements on the Class DM3 locomotives before and after application of resilient wheels. The noise experienced by the locomotive crew was reduced from 95 to 85 dBA with the application of resilient wheels.

ACKNOWLEDGMENTS

The sections entitled "Introduction" and "Theoretical Summary" of the discussion of resilient wheels are based on Radford's paper.[21] This paper received the ASME Rail Transportation Award as the best ASME Rail Transportation Division paper of 1977.

The section entitled "The SAB Resilient Wheel" is based, except where otherwise footnoted, upon Cavell's article.[22]

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[21] Radford, op. cit.

[22] Cavell, op. cit.

SECTION 3. LIGHTWEIGHT AXLES

INTRODUCTION

The effect of unsprung mass has been discussed at length in the "Resilient Wheels" section. It was shown that any decrease in effective unsprung mass, such as through the use of resilient wheels, would make a substantial contribution to the reduction of track damage by high-speed trains.

The resilient wheel accomplishes the reduction of unsprung mass through quasi-isolaton of most of the wheelset from the direct experience of wheel/rail contact forces. Any other method of reducing wheelset mass, such as use of a lighweight axle, will produce a similar result.

Hollow axles have been used for this purpose; they have presented durability problems[23] when the normal press-assembly method has been used. A particularly lightweight approach would be the use of an axle made of a composite material. British Rail[24] may be doing research on a fiberglass-composite axle but no details are known at present.

TORSIONAL FLEXIBILITY

Any modification to axle-design must consider the effect of the modification on the torsional flexibility of the axle. Since the wheel/axle combination amounts to an isolated torsional spring/mass system that is constrained by the wheel/rail friction force, the possibility of torsional oscillations combined with stick-slip phenomena exists.

The Austrian Federal Railways encountered this problem on driven axles[25] when their Series 1044 electric locomotives were introduced. The problem did not stem from an axle modification, but rather from a drive-system change.

The locomotive utilized a flexible drive to a gear on one end of each driven axle. The more usual arrangement was of two symmetrically located drive gears on each axle.

Two modes of vibration were encountered. One was between the masses of the armature and wheelset through the flexible drive connection; the other was the wheel-to-wheel oscillation discussed above. The latter mode was considered most serious since axle fractures occurred as a result of it.

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[23] F. Hegenbarth, "A Lightweight Wheelset for High Speeds in the Case of the Wheel/Rail System" (translation of German title), Glasers Annalen ZGV, N11, 1972.

[24] Private discussion with British Rail delegate at Sixth International Wheelset Conference, Colorado Springs, Colorado, October 1978.

[25] "SGP Investigates Torsional Vibrations in Driven Axles," Railway Gazette International, August 1978.

The occurence of the wheel-to-wheel oscillation was due to the offset location of the drive gear. The greater proportion of the driving torque was taken up by the wheel on the same side which then tended to slip and start the wheel/axle torsional oscillations. The resonant oscillation would then set off slip in the other wheel about the time the first one had regained its adhesion.

The introduction of a lightweight axle is most likely to be synonymous with introducing more axle flexibility since either or both the polar moment of inertia of the cross-section and shear modulus of the material will probably be reduced. This in turn lowers the torsional stiffness of the wheelset.

In the offset drive situation, the tendency for single wheel slip and accompanying lower frequency resonant oscillation would be exacerbated by the reduced axle torsional stiffness. Symmetric location of the drive system solves this problem by simply not forcing the mode. There is, however, another arena for this mode to play in -- truck stability.

EFFECT ON TRUCK STABILITY

Doyle and Prause[26] have investigated the hunting stability of rail vehicles with torsionally flexible wheelsets. The primary model used was that of a single truck with two torsionally flexible wheelsets, although a complete car hunting model was also studied. The influence of various parameters on the stability characteristics of a typical rail vehicle was shown for varying wheelset stiffness.

The rail vehicle model consisted of a carbody, 2 rigid truck frames with primary and secondary suspension elements, and 4 torsionally flexible wheelsets. The carbody and truck frames each had the yaw, roll, and lateral degrees of freedom. Each wheelset had the normal yaw and lateral degrees of freedom; in addition there was a torsional degree of freedom about the axle centerline.

The equations of motion were based on the assumptions that: all suspension elements are linear; all vehicle components are rigid bodies; the vehicle is moving at constant velocity on rigid, constant gage tangent track; the wheels remain in contact with the rail; the lateral displacement of the wheelsets is small so that the effective conicity at the wheel/rail interface is independent of wheelset displacement and flanging does not occur.

The model was thus amenable to analytical solution (rather than numerical integration by computer). The eigenproblem (i.e., modal shapes and frequencies) was solved for each parameter set and vehicle speed, with the wheelset torsional stiffness being successively relaxed from rigidity until a positive real root appeared and signified truck hunting.

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[26] G. R. Doyle, Jr., R. H. Prause, "Hunting Stability of Rail Vehicles with Torsionally Flexible Wheelsets," American Society of Mechanical Engineers, paper number 75-WA/RT-2, July 1975.

Figures 19 through 27 illustrate that all of the critical speed vs. wheelset torsional stiffness loci have the same general shape regardless of which other parameter is varied. For sufficiently low torsional stiffness, critical speed decays badly for any parameter set; for high enough torsional sitffness, critical speed is relatively insensitive to it.

Doyle and Prause also observe that while low axle stiffness is the normal source of wheelset torsional flexibility, the addition of rubber elements to form resilient wheels can potentially have the same effect where truck hunting is concerned. Radford[27] also noted the increased potential for truck hunting with resilient wheels. Cavell, as discussed above, indicated that British Rail successfully dealt with the problem by adding a hydraulic yaw damper. It would thus appear that if truck hunting was the only problem standing in the way of using lightweight axles, then secondary hydraulic yaw damping is a potential solution.

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[27] Radford, op. cit.

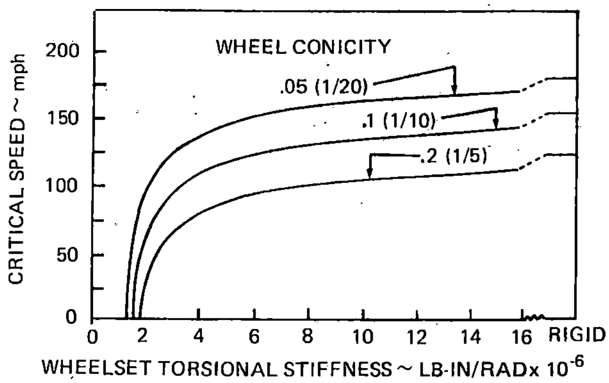

FIGURE 19. THE EFFECT OF WHEEL CONICITY ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

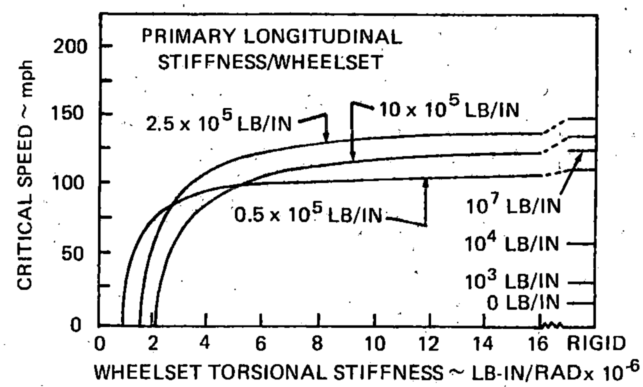

FIGURE 20. THE EFFECT OF PRIMARY LONGITUDINAL STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEEL SET TORSIONAL STIFFNESS.

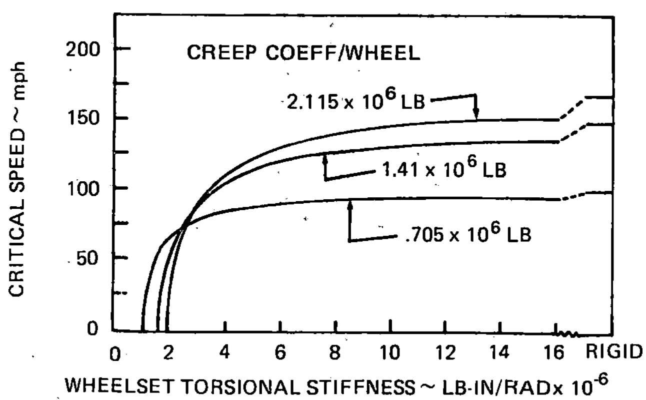

FIGURE 21. THE EFFECT OF CREEP COEFFICIENT ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

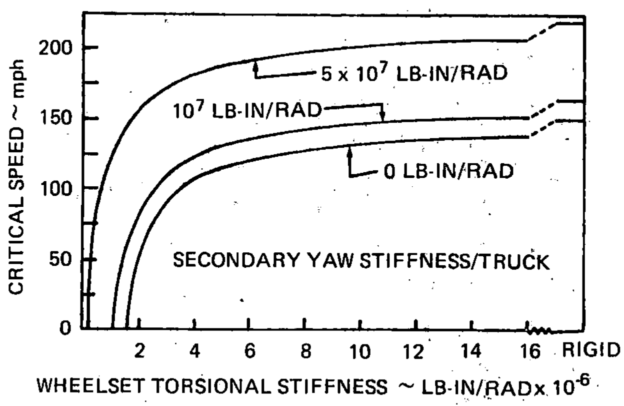

FIGURE 22. THE EFFECT OF SECONDARY YAW STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

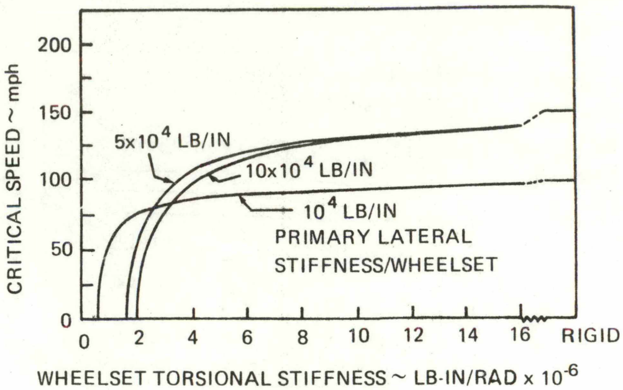

FIGURE 23 THE EFFECT OF PRIMARY LATERAL STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

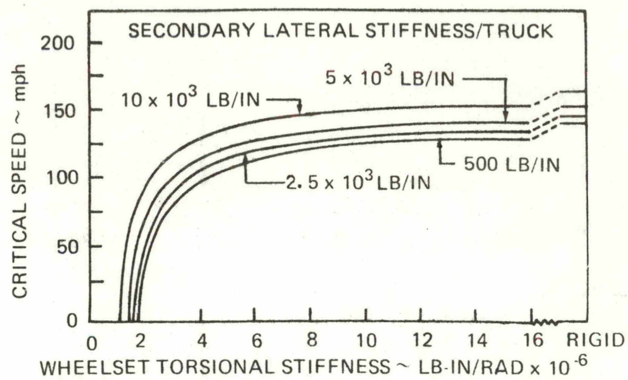

FIGURE 24 THE EFFECT OF SECONDARY LATERAL STIFFNESS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

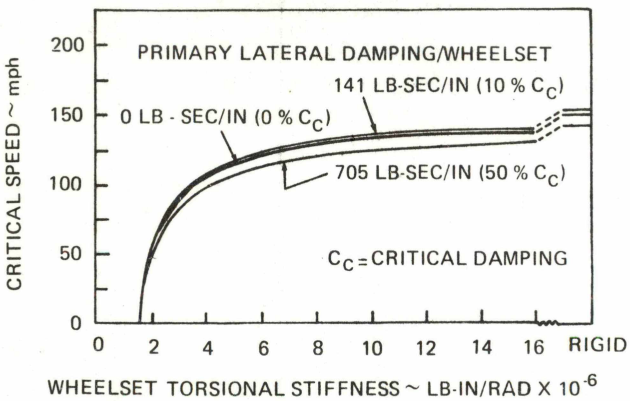

FIGURE 25 THE EFFECT OF PRIMARY LATERAL DAMPING ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

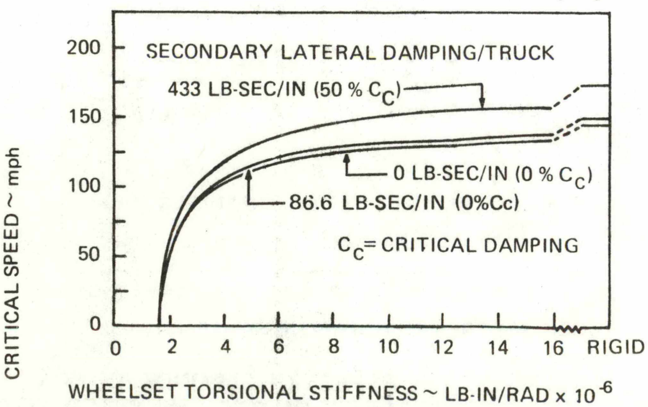

FIGURE 26 THE EFFECT OF SECONDARY LATERAL DAMPING ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

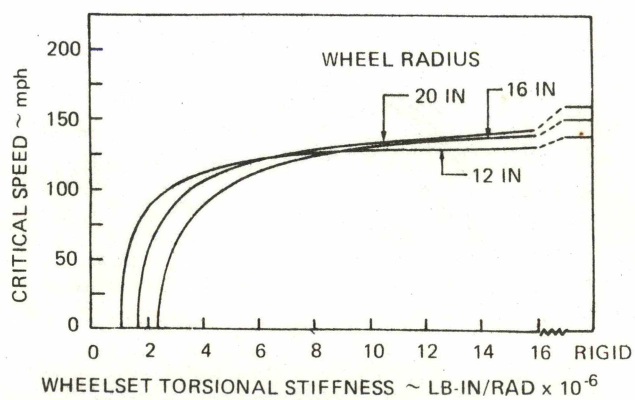

FIGURE 27 THE EFFECT OF WHEEL RADIUS ON CRITICAL SPEED WITH VARIABLE WHEELSET TORSIONAL STIFFNESS.

SECTION 4. THE AUTOMATIC COUPLER

INTRODUCTION

The development of automatic couplers has progressed to the state where a number of train line connections and car-to-car coupling can be accomplished without manual effort. This section of the report addresses the technology of recently developed automatic couplers in the United States and in Europe.

There have been a number of developments in railroad operations and car design that have made the standard couplers of the 1940's and 1950's inadequate for a number of newly designed cars. Also these couplers are not compatible with recent improvements in railroad operations. This has been due to: the introduction of longer cars and passenger coach designs that feature increased comfort (i.e., more motion at the coupler due to flexible suspension and longer cars); the need to eliminate the manual operation of connecting the air hoses and angle cocks; and the need for control mechanisms which can be activated from either side of the coaches. Coupler modifications and replacements involve heavy costs and must be achieved with a minimum disruption to rail operations. This is complicated by the fact that the one part of a railroad car which must be standard or universally compatible is the coupler. An automatic coupler, of simple design, rugged and consistently reliable, and interchangeable with others, to provide maximum safety and long-term savings is needed in the United States.

DEFINITION OF AUTOMATIC COUPLING

According to the American concept, mechanical coupling is considered to be automatic when the following occurs: coupler alignment is within specific narrow limits, at least one of the couplers is open, and the cars impact at the proper speed and on tangent. The European concept states that if full mechanical coupling occurs upon contact it is automatic--even if uncoupling is manual. Uncoupling is considered to be automatic when a lever is manipulated to unlock the coupler, eliminating the need for anyone to go between cars for manual unlocking.

Technically speaking, the term "coupling" refers to the mechanical phase only. The fully automatic coupler is one that can be operated by the push of a button from a remote position to control mechanical, electrical, and pneumatic coupling and uncoupling. The only domestic couplers that could qualify as being fully automatic are those in use on Metroliners and subway or rapid transit cars. The couplers predominantly in use are semiautomatic, requiring some manual operation for reasons of cost and safety.

SCOPE OF REVIEW

This review is concerned primarily with the lastest coupler designs from the mechanical, pneumatic, and electrical train line concepts. The evolution and development of advanced couplers in both the United States and Europe are pertinent and are included. From a functional standpoint, there are over 100 possible coupler systems or component application concepts to be considered. They vary from designs that provide automatic train line connections to consideration of the addition of centering or positioning devices.

SELECTION OF COUPLER DESIGNS

Selection of the latest coupler designs has been facilitated as the result of work done by two coupler committees operating almost concurrently in Europe and in the United States. The timing of this review is propitious inasmuch as both of these committees have recently reached major milestones as follows:

a. The International Union of Railways (UIC) recently completed the development of a standard coupler for use in European railways more than 20 years after setting this goal. The coupler is described in detail in the voluminous reporting of the ORE B-51 Committee (Office for Research and Experiments on Question B-51) which, in 1969, produced the basic design of an automatic coupler from three prominent European models[28] and labeled it the UIC69e. After refinements and modifications were added to assure interchangeability of subassemblies, a UIC jury made the final selection in 1976. Production is presently underway. Studies have been completed and direct coupling with the automatic coupler for freight cars is assured.

b. This "Unicoupler/Eurocoupler" is not significantly different from the three United States standard couplers, which are capable of interchange with one another: Type E Interlock and Type F Interlock for freight cars and the Type H Tightlock for passenger cars. Although railroads in the United States have been deterred from pursuing a number of significant patents for improving couplers because of the decline in rail utilization (and profits), more recent events, as related to energy and environmental factors as well as cost and safety, reflect an increased interest and emphasis. The Federal Railroad Administration (FRA) initiated a number of projects intended, on a technical basis, to improve rail transportation service, efficiency, and productivity. This work is closely coordinated with the Association of American Railroads (AAR) Advanced Coupling Concepts Steering Committee, which is part of a larger program, the FRA and AAR Joint Study Group.

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[28]The Candidate Coupler Models were placed into three groupings: the UNICOUPLER, the Associated Willison, and the Eurocoupler.

SECTION 5. DEVELOPMENT OF THE U.S. COUPLER

EARLY TRAIN COUPLERS

Present day couplers are direct descendants of early train couplers. The earliest connectors consisted of the use of any eyebar or links of chain connecting the cars. In Great Britain, and elsewhere in Europe, a chain system is still in use. It has spring buffers at the corners of adjoining cars to tension the chain, which is adjusted by a turnbuckle tying passenger cars together. In the United States, the early couplers evolved into the "link and pin" mechanism used on July 30, 1831, for the famous trip of the DeWitt Clinton (figure 28).[29]

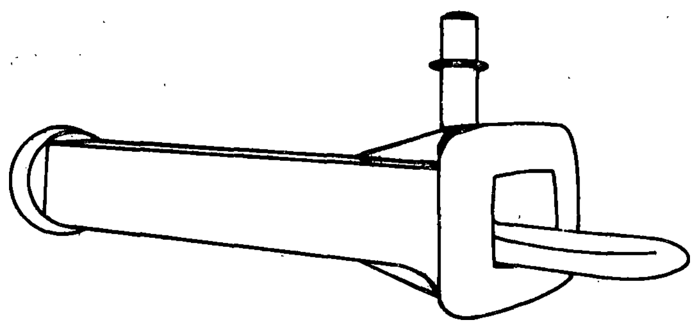

FIGURE 28. LINK AND PIN (1831)

The link and pin coupler worked reasonably well, but was not safe. The need for an improved vehicle connection was recognized early in domestic railroad history, and a design by Ezra Miller was accepted by most railroads in this country and abroad. The "Miller Hook" (figure 29), which combined a safety platform, coupler, and buffer, was heralded as the greatest life saving invention of the age.

[29] "Railroad Couplers--From the Historical Collection of the National Castings Division," Midland-Ross Corporation, Sharon, Pennsylvania, 1976.

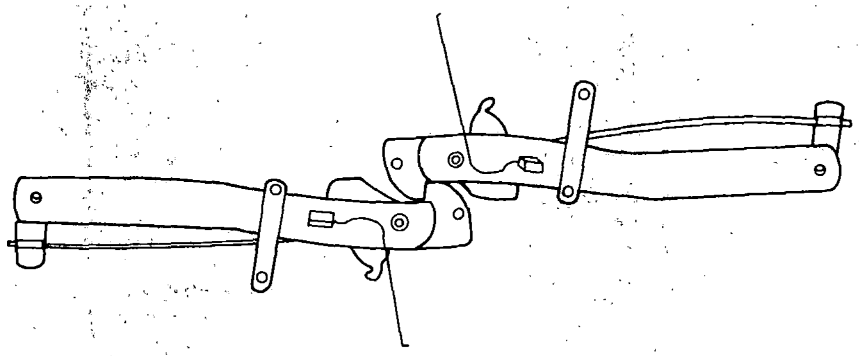

FIGURE 29. MILLER DRAW HOOK (1863)

Despite this acclamation, the Pennsylvania Railroad opted for a device patented by Major Eli H. Janney in 1866, which has become the common ancestor of couplers in use today. It was the first coupler to feature a vertical lock actuated on the closing movement of the knuckles (figure 30). More important, it could couple cars automatically.

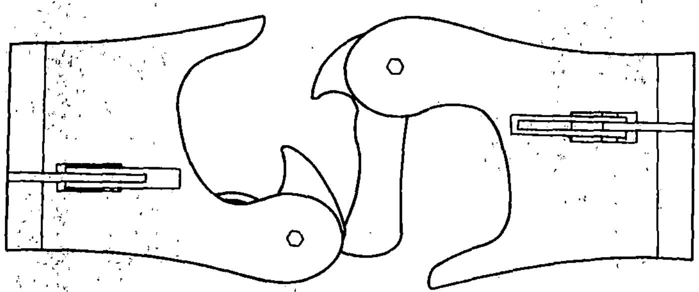

FIGURE 30. JANNEY COUPLER (1873)

The Janney coupler was simple in design, rugged, and reliable. Subsequent models, all based on the "clasped hand" principle (figure 31) provided additional improvements until the "swinging knuckle" design was selected for standardization in 1882 (figure 32). To couple automatically, at least one knuckle must be open when the cars are pushed together; the knuckle swings to the closed position and a lock drops in place and keeps it closed. Uncoupling is accomplished by opening the side lever by hand.

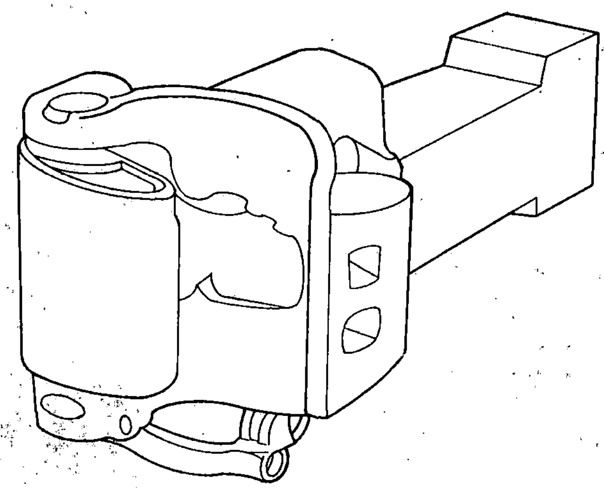

FIGURE 31. JANNEY CLASPED HAND COUPLER

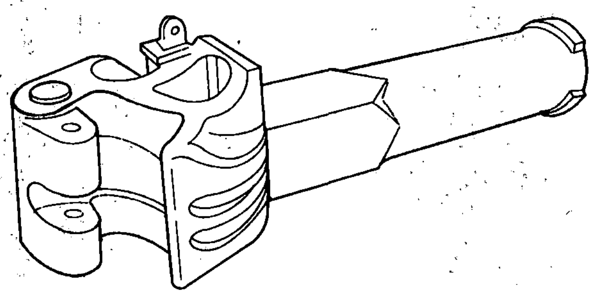

FIGURE 32. JANNEY SWINGING KNUCKLE DESIGN

In 1916, an improved version of the Janney coupler, the Type D was selected for standarization because it featured interchangeable parts from many different manufacturers (figure 33).

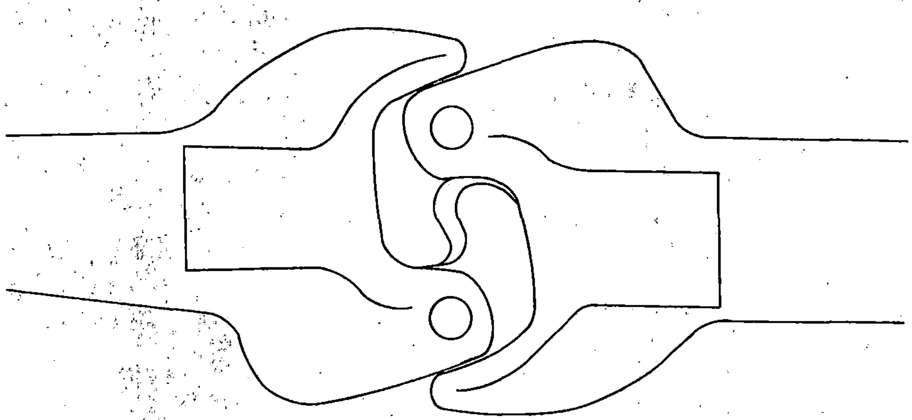

FIGURE 33. JANNEY TYPE D (1916)

During the World War I time frame, the Willison coupler appeared, operating on a different principle and offering better advantages in automatic coupling and coupling-up range. It failed to gain acceptance in North America because there were already too many Janney couplers on hand to justify a change. The Willison then became the basic European coupler design. It operates on a semirigid "spread claw" principle (figures 34 and 35). The single cast unit makes it cost effective to produce and gives it a highly favorable strength to weight ratio. Unlike American trainmen, their European counterparts do not have to couple the brake pipes by hand. The Willison design gained prominence in specialized operations that relied on its ruggedness to counter the continuous movement of rail cars.

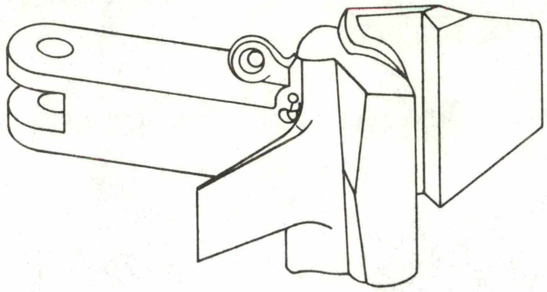

FIGURE 34. WILLISON INDUSTRIAL DESIGN (1916)

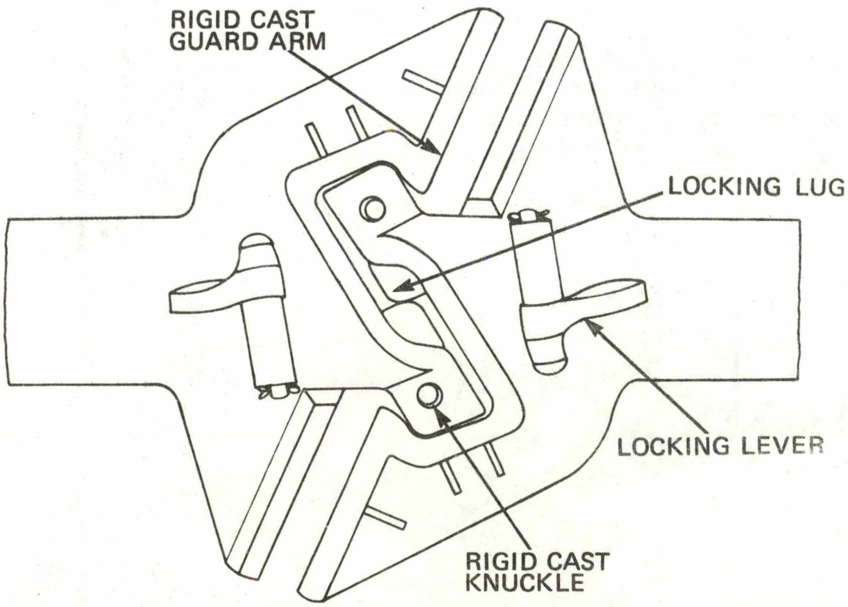

FIGURE 35. WILLISON SPREAD CLAW PRINCIPLE

MODERN AMERICAN PASSENGER CAR COUPLERS

The most successful product of a manufacturer's mechanical committee (formed in 1918) was the development of the Tightlock coupler. An early version called Type T was improved and evolved into the Association of American Railroads Standard Type H Tightlock coupler, an alternate coupler designed for high-speed passenger cars and adopted for use on March 1, 1938 (figure 36). It improved on previous types by having less free slack and no vertical movement between adjoining couplers.

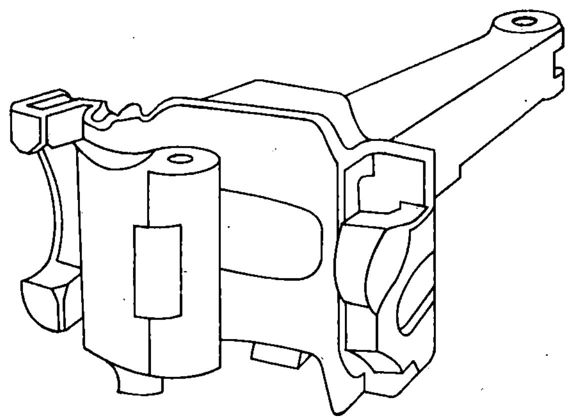

FIGURE 36. AAR STANDARD H TIGHTLOCK (1947)

The Tightlock coupler is provided with aligning wings and interlocking lugs at the sides of the headends, a feature to appear later on the Type F. All the important wear surfaces are machine finished, thus eliminating any problems of slack between mating surfaces. The coupler, knuckle, lock, yoke, and seat are of high tensile steel.

The significant advantages of the H Tightlock are summarized as follows:[30]

a. Practically assures that derailed cars will not overturn or telescope when a car with an H coupler is coupled to another car with an H, F, or Controlled Slack coupler. This safety feature has been proven by car actions during actual train derailments.

b. Effectively reduces car shocks and noise due to elimination of coupler contour slack.

c. Insures a more positive engagement when coupling at slow speed.

d. Results in a reduction of train slack by reducing the contour slack when coupled to cars with other types of couplers.

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[30] "Couplers, Coupler Parts, Coupler Yokes," Product Library, McConway & Torley Corporation, Pittsburgh, Pennsylvania, p. H-1.

e. Provides an increased lateral gathering range to facilitate making car couplings.

f. Provides the needed vertical and lateral angling for proper train operation when coupled to cars with other types of couplers.

g. All buffing stresses are taken through the central column of the H coupler.

h. Assures an extended coupler service life due to the reduction in wear afforded by the H design.

The Type CS (Controlled Slack) coupler incorporates the interlocking and free slack features of the Type H and Type E rigid couplers and the desirable features of its predecessor, the Type E Swivel Coupler, which provided excellent curve negotiability. It was adopted by AAR in 1956 for passenger service and has proven to be readily capable of direct substitution in existing cars.

The AMCOACH/AMCAFE and Amtrak Turboliner have been using the standard AAR "J" Tightlock Coupler which has the necessary gathering ranges, reduced free slack, the interlocking features, positive coupler engagement, and proven safety features that resist telescoping or overturning during a derailment. It is also possible to match up the Amfleet Type H and conventional couplers.

DEVELOPMENT OF TRAIN LINE CONNECTIONS

A train line may be electrical or pneumatic. A pneumatic train line is the complete line of air brake pipes in a train. These pipes include the rigid piping secured under the cars and the flexible connections between cars and locomotives. Miscouplings can cause safety problems as well as costly delays, and the process of making up a train is a time consuming process. For example, when air brake hoses are manually connected in the yards, a trainman has to straddle a rail and reach down below the mechanical couplers to complete the connection. During uncoupling, the air brake hoses are placed in tension as the cars separate, which causes the "glad hand" connections to angle and separate, and can eventually cause them to fail. Associated with the air brake hose is the air hose shut-off valve ("angle cock"), which also requires trainmen to reach over the mechanical coupler to activate.

Early tests on several different automatic air line and steam connectors were conducted at Purdue University in the 1930's under the auspices of the American Railway Association (ARA), predecessor of the AAR.[31] The test rigs checked the capacity of train air line connectors to gather at various offsets--vertical and lateral--under conditions similar to actual coupler operations. Figure 37 shows the Robinson wing connector, where the wings perform the gathering function. This passenger version features one steam and two air connections.

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[31] S. K. Punwani, "The Estimation of Potential Benefits from Improved Operation with Advanced Coupling Systems," 1974 Proceedings of 11th Annual Railroad Engineering Conference, pp. 61-67.

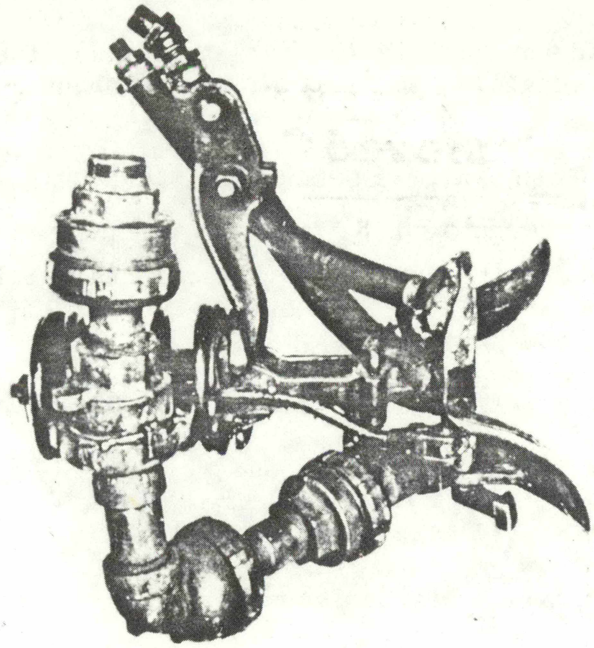

FIGURE 37. ROBINSON WING TYPE PASSENGER CAR CONNECTOR FOR STEAM AND TWO AIR CONNECTIONS

Figure 38 shows a Robinson connector, with the pin and funnel design in place of wings, for performing the gathering function on passenger cars.

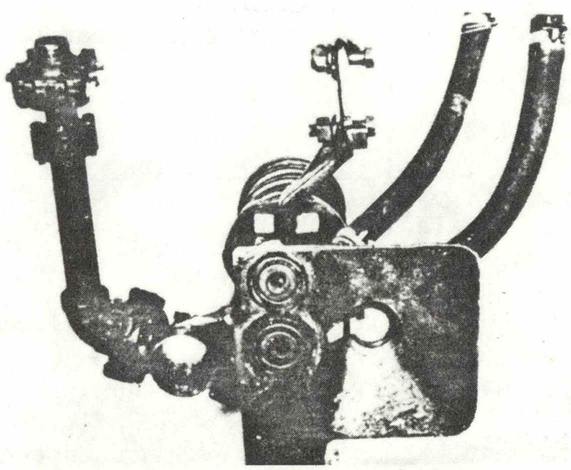

FIGURE 38. ROBINSON PIN AND FUNNEL PASSENGER CAR CONNECTOR

Many new concepts and designs have been advanced, and almost all the coupler manufacturers have made investments in anticipation of future developments. The technology is on hand to achieve any part of or all of the goals if there is economic justification.

The AAR/RPI (AAR/Railway Progress Institute) Advanced Coupling Concepts project delineated logical functional characteristics of each design for mechancial and train line air connections. These are discussed at the 11th Annual Railroad Engineers Conference in 1974,[32] and are shown in tables 4 and 5.

In addition to these basic characteristics, there are those that provide for auxiliary services, most of which come about only with an electrical train line.

TABLE 4. COUPLER MECHANICAL CHARACTERISTICS

Improved gathering range (lateral and vertical)
Always ready to couple
Coupling reliability (reduction in bypasses and no-couples)
Lower minimum coupling speed
Compatibility with present couplers
Reduced slack between couplers (longitudinal)
Reduced vertical misalignment
Reduced contour angling
Increased strength (vertical, longitudinal, and rotational)
Improved mechanical uncoupling means
Eliminate need to provide slack prior to uncoupling

TABLE 5. COUPLER AIR LINE CHARACTERISTICS

Make the physical connection between cars for train air line communication
Means to switch air on upon mechanical coupling
Means to switch air off upon mechanical uncoupling
Emergency brake application upon unintentional uncoupling
Compatibility with "glad-hands" and angle cocks

[32]Punwani, ibid.

Concepts relating to electrical train line systems have to deal with problems relating to automatic connection devices, contact capacities, automatic sequencing of connectors, and environmental protection of the connectors. Several concepts deal with compatible add-ons and a much lesser number of concepts concern train sensing and control systems. In rapid transit applications, the sensing systems are specifically used to assure proper coupling and braking conditions within each car. In fact, all of these concepts have been embodied successfully within the rapid transit service.

FULLY AUTOMATIC COUPLERS

The fully automatic coupler is one which makes electrical and pneumatic trainline connections at the same time the mechanical connection is made for buff and draft loads.

The fully automatic coupler was developed for and used in rapid transit operations for may years. Fully automatic couplers are also in service on main line commuter cars. Figure 39 gives three views of a rapid transit car coupler designed by Symington Wayne over 20 years ago.[33] This coupler may be used to illustrate the functions of a fully automatic coupler. The front view shows the two pivot points, one for horizontal angling at the rear, and one for vertical angling near the pulling face. The mechanical coupler is a hook-type with a flat face and interlocking pins. Electric contact boxes, shown attached to the sides of the mechanical coupler, could also be attached below or above the coupler if required.

The electro-pneumatic switching unit shown in figure 40 has a very important function in automatic coupling. It is required to control the make/break of the electric circuits, cut the air train lines in and out, and operate the coupler lock device. The switching unit works in conjunction with the electric coupler box (figure 41).

Figure 42 shows: a, two mechanical couplers fully coupled with electrical circuits made; b, the breaking of the electrical circuits and consequent closing of the air train line; c, actuation of the hooks of the mechanical coupler with the couplers free to be separated; and d, the cars separated and the hooks back to normal, ready to be recoupled.

The innovations in modern couplers give enough versatility to select options ranging from complete automation to those having pushbutton control, including provisions for an emergency release mechanism. Present day design techniques for control systems are applicable not only to passenger car service but to automate freight trains as well.

Fully automated couplers require design changes to improve reliability under all operating conditions. The metroliners operating in winter weather conditions of heavy rain and freezing temperatures have encountered electrical

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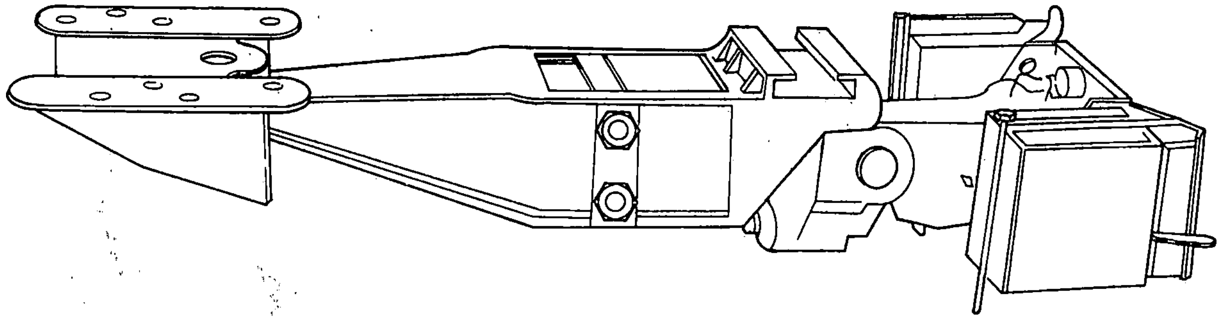
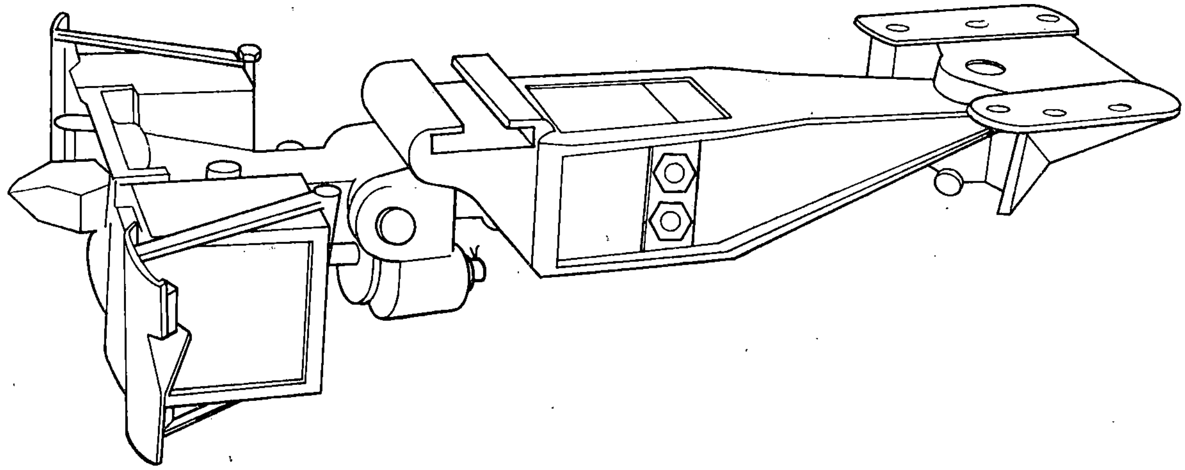
[33] Geoffrey W. Cope, "Development in Fully Automated Couplers and Potential on Automated Freight Cars," 1964 Proceedings of 1st Annual Railroad Engineering Conference, pp. 24-30.



RIGHT-HAND VIEW



LEFT-HAND VIEW

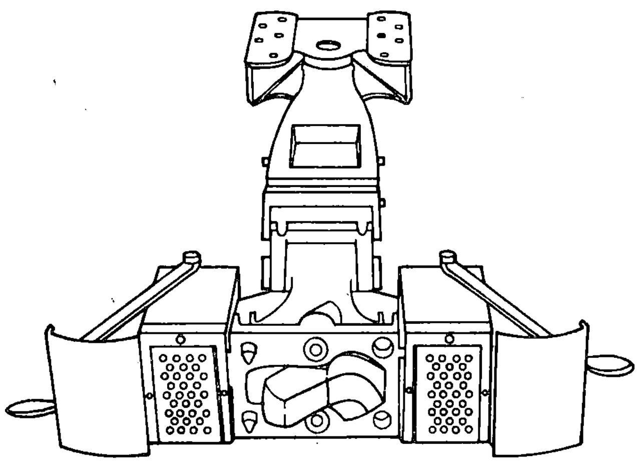


FIGURE 39. TRANSIT CAR COUPLER
(COURTESY OHIO BRASS COMPANY)

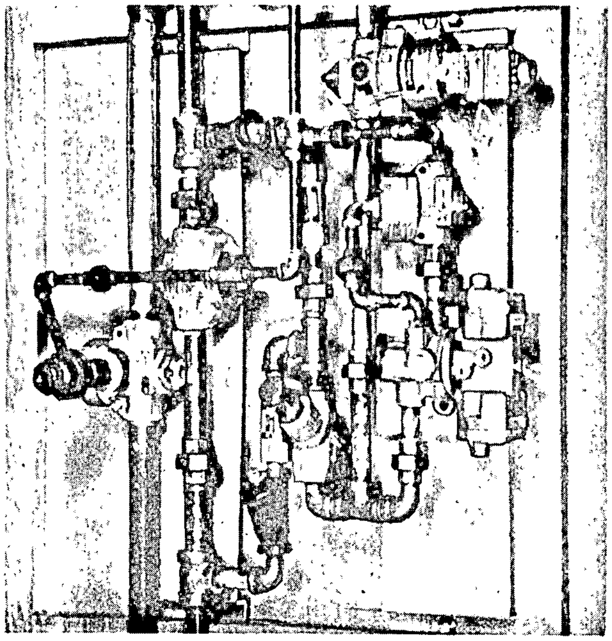


FIGURE 40. ELECTRO-PNEUMATIC SWITCHING UNIT

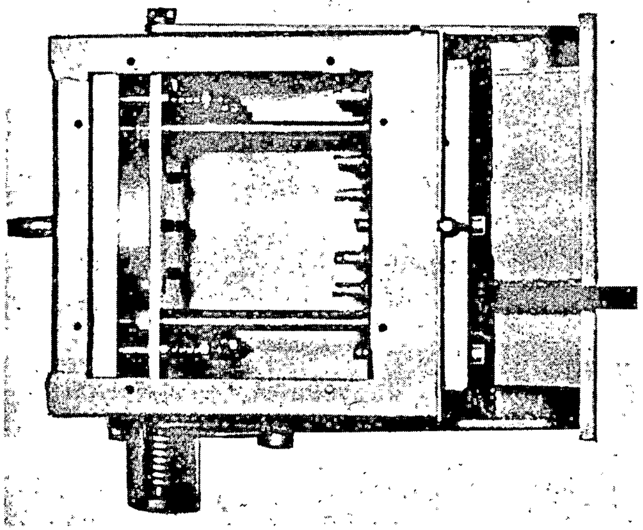


FIGURE 41. ELECTRIC COUPLER BOX

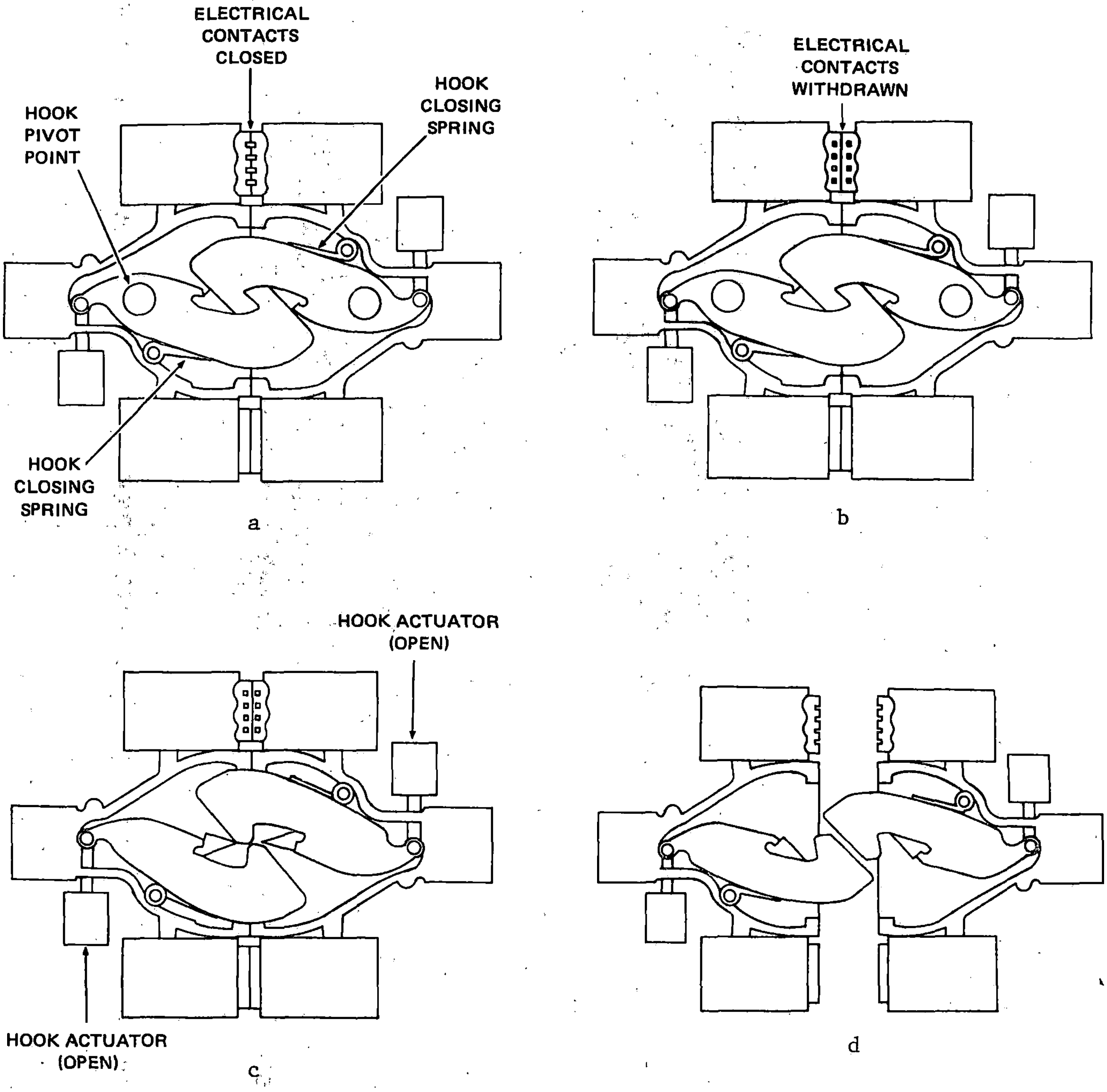


FIGURE 42. SYMINGTON-WAYNE COUPLER OPERATION

failure in the connector box below the coupler hook. The Symington-Wayne automatic coupler electrical junction box is now bypassed by a 27 point jumper cable. However, the air connections are still made automatically through the coupler.

The Metroliner train and commuter lines require frequent make-up and break-up of trains which causes wear and added maintenance problems with the automatic couplers. WABCO pin type electrical connectors have been broken on coupling and are not easily repaired. This caused them to be replaced with Ohio Brass button type connectors (figure 43). However, there is a recent trend toward the WABCO N-2/N-2-A (Westinghouse Air Brake Company mass transit type coupler) of rigid design with integral air lines and an electric head; it has the advantages of reduced slack, automatic air connection and an increased gathering range (figure 44).

Table 5 lists the coupler trend of changes on the Northeast Corridor and some connecting routes.

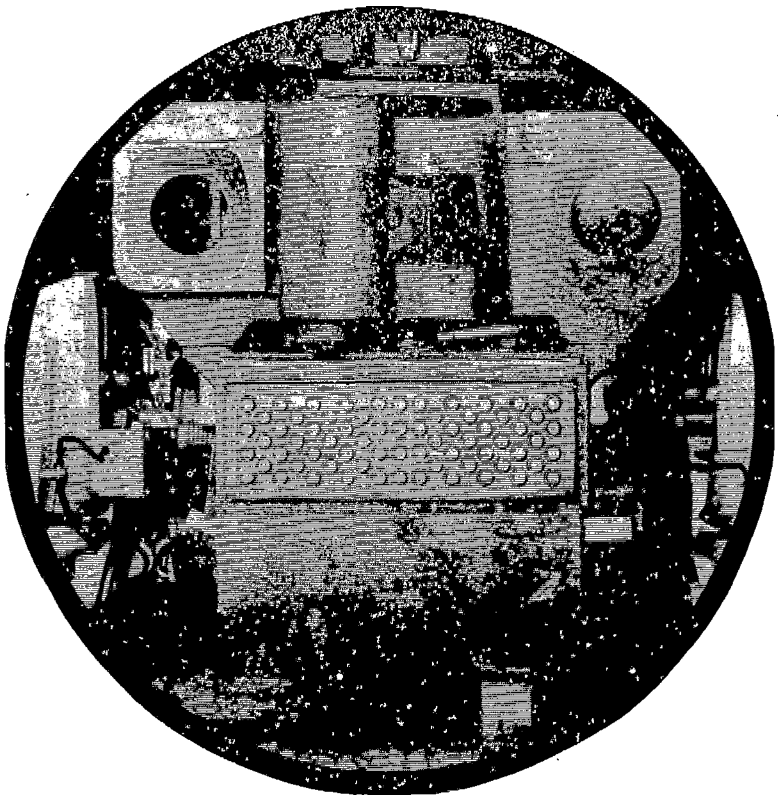


FIGURE 43. OHIO BRASS TRANSIT CAR COUPLER

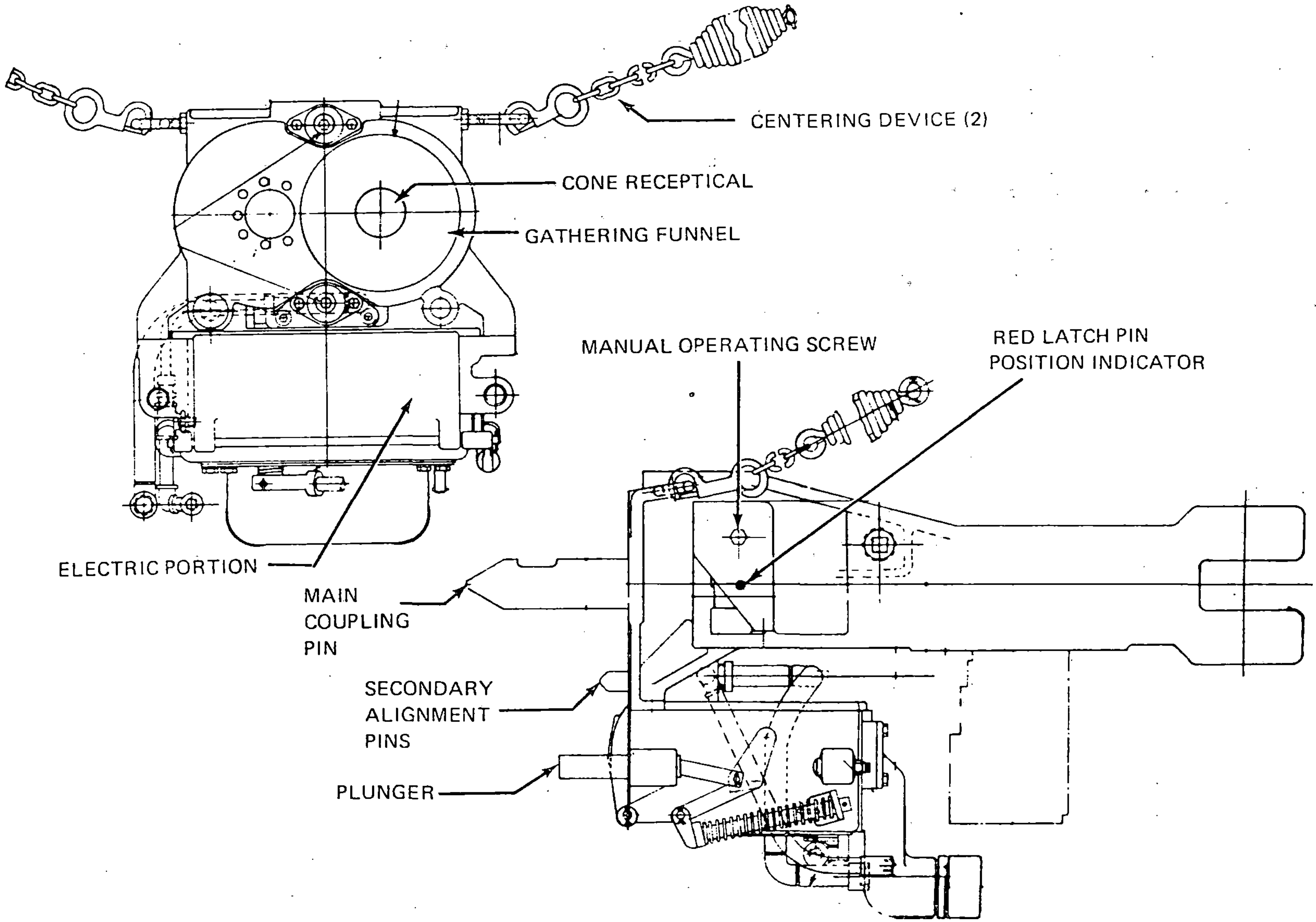


FIGURE 44. WABCO N-2-A RAPID TRANSIT AUTOMATIC CAR COUPLER
(COURTESY OHIO BRASS COMPANY)

TABLE 6

U.S. RAILROAD APPLICATION OF AUTOMATIC COUPLERS IN NEC AND CONNECTING ROUTES

Owner	Car	Coupler	Comment
Amtrak	Metroliner 61 cars	was: Symington/Walton electric now: Symington w/jumper cables	electric head replaced
SEPTA (Philadelphia area)	Silverliner I 5 cars	Ohio Brass	cars not compatible with newer cars
SEPTA (Philadelphia area)	Silverliner II 54 cars	was: Ohio Brass now: Wabco N-2	mechanical parts worn out, replacement required, changed to N-2 for standardization
SEPTA (Philadelphia area)	Silverliner III 20 cars	was: Ohio Brass now: Wabco N-2	
SEPTA (Philadelphia area)	Silverliner IV 232 cars	Wabco N-2	
MTA (Long Island)	M-1 950 cars	Wabco N-2	
MTA/CDOT (New Haven)	M-2 244 cars	Wabco N-2	
NJDOT	Jersey Arrow I 34 cars	was: Symington/Walton now: Wabco N-2	being replaced to standardize equipment
NJDOT	Jersey Arrow II 70 cars	Wabco N-2	
NJDOT	Jersey Arrow III 230 cars	Wabco N-2	

SECTION 6. THE NEW UIC (EUROPEAN) AUTOMATIC COUPLER

During the development of automatic couplers for use in European passenger trains it was decided to standardize the design of the automatic coupler to include those employed in freight service. The Western European members of UIC drew up the original specifications to replace the screw couplers, in 1960. These were later revised again to include the requirements of the railways of Eastern Europe, and the Russian SA-3 coupler.

The European effort was directed primarily on the Willison concept (figures 34 and 35) with rigid knuckle design which reduces wear and assures automatic connection of trainlines and the electric circuits. The three main groups of manufacturers, whose proposals were originally accepted, all offered designs embracing the Willison profile. The Unicupler group, headed by Knorr-Bremse of Germany, offered a wedge-shaped nose piece under the head with limited vertical movement. Bouralt-Sambre et Meuse (BSM) developed a very similar design which was later designated Unicupler/BSM before the design (and BSM) was absorbed into the International Eurocoupler Association. The Associated Willison group incorporated a wing pocket interlocking system instead of the horn/interlocking system used by the others. Later the International Eurocoupler Association provided the coupler design shown in figure 45, along with a transition coupler design to cover the period when center-coupler and screw-coupling stock would have to work together.[34]

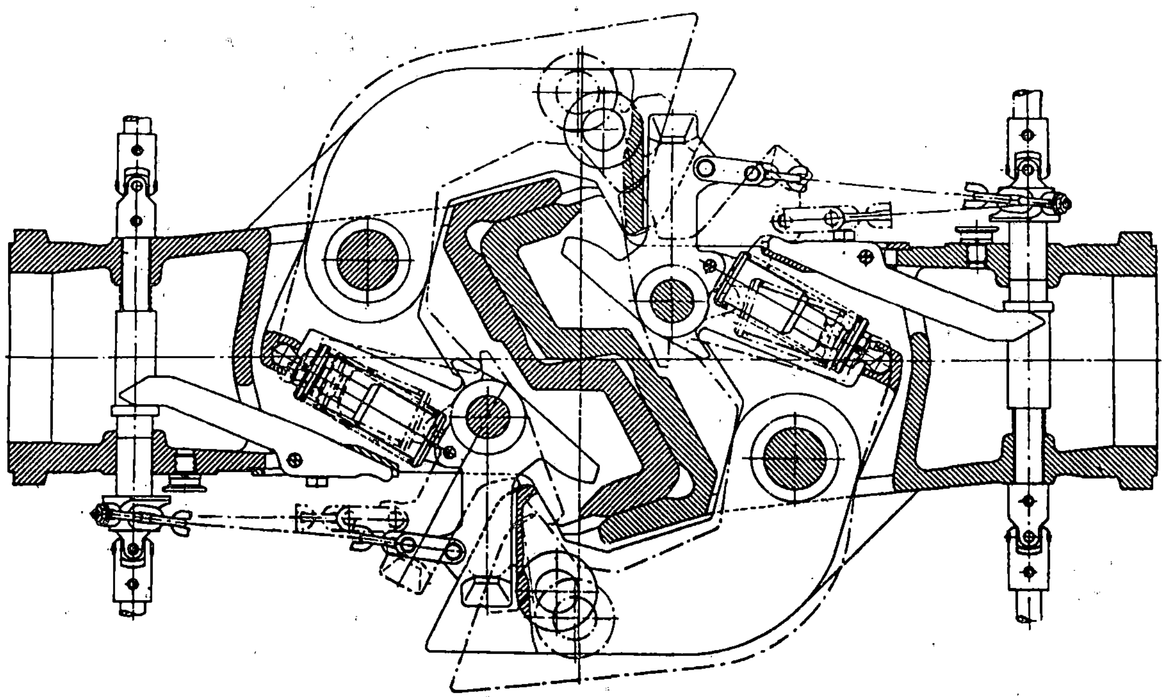


FIGURE 45. THE EUROCOUPLER CONCEPT

[34] "O.V.S. Bulleid, "Automatic Couplers and Compressed Air Brakes," Railway Gazette, April 26, 1963, Vol. 118, p. 469.

The final coupler selected by the UIC Automatic Coupler Committee is shown in figure 46. This version is supplied by Knorr-Bremse. It is interchangeable with couplers of other suppliers and embodies an electrical connector with 10 contacts and air pipes for steam and brake pressure. In addition, the mechanical coupling permits increased train lengths and higher buffing loads. For the future highly automated shunting processes can be developed because of the increased load bearing capacity.

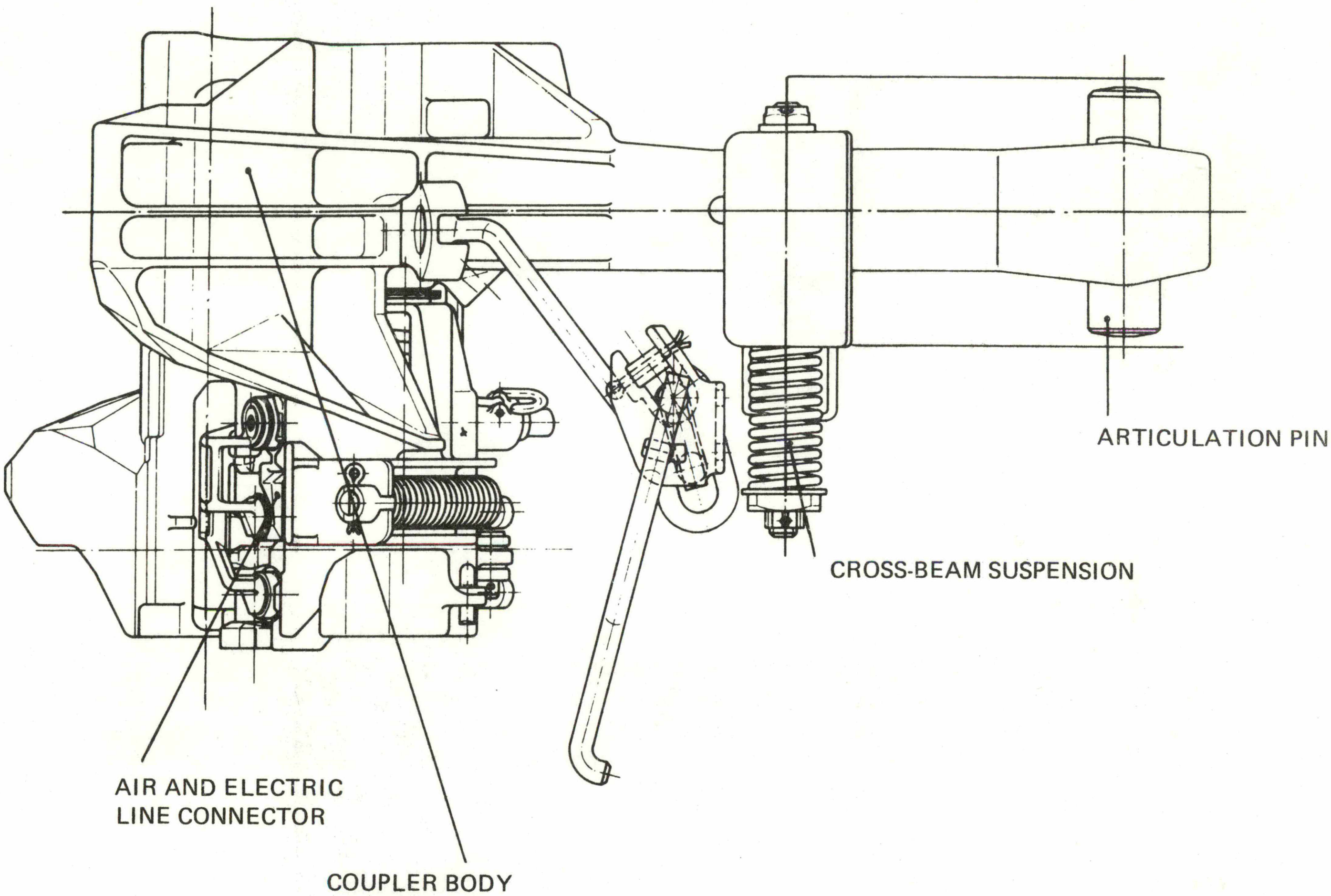


FIGURE 46. UIC AUTOMATIC COUPLER WITH CROSSBEAM SUSPENSION (COURTESY KNORR-BREMSE)

Two versions of the automatic coupler have been developed because of varying carbody structures for attachment. One, where carbody structure is available to support the heavy weight of the coupler with cross beam suspension, is shown in figure 46. The second, with telescopic leg suspension from below the coupler is shown in figure 47.

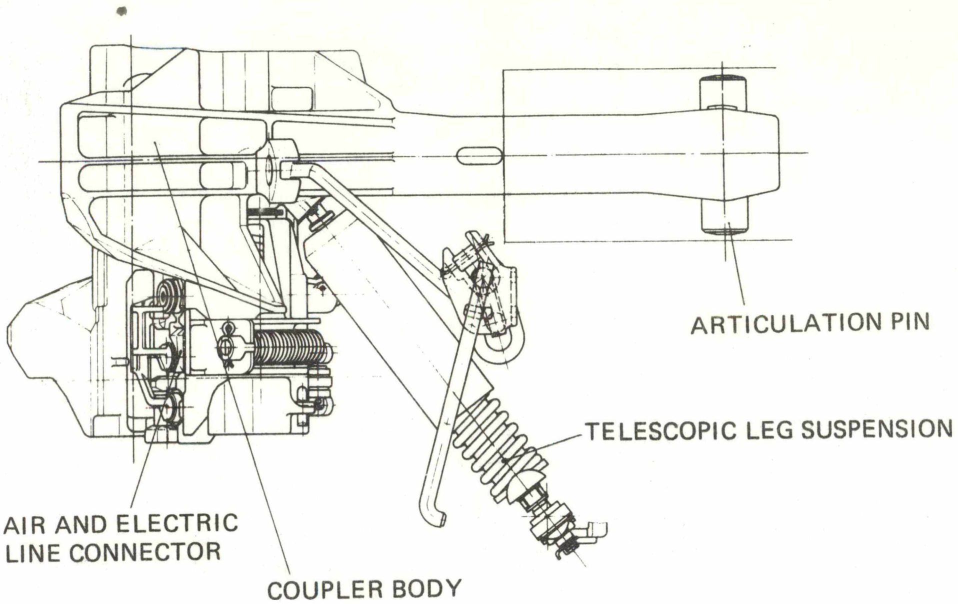


FIGURE 47. UIC AUTOMATIC COUPLER WITH TELESCOPIC LEG SUSPENSION (COURTESY KNORR-BREMSE)

The UIC69e automatic coupler is considered to have the best overall mechanical coupling features within its category (spread claw type). The spring-loaded hinge locking hook is considered to be an excellent locking mechanism. It is also considered to be a highly cost-effective locking design. The system also has the greatest potential for mechanical uncoupling. The addition of a rotary cam uncoupling concept would increase the potential for uncoupling in draft.[35]

The gathering features of the UIC coupler (figure 48) of ±8.7 inches laterally and ±5.5 inches vertically represent almost four times the gathering capability of the standard U.S. Type E coupler. This wide gathering range makes it possible to have just a centering device for effective coupling. This is a key characteristic of the UIC coupler. Another one is that it is always ready to couple.

The reduced free slack (about 0.3 inch) permits incorporation of air and electrical connectors in one block as long as they are cabable of self seating.

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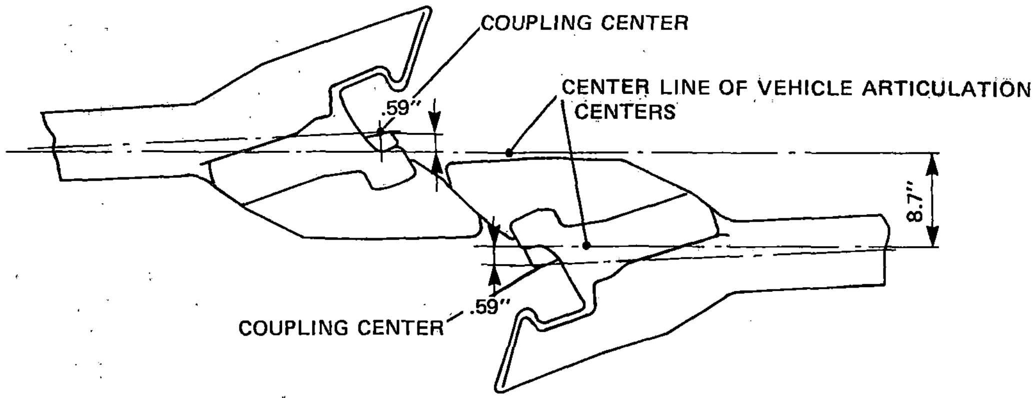
[35]"Coupling System Design Optimization," Report No. FRA-ORD-78-11, II, May 1978 p. 87.

HORIZONTAL GATHERING RANGE (inches)



VERTICAL GATHERING RANGE (inches)

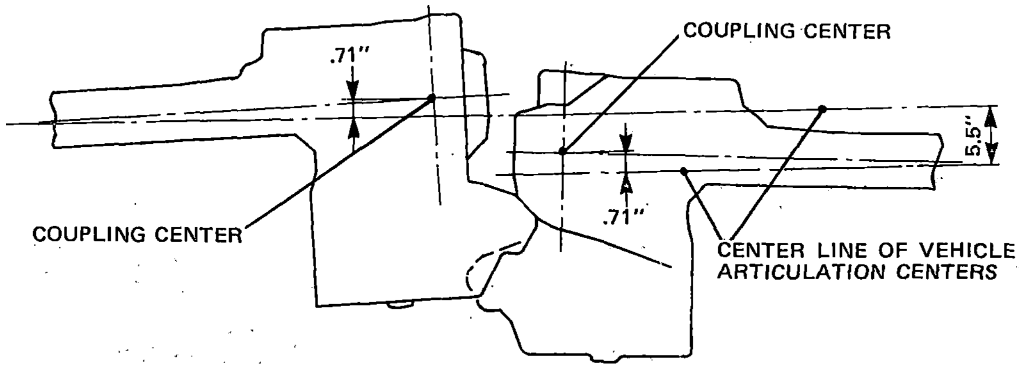


FIGURE 48. UIC COUPLING RANGE
(COURTESY KNORR-BREMSE)

The locking hooks achieve a positive snap when coupling. The hook location permits enough room for the air connection system on the lower center line of the coupler.

A unique feature of the UIC69e is the capability for a manual "locking impossible" (lock open) of the locking panels which prevents the coupler from locking upon contact.

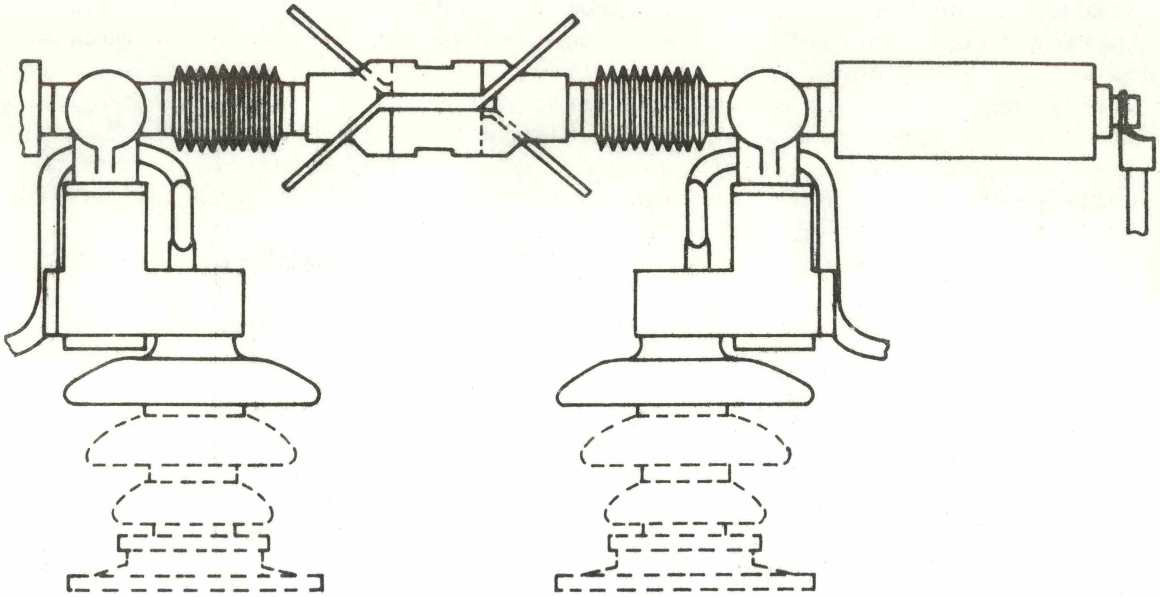
The automatic control of the main valves is a function of mechanical forces, which along with the automatic air connection, eliminates the requirement for anyone to go between cars during coupling or uncoupling operations.

It is significant to note that because of economic considerations, determinations for follow-on standard automatic couplers require railways to adapt their designs to those more prominently in use. Therefore, it is not surprising that the United States is considering modifications to the Janney Coupler while Europe has opted for a modification of the Willison profile. European railways (UIC/OSJD), however, have already absorbed the heavy costs related to make the transition.

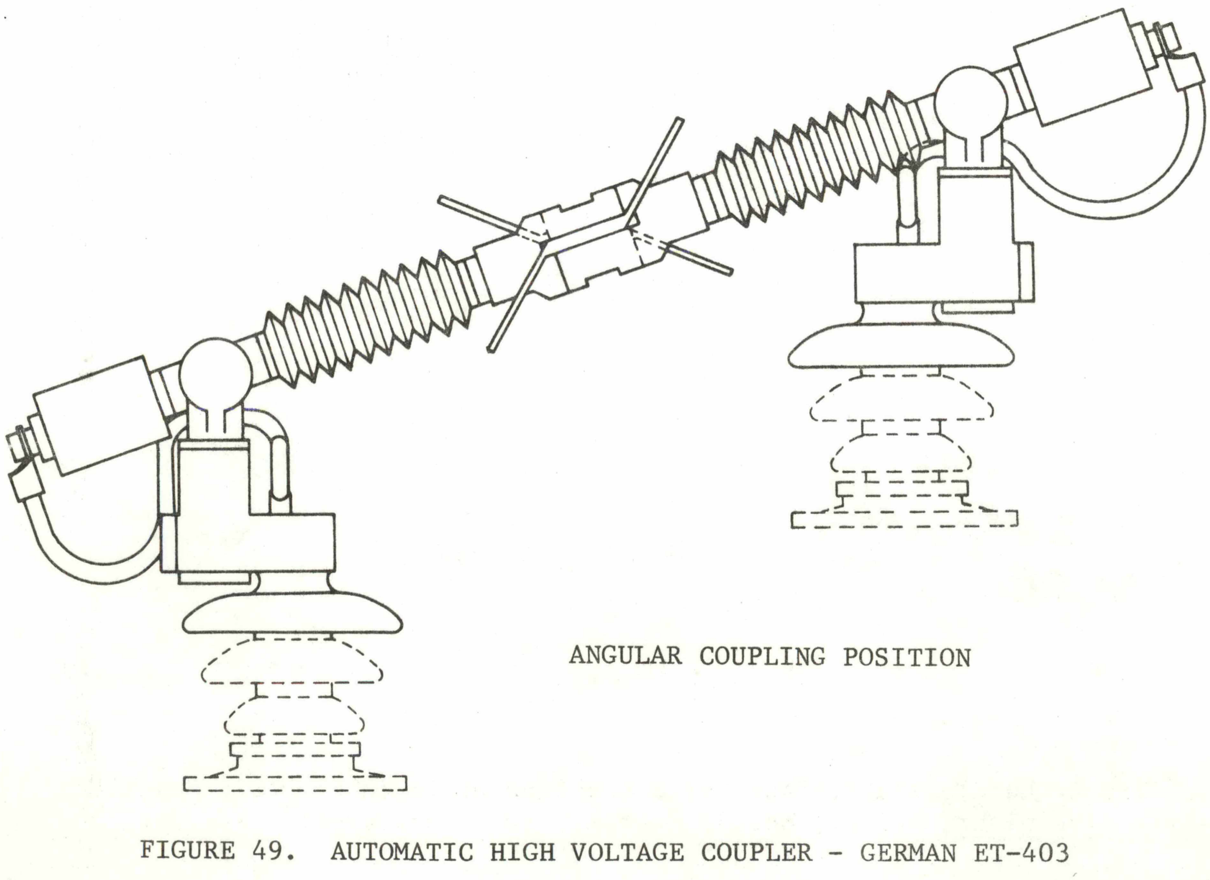
SECTION 7. AUTOMATIC HIGH VOLTAGE COUPLER

One device presently in use on the German ET403 high-speed train has aroused considerable interest. It is the use of high voltage bus couplers trainlined on the roof of the train from the rear car. It is automatic, has a gathering range, is spring loaded, and capable of trainlining as much as 15,000 volts at 700 amps. The high voltage bus runs down the middle of the car with capacity to make connections from car to car by swinging and locking the high voltage arms to one another. Figure 49 shows the two positions from the level and maximum angles of coupling. Due to the power limit from the 15 kilovolt 16-2/3 Hz catenary the high powered Multiple Unit (MU) car is limited to an eight car consist which has two pantographs. As a result of R & D work done at their Munich Research Center, the Germans developed the automatic coupling for electric bus-bars to meet the requirements for remote-controlled automatic coupling. The system was also tested for application to electric heating.[36]

[36] Dr. Arthur Tetzlaff, "Automatic Coupling of Train Electric Bus-Bars," Eisenbahningenieur, Vol. 27, No. 2, February 1976, pp. 59-61.



LEVEL COUPLING POSITION



ANGULAR COUPLING POSITION

FIGURE 49. AUTOMATIC HIGH VOLTAGE COUPLER - GERMAN ET-403