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EVALUATION OF THE COSTS AND BENEFITS OF ADVANCED BRAKING AND COUPLING SYSTEMS

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16. Abstract

This report presents an evaluation of the costs and benefits of sixteen advanced railroad braking and coupling systems. Most of the benefits result from improved classification yard efficiencies, with secondary benefits accruing through reduced accident rates, road delays, and maintenance related to component wear and failure. The most promising systems are couplers with wide gathering ranges, a brake condition monitoring system, and a remote controlled brake locking system. In addition, ultrasonic brake control on cars presently requiring special handling and direct electronic brake control all show promise of improving railroad productivity.

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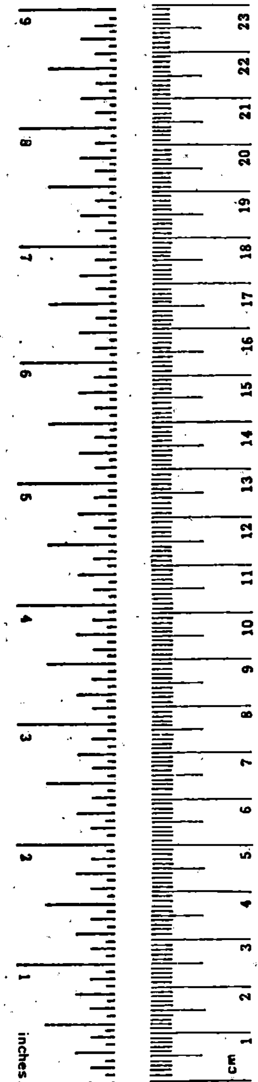
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

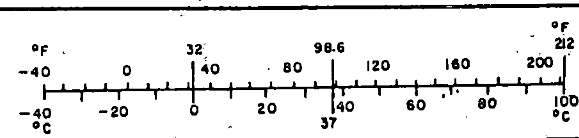
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	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



PREFACE

This is the second in a three-volume series of reports on advanced braking and coupling systems. The first, "Methodology for Evaluating the Cost and Benefit of Advanced Braking and Coupling Systems [1], established the techniques that were intended for use in evaluating a broad range of candidate systems. In this report, those techniques are applied to 16 such systems. Some systems, such as electrical command, control, and monitoring of coupler and brake systems, show promise of significant financial payoff. For others, such as disk brakes or automatic air line connectors, the technology does not appear sufficiently well developed to hold much promise for the near term. The third report, "Recommendations for Research and Development on Advanced Railroad Braking and Coupling Systems" [2], deals with those systems that are presently attractive and also with needed improvements in analytical techniques and experimental data.

The authors express their appreciation to the people and organizations that have helped considerably throughout this project. The FRA COTRs, Ms. Marilynne Jacobs and subsequently Dr. N. Thomas Tsai, have provided invaluable guidance and direction. In addition, an industry committee composed of Messrs. Geoffrey Cope of Dresser Industries, John Punwani of the Association of American Railroads, Bruce Shute of the New York Air Brake Co., Donald Whitney of the Burlington Northern Railroad, and Carl Wright of Westinghouse Air Brake Co. have performed important review and consultation. The American railroad industry, in particular the Southern Railway, Boston and Maine, Conrail, and several other railroads, has graciously provided information and an opportunity to observe railroad operations. The Union Internationale Des Chemins De Fer provided valuable information on the design and costs of a coupler proposed for eventual use on European railroads.

EXECUTIVE SUMMARY

The purpose of the study reported here has been to determine a future course of R&D for railroad braking and coupling systems. The motivation for the study has been a desire to develop systems that will improve railroad efficiency, productivity and, concomitantly, profitability. Emphasis is therefore placed on evaluating, strictly in monetary terms, the costs and benefits of a broad range of possible systems. Underlying much of the study is the recognition that recent advances in electronics, communication, and control technologies could be applied to braking and coupling systems with dramatically beneficial results. Indeed, combined benefits on the order of a billion dollars a year may ultimately be realized.

It should be recognized that often considerable uncertainty is embodied in the results of a study such as this. Many of the systems evaluated exist only at the conceptual level and are costed in a preliminary way. Furthermore, quantification of benefits in financial terms is made difficult by the enormous complexity of the U.S. railroad network, and the absence of readily available data needed for certain analyses. Accordingly, results should be viewed as reasonable indicators of future directions for R&D, but not viewed as having greater precision than warranted by supporting data and computational procedures.

The systems investigated in this report and a brief description of their features are presented below. They are classified as mechanical or electrical and according to whether their principal function is to improve operating efficiencies (mainly in yards) or the dynamics of train performance (mainly over the road).

Mechanical Systems - Improved Operations

1. Wide-range couplers - Coupler systems that are designed to ensure that coupling takes place when two cars are brought together. Examples are knuckle-open devices, guard arm extensions, and centering and positioning devices.

2. Automatic air line connector - Automatically connects train air line during mechanical coupling. Includes optional feature of closing airline when deliberately uncoupled.

3. Incompatible coupler - A mechanical coupler that is incompatible with present knuckle coupler and that could include integral air and/or electrical connector. Examples include the Willison spread-claw, the flat-face hook, and the pin and funnel.

Mechanical Systems - Improved Dynamics

4. Truck-mounted brakes - Brake cylinders are mounted on trucks rather than car body. Examples are WABCOPAC and NYCOPAC.

5. Disk brakes - An axle-mounted disk braking surface and brake pad application assembly to supplement or replace conventional tread brakes.

6. E Couplers with shelves - Provide interlocking shelves on standard E coupler to prevent vertical disengagement.

7. High-strength draw gear - Couplers, knuckles, and yokes manufactured from high strength steel to mitigate failure under repeated heavy loads.

8. Zero slack systems - Couplers and draft gear with no slack to minimize run-out and run-in forces.

9. Mechanical load sensor - Senses the weight of the car pneumatically or mechanically and allows the brake system to generate a force that is either heavy or light (load/empty) or is proportional to the weight of the car (load proportional).

Electrical Systems - Improved Operations

10. Remote-controlled coupler - A uniquely addressable, pneumatically actuated, coupler that can be opened by a signal transmitted from the locomotive. Includes optional feature of automatically closing air line when activated.

11. Remote-controlled brake lock - Uniquely addressable car brake

that can be set and released from the locomotive. Can include a mechanical device to prevent undesired release caused by gradual air leakage.

12. Ultrasonic brake control - A car-mounted braking system incorporating an ultrasonic sensor and electronically actuated brakes for controlling the speed of a freely rolling car before impact and coupling with another car.

13. Train condition monitor - Adaptable electrical system that allows several variables, such as angle cock position, brake pipe pressure, or brake piston travel, to be monitored and transmitted to the locomotive or other station.

Electrical Systems - Improved Dynamics

14. Electronic brakes - A brake system in which an electronic control module on each car is activated by a signal transmitted from a locomotive or other remote station, or by a sensor in the brake pipe. The module activates a brake control valve to apply or release brakes.

15. Electrical load sensor - Senses the weight of the car electrically and allows the brake system to generate a force that is proportional to the weight of the car.

16. Electro-pneumatic brakes - A system in which application and release of brakes on each car is controlled through electro-pneumatic valves that are energized by current taken from contacts on the motorman's brake valve and continuous train wires [3].

The procedure, or methodology, for evaluating the costs and benefits of each candidate system identified above is shown schematically in Figure E.1. The top portion of the figure illustrates the estimation of incremental equipment costs per car. The bottom portion shows the estimation of maximum allowable costs per car, based on the financial benefits that each candidate system is likely to generate. The maximum allowable cost is the largest investment per car that a railroad could justify in order to realize an overall return corresponding to the net benefits provided by the system and to the railroad's required rate of return on investments of this type.

To estimate system costs, a conceptual design is first developed in enough detail to identify significant components and establish such relevant parameters as size and load capacity. The cost analysis involves a synthesis of manufacturers' quotes for off-the-shelf component prices and engineering estimates for unavailable components, assembly, and installation. Maintenance costs are estimated as a fraction of the incremental equipment costs per car.

To estimate the allowable cost per car for a candidate system, the gross benefits that the system will provide are first evaluated. Reduced operating costs are determined primarily from a classification yard simulation model developed for this project [1]. Reduced accident and maintenance cost estimates are also based on statistical data assembled earlier for this purpose [1]. Subtracting annual maintenance costs from estimates of gross benefits yields an estimate of net benefits which provides the input to the financial analysis portion of the methodology. This analysis uses a net present value

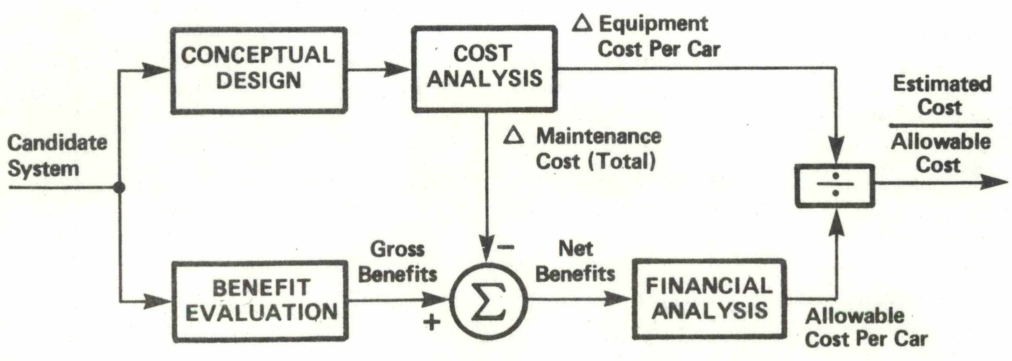


FIGURE E.1. OVERVIEW OF METHODOLOGY

technique to calculate the worth of a candidate system. The analysis is based on a specified implementation scenario in which the future stream of costs and benefits is to be realized, and on such financial parameters as discount rate, depreciation schedule, and tax rate.

The resulting cost/benefit figure of merit is expressed as the ratio of the estimated cost to the allowable cost. When this ratio is less than unity, it indicates that the system is favorable; when it is greater than unity the return on investment is insufficient to justify its cost.

The results of this study are summarized in Table E.1. For each system the table identifies the report section in which the evaluation is presented and several of the major variables corresponding to Fig. E.1. In addition, the difference between the allowable and estimated costs per car is given to indicate the magnitude of the potential savings (or loss) per car.

The results of this study lead to several important conclusions. Of the nine mechanical systems evaluated only wide-range couplers and high strength knuckles appear promising. The coupler could have a particularly large impact, primarily because it speeds the assembly of trains in classification yards.

Most of the electrical systems show promise of substantial financial benefits. The connectors and wiring, shown first, are the framework upon which other systems are to be built. Remote controlled couplers are very attractive <u>if</u> their implementation is accompanied <u>by</u> a crew-size reduction of one member. Remote controlled brake locks replace the use of hand brakes in

TABLE E.1. SUMMARY OF SYSTEM EVALUATION

Report Section	System	Net Benefits ($M)	Allowable Cost Per Car ($)	Estimated Cost Per Car ($)	Per Car Savings (Loss) ($)	Cost/Benefit Ratio:† Estimated Cost / Allowable Cost
	Mechanical: Improved Operations					
2.2.1	Wide range couplers	503	2157	874	1283	0.33
2.2.2	Automatic airline connector	101	318	765	(447)	2.4
2.2.3	Incompatible coupler	597	1717	10,248	(8531)	5.97
	Mechanical: Improved Dynamics					
2.3.1	Truck-mounted brakes	*	*	*	*	*
2.3.2	Disk brakes	*	*	11,700	*	*
2.3.3	E couplers with shelves	12	58	112	(54)	1.9
2.3.4	High strength draw gear					
	knuckle	18	20.71	8.90	11.81	0.43
	coupler body	35	6.43	15.25	(8.82)	2.37
	yoke	13	1.77	5.75	(3.98)	3.25
2.3.5	Zero slack systems	31	91	*	*	*
2.3.6	Mechanical load sensor	38	51	405	(354)	7.94
	Electrical: Improved Operations					
3.1	System Framework	0	0	135	(135)	*
3.2.1	Remote controlled coupler:					
	a) time savings only	31	87	1,060	(973)	12.2
	b) crew size reduction	493	1373	1,060	313	0.77
3.2.2	Remote controlled brake lock	703	1957	346	1611	0.18
3.2.3	Ultrasonic brake control (on 5% of cars)	198	5340	2,000	3340	0.37
3.2.4	Train condition monitor	479	1334	221	1113	0.17
	Electrical: Improved Dynamics					
3.3.1	Electronic brakes (direct control)	*	1275	917	358	0.72
3.3.2	Electrical load sensor	54	73	120	(47)	1.6
3.3.3	Electro-pneumatic brakes	(300)	*	6,225	*	*

*It was either unfeasible or inappropriate to estimate values for these elements. Further discussion is provided in the report sections indicated in the left-hand column of the table.

†The smaller the ratio in this column, the more attractive the system.

many situations and can result in significant annual savings. The ultrasonic brake control system, applied to cars requiring special handling (about 5% of the fleet) can save significant classification time and an undetermined amount of loading damage. The train condition monitor, by speeding the power brake test, could potentially save several hundred million dollars annually in labor and equipment utilization.

Of the remaining electrical systems (those designed to improve dynamics), only the electronic brake system shows much potential. It can probably be made for less cost than the conventional pneumatic system. Even at $120 per car set, electronic load sensors do not appear ready for fleet-wide use but may be appropriate for cars with unusually high gross/tare weight ratios. Passenger type dual air line pneumatic brakes are too expensive, and sufficiently incompatible with freight brake systems to consider further.

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1. INTRODUCTION

Whether an advanced railroad component or system is suitable for implementation, development, or research depends on whether it shows sufficient promise to be cost-effective. Determining this involves a multi-stage process. First, a realistic assessment must be made of the future stream of anticipated costs and benefits. Costs are usually assessed in terms of monetary outlays, but benefits will tend to appear as reduced time for crews to perform certain functions, greater safety of operation, or extended component life times. To facilitate a comparison of costs and benefits, these factors must all be reduced to the same financial basis. However, these costs and benefits occur unevenly over time (costs invariably precede benefits) and must also be reduced to common terms through a net present value technique to discount future cash flows. Once this has been done, costs and benefits can be compared as a ratio which provides a measure of the attractiveness of a particular candidate system.

The methodology for evaluating the costs and benefits of alternative systems is described in some detail in a companion report [1]. Briefly, it involves the following four major steps that will be applied here to each system:

- Definition of an implementation scenario
- Estimation of equipment and maintenance costs
- Estimation of gross monetary benefits
- Determination of a cost/benefit ratio

A quantitative implementation scenario is needed to identify the future anticipated stream of costs and benefits. Equipment costs are always incurred before benefits start to accrue. In some cases, there are virtually no benefits until a system is completely converted. An automatic airline connector or an electrical train line for cars used in interchange service are examples. Since the present values of costs and benefits depend on when they occur, it is necessary to project their implementation at the start of the analysis.

The two principal costs associated with an advanced system are for equipment and maintenance. Equipment cost includes the purchase of hardware and its installation on a car or locomotive. For this study maintenance includes scheduled and unscheduled service as well as replacement of worn parts. Other costs, such as R&D and inventory, are assumed to be minor in comparison to equipment and maintenance costs.

Gross benefits are determined by calculating the annual monetary value of all benefits brought about by a given system. These generally include labor and equipment time saved while classifying cars, reduced accident frequency and severity, and reduced component fatigue and wear.

For purposes of this study, the cost/benefit ratio is the estimated per car cost of a system divided by the allowable cost per car. The allowable cost is estimated by first calculating the net benefit (gross benefit minus maintenance cost) and then computing the maximum equipment cost that the railroads could incur to realize these net benefits. A cost/benefit ratio (C/B) of unity implies that the investment would just pay off, not as a break even situation, but one in which they would realize the required return on investment. A cost/benefit ratio of greater than one not only implies that the investment is presently unattractive, but indicates the degree by which costs would have to be reduced to make the system viable. If, for example, C/B = 2.0, a 50% reduction in equipment costs would make the system economically justifiable. Such a number may be used as an R&D target for those systems which appear desirable but which are not yet cost effective. Conversely, a system with a C/B value less than unity is immediately attractive for research, development or implementation, depending on its state of development.

It should be clear from the outset that uncertainty is inherent in the process of estimating costs and benefits and that the results of this study will embody these uncertainties. Some of the sensitivities of results to variations in parameters will be treated in Sec. 2. Beyond this, however, it should be recognized that there are a great variety of operational procedures and conditions that cannot realistically be taken into account in a study as broad as this. By keeping these caveats in mind, the reader will not look for more precision than the data warrant.

The remainder of this report is divided into two sections. Sec. 2 deals with the evaluation of mechanical systems, while Sec. 3 deals with electrical systems.

2. MECHANICAL SYSTEM EVALUATION

2.1 Common Assumptions and Data

Underlying most of the system and component cost and benefit analyses that follow this subsection are a common set of assumptions and data. It will be useful at the outset to identify these commonalities explicitly, and, in certain cases, to show the sensitivity of results to variations in key parameters.

Financial Parameters

The financial analyses of many systems incorporate the following two assumptions:

- Ten years are required to convert the U.S. rail fleet to a new system.
- A 20% internal rate of return (IRR) is required to justify an investment, while the inflation rate will run at 10% per year.

The 10 year conversion figure is a consensus among the industry consultants advising this project. It implies the stream of costs and benefits illustrated in Figure 1. Initially, a substantial investment is made to convert the fleet, during which little or no benefits accrue. After the 10 year period, benefits begin and continue at a constant level, while costs drop to those needed to equip new cars only.

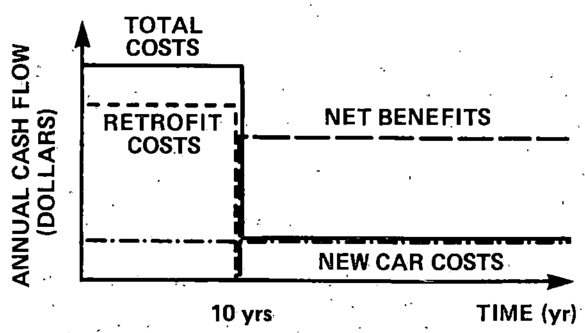


FIGURE 1. STREAM OF COSTS AND BENEFITS

The 20% IRR figure is used as a minimum threshold for capital investment decisions by several railroads contacted during the study. The assumed 20% significantly discounts the present value of future benefits. Accounting for both the 20% IRR and 10% inflation (i.e. using a net discount rate of 10%)reduces the present value of a dollar earned 10 years hence to about $0.39. Accordingly, a lengthy conversion period can adversely affect projected cost/benefit ratios.

Fig. 2 shows the relationship between funds available to retrofit a car and IRR for several implementation periods. The results are clearly sensitive to both parameters, particularly the IRR. Demanding an IRR of 15% instead of 20% increases the funds available to equip a car by 74%. Reducing the conversion period to 5 years increases available funds by 29%,while increasing the period to 15 years reduces funds by 25%.

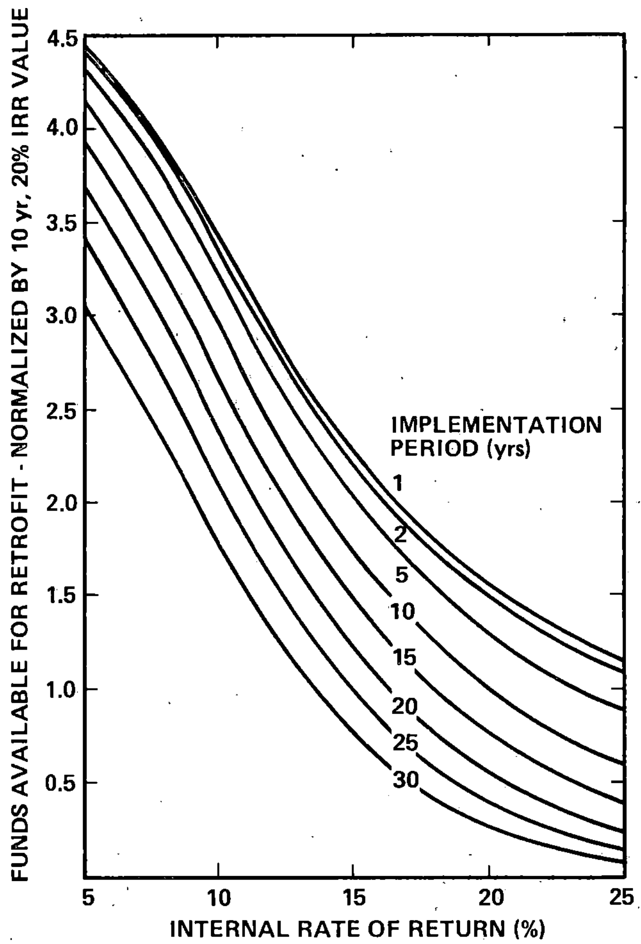


FIGURE 2. INTERNAL RATE OF RETURN (%)

Escalation factors

Cost data used in this study are acquired from a variety of sources, prepared and published at different times. In view of the inflation that has characterized recent years, it is necessary to adjust all costs to a common point in time. Table 1 shows the

Producer Price Index determined by the Bureau of Census for railroad equipment, and the inflation factor that we will use to escalate cost data to the base year 1979. The inflation factor for any year is the ratio of the Producer Price Index for 1979 to the Producer Price Index for that year.

TABLE 1. INFLATION FACTORS FOR ESCALATING PRICE DATA TO 1979

	Producer Price Index for Railroad Equipment[1]	Inflation Factor[2]
1965	97.4	2.88
1970	115.1	2.44
1971	121.1	2.32
1972	128.7	2.18
1973	134.7	2.08
1974	163.8	1.71
1975	201.2	1.39
1976	216.7	1.29
1977	233.5	1.20
1978	252.8	1.11
1979	280.6	1.00

[1] *Statistical Abstract of the United States, 1978,* U.S. Dept. of Commerce, p. 486.

[2] The number by which price data are multiplied to scale them to 1979 levels.

Maintenance costs

Maintenance cost data are not available for many of the advanced braking and coupling systems under consideration. In some cases prototype hardware has been developed but never entered into service. For those prototypes that have received limited service testing, maintenance data are not particularly relevant, since prototypes generally require a great deal more attention than volume production systems. In most cases, the advanced systems that we evaluate exist in conceptual form only and have not been reduced to practice or tested. Nevertheless, maintenance costs can be an important part of equipment life cycle costs and need to be accounted for. To do this, we look to existing equipment to establish maintenance costs as a fraction of new equipment costs, and then extrapolate the results to advanced systems.

Brake valves require scheduled maintenance and also receive unscheduled maintenance when problems arise. The ABDW valve requires a clean, oil, test, and stencil (COT&S) service at 14 year intervals. The AAR establishes standard costs for this service in its Office Manual of Interchange Rules (Job Code 1050) as

Labor (7.003 hrs)	$196.50
Materials(new grade)	87.34
Total	$283.84

Thus the average annual scheduled maintenance cost is $283.84/14 = $20.27.

A study was conducted by the New York Air Brake Co. of 20,000 cars equipped with ABD valves to determine the frequency with which the valves were removed for unscheduled service [4]. Reasons for such service include contamination, wrecks, flood, and fire damage. They estimated 19% of valves required unscheduled maintenance during what was, at that time, a 12 year normal service interval. Assuming that a full COT&S would be performed, the annual costs for this maintenance would be $283.84 x 0.19/12 = $4.49 per year.

Combining the above maintenance cost figures and dividing by the OEM brake control valve price of $1275 results in the following maintenance cost ratio:

Brake control valve maintenance ratio = 1.94%

Coupler maintenance costs may be evaluated from AAR car repair billing data that have been reported for couplers, knuckles, and yokes, and from standard labor and materials rates used by AAR member railroads for costing and billing purposes. These data are shown in Table 2 for cases in which new or reconditioned materials are used. The total costs in 1978 dollars corresponding to the number of repairs made in 1972 would be $90.93 M* if only new materials were used in repairs and $37.44 M if only reconditioned materials were used. Escalating these figures to 1979 (see Table 1) gives

* Throughout this report we use M for millions of dollars

TABLE 2. COST DATA FOR COUPLER, KNUCKLE, AND YOKE REPAIRS

Component (Job Code)	Labor Cost ($)[1]	Material Cost ($)[1]	Scrap Credit ($)[1]	No. Repaired Annually[2]	Total Cost ($m)
Couplers (2022)	26.43	217.37 (new) 76.13 (recond)	<9.00>	227,000	53.30 (new) 21.24 (recond)
Knuckles (2052)	3.51	47.48 (new) 16.62 (recond)	<2.37>	498,000	24.21 (new) 8.84 (recond)
Yokes (2314)	57.88	127.87 (new) 45.92 (recond)	<4.38>	74,000	13.42 (new) 7.36 (recond)

[1]AAR Office Manual of Interchange Rules, data are for October 1978.

[2]Ref. 1, Table 24, data for calendar year 1972.

Coupler, Knuckle, and Yoke Maintenance Cost

New Material: $100.9 M
Reconditioned Material: $ 41.6 M

In 1972, there were 1,717,000 cars in the fleet. The following are the 1979 draw gear component prices:

E60 CHT Coupler	$320
Y40 AHT Yoke	150
Total	$470

The total coupler and yoke investment (in 1979 dollars) in the 1972 fleet was 470 x 2 x 1,717,000 = $1.614 billion. Accordingly, maintenance costs, as percentage of this investment are:

Coupler maintenance ratio = 2.5% to 6.1%.

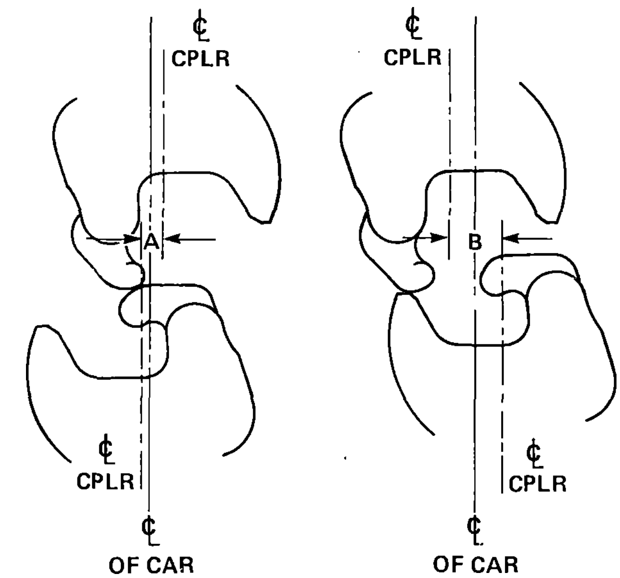
In summary, annual maintenance costs for braking and coupling systems appear to range from about 2-6% of component purchase price. We shall use an intermediate value of 4% for mechanical systems in subsequent analyses.

2.2 Improved Operations: Mechanical and Air Coupling

2.2.1 Wide Range Couplers

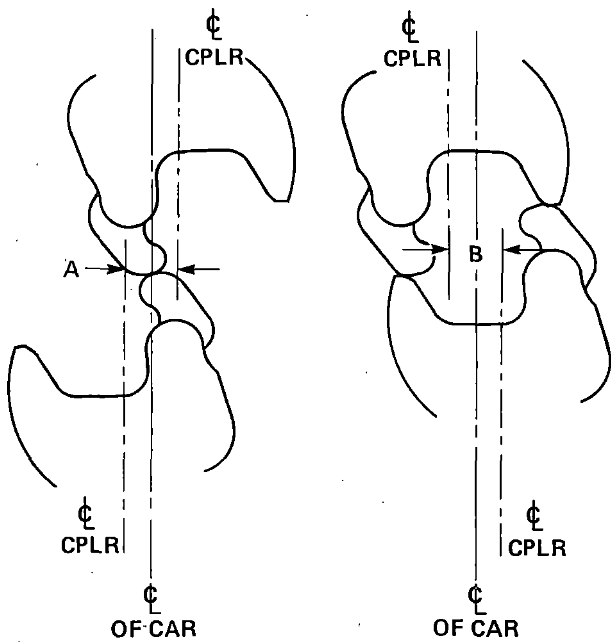
Several devices have been developed to increase the probability that cars will successfully couple. Included are knuckle open devices, guard arm extensions on E-type couplers, and positioning devices for long shank couplers. In analyzing these systems, we have assumed that the probability of coupling would have to be increased to nearly unity before there would be any significant savings, and therefore we have lumped these items together. The major benefit of a very high probability of coupling is that walking the classification tracks to supervise couplings during the pull down operation could be eliminated. This would save costs, both in labor and equipment time. Other benefits include less equipment damage, and elimination of the "cross over open knuckle" tasks in some flat switching.

Present E and F couplers have a relatively small gathering range. Figure 3 presents some data on the gathering range of E and F couplers for the situation where the center lines of the couplers are kept parallel [5]. This gathering range is reduced when the center lines are not parallel. For the parallel centerline case the total gathering range of an E coupler is 4-3/4 inches with one knuckle open and 7-1/2 inches with both knuckles open. The parallel centerline values for an F coupler are 6 inches with one knuckle open, and 8-3/4 inches with both knuckles open. These gathering range values are probably a bit conservative; that is, the actual gathering range may be slightly greater.



ONE KNUCKLE OPEN

	A	B	Total Gathering Range (A+B)
E	1¼ in.	3½ in.	4¾ in.
F	1¼ in.	4¾ in.	6 in.



BOTH KNUCKLES OPEN

	A	B	Total Gathering Range (A+B)
E	4 in.	3½ in.	7½ in.
F	4 in.	4¾ in.	8¾ in.

FIGURE 3. COUPLER GATHERING RANGES [5]

Description of systems

Knuckle Open Device - It is clear from Fig 3 that a device that would always keep the knuckle open when the couplers are not mated would significantly increase the gathering range of present couplers. One device that keeps the knuckles open where couplers are not mated is a patented design known as the Compatimatic coupler. This device has been developed and tested by the National Casting Division of Midland Ross, but at this time it is not in production.

Guard Arm Extension - This would simply consist of extending the guard arm of E couplers to have the same geometry as the interlocking lug of an F coupler (see Figure 4).

With both the knuckle open device and the guard arm extension, all E and F couplers would have a total gathering rage of 8-3/4 inches for the parallel centerline situation. This is almost twice the gathering range of the present most common situation which, is two E couplers with one knuckle closed.

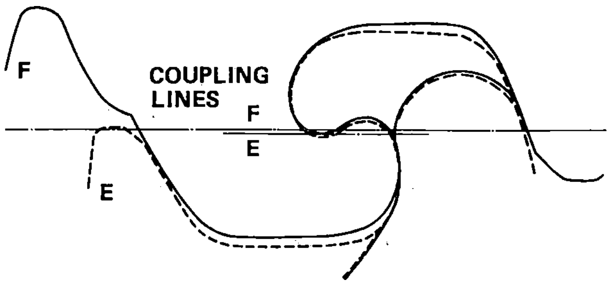


FIGURE 4. GUARD ARM EXTENSION OF F COUPLER RELATIVE TO E COUPLER [5]

Coupler Positioning Device - Increasing the gathering range, however, is not enough to ensure a very high probability of coupling for all cars. Approximately 4% of the present U.S. fleet of freight cars are equipped with couplers with 60 inch shanks. These cars are built with long overhangs; that is, the truck centers are located farther inward from the ends of the car than ordinary cars. Consequently, the coupler shanks must be made long enough that they will not cause a derailment on sharp curves when coupled to a car with a short shank coupler. Because these long shank couplers have such a large range of lateral freedom, they are also responsible for a high percentage of unsuccessful and bypassed couplings. The yardmasters at several of the yards visited as part of this project felt that these couplers were involved in 80 to 90 percent of their bypassed coupler problems.

If car coupling operations always took place on tangent track, this problem could be solved with coupler centering devices. However, many yards still have portions of curved track, and curved track is often encountered at local pick up and delivery sites. Consequently, coupler positioning devices are needed for these long shank couplers. Such a device steers the coupler based on the relative rotation between the truck and the car body. A picture of one type of coupler positioning device is presented in Figure 5 [6].

Implementation Scenario

We have assumed a 10 year period to implement the change over to knuckle open devices, guard arm extensions, and coupler positioning devices for long shank coupler cars. This is the consensus of the industry consultants advising this project. During this period new cars being introduced into service and cars being refitted with new couplers would be supplied with the above items, and cars that are expected to be

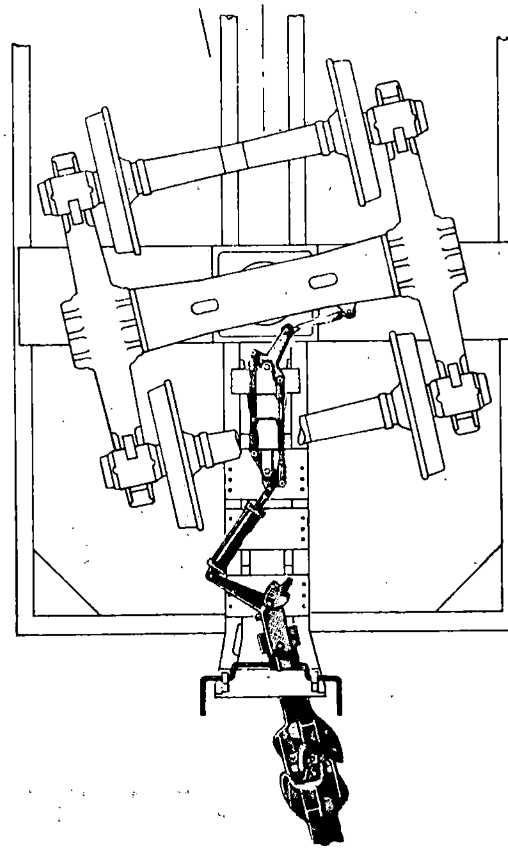


FIGURE 5. COUPLER POSITIONING DEVICE VIEWED FROM UNDERSIDE OF CAR [6]

operational after the conversion period would be retrofitted with them. Cars with less than 10 years of life remaining would not be retrofitted.

There would be some cost savings in normal coupler repairs (see Table 2) because there would be more newer couplers in the fleet and there would be a surplus of parts for the cars not being retrofitted. We have assumed that these savings of up to $53 million would be balanced by the additional labor expense for retrofitting cars with new couplers and positioning devices as necessary.

Note that the coupler change over would probably take less than a few man-hours per car, and the cars would always be compatible.

Unit costs

The average cost per car of knuckle open devices, guard arm extensions and coupler positioning devices (for the cars with long shank couplers) is made up of several components. The only presently available design of a knuckle open device requires a coupler body with more room inside to accommodate the extra parts. Therefore, the retrofit cost of a knuckle open device is both the incremental cost of the knuckle open hardware and the price of the coupler. The cost of phasing in knuckle open devices on new cars is just the incremental cost. The incremental cost is $125 per car in 1979 dollars [6], and the cost of a single new grade E steel coupler less the knuckle is $263.

The cost of a guard arm extension was estimated to be equal to the cost of a bottom shelf. This is because the amount of steel involved and the complexity are similar. This cost is $112.

The cost of a coupler positioning device is $2000 per car in 1975 dollars [6], which adjusts to $2780 in 1979. Since this item is for only 4 per cent of the fleet, the average cost per car is $111. The total average cost per car is $125 + 2($263) + $112 + $111 = $874. The new car cost is just $125 + $112 + $111 = $348, or 40 percent of the retrofit cost.

Gross benefits

All the devices presented here are designed to ensure a higher likelihood of coupling, and in combination it is expected that they would have a probability of coupling that is high enough to eliminate walking the classification tracks and supervising coupling cars during the pull down operation. Clearly these devices are for cars that are frequently coupled in normal classification yards; cars that are confined to unit trains would not have to be modified. The benefits of eliminating the supervision of coupling during the pull down operation were evaluated by using the yard simulation model described in the Methodology report [1]. These benefits are a savings of $167.2 million per year in labor, and $291.0 million per year in equipment utilization.The labor savings includes not only the man walking the classification tracks, but also the remaining crew in the switch engine. The equipment utilization savings includes the value of the freight car time as well as the value of the switch engine time.It is assumed that these benefits would not be realized until the entire fleet was changed over, that is, after 10 years.

Another benefit is elimination of the "cross over-open knuckle" task frequently performed in flat switching yards. This occurs when the pin puller cannot pull the pin of the car that remains with the locomotive, and because the trailing coupler of kicked cars may close on impact. Therefore the cut must sometimes be stopped to allow the pin puller to open the leading coupler on each car before it is kicked. The knuckle open device would eliminate the need to cross over and open the knuckle on these cars. The savings associated with the elimination of this task was determined by the yard simulation model to be $21.4 million for labor and $37.2 million for equipment utilization when fully implemented. In the interim time period, as the cars are being retrofitted, the benefits would be proportional to the number of knuckle open devices in service.

Additional Benefits There are two additional benefits which we have not included. These are (1) the reduction of equipment damage due to coupler bypass, and (2) increased safety due to the less frequent need to supervise the coupling of cars and the need to open a knuckle by hand. The magnitude of coupler bypass damage is difficult to estimate as it is not a separately identified item in the AAR car repair billing system. Interviews with yard supervisors conducted by A.T. Kearney suggest that the average cost to repair damage caused by coupler bypass was $800 in 1975 dollars, and that this occurred once every three years [6]. It is not clear whether this rate of damage was for all cars or just cars with long shank couplers.

Increased Maintenance Costs

The increased maintenance costs of these devices are taken as 4 percent of the incremental cost increase. That is, 0.04 x $236 = $9.44 per car. For a future fleet of 1,444,000 cars this amounts to $13.6 million per year.

Net Benefits

The net benefits are summarized in Figure 6.

The sloped early portion is the benefit for elimination of the crossover-open knuckle task, and the steady benefit after 10 years includes that benefit plus the reduced pull down time minus the increased maintenance.

Cost/Benefit analysis

The above benefit information was used in the financial model [1]. Cash flows were computed for 50 years, and it was assumed that the new car cost of the knuckle open device, guard arm extension, and coupler positioning device was 40 percent of the retrofit

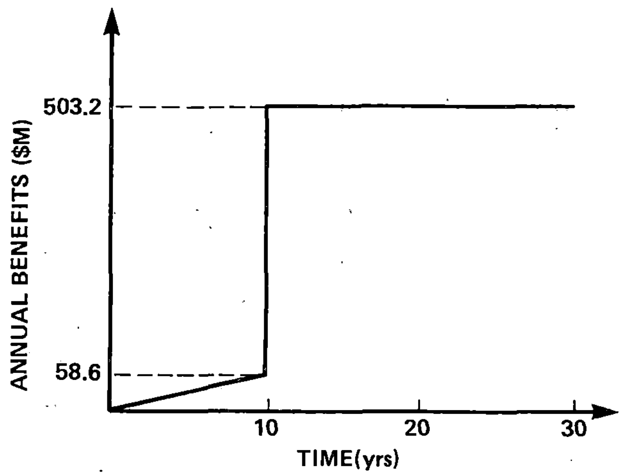


FIGURE 6. NET BENEFITS OF GREATER GATHERING RANGE

cost. It was also assumed that none of these benefits would be subject to labor union reimbursements.

For a 20 percent internal rate of return, the financial analysis estimated that $2157 would be an acceptable cost per car. The estimated cost per car is $874, so the ratio of the estimated cost to the allowable cost is

$$\frac{\text{Est. Cost}}{\text{Allowable Cost}} = \frac{\$874}{\$2157} = 0.41$$

Since this ratio is clearly less than unity, it indicates that these devices are a potentially favorable investment.

2.2.2 Automatic Air Line Connector

The moving knuckle type coupler presently in use in North America is sometimes referred to as an automatic coupler, but it is only automatic in the way that it makes the mechanical connection. The air line connection still must be made by hand, by a man stepping between the cars. Generally the air hoses for a train are coupled or laced after all the cars are assembled on the departure tracks. One or two men walk the train, stopping at the ends of each car to couple the hoses. After the hoses are coupled, the train is walked one or two more times - once to check for proper brake cylinder travel after brakes are applied, and possibly once again to see that the brakes subsequently release properly. Instead of taking the time for a man to walk the length of the train to inspect for brake release, many railroads will perform a roll-by inspection in which brakes are observed as the train departs.

An automatic air line connector would automatically connect the air line at the same time the cars are mechanically coupled. The use of this device would reduce the amount of labor needed to couple the air hoses, and save the amount of car time that it now takes to couple all the air hoses on a train.

Description and Implementation

One design of an automatic air line connector is shown in Figure 7. It uses wing type arms to guide the final alignment during coupling. The pressure seal is accomplished by simply butting the connectors together with stiff supporting springs that supply a compressive force greater than one kip.

In this analysis we have considered a minimum system that would consist of the air line connector but still maintain the air hose and glad hand connector so it would be compatible with present cars. This system would also include a new three-way angle cock that would allow the air line to be closed or routed through either the new automatic connector or the hose and glad hand. While this system was being phased in there would be no savings because someone would still have to walk the train to see that the new angle cocks were correctly set. After these units are fully phased in, it will only be necessary to check the angle cocks at the ends of the train. Again we have assumed a ten year implementation period.

At least one manufacturer has developed an automatic air line connector that also incorporates an automatic angle cock, but it is a more expensive system which is not required in order to achieve the benefits discussed here. The automatic angle cock would allow a car to be classified without having to be bled at one end of the yard and charged again at the other end. However, leakage would set brakes.

Unit costs

The estimated cost for the automatic air line connector comes from the earlier DOT coupler study [6]. In that study the reported cost was $300 per car in 1975 dollars. The estimated installation cost of this and other devices (such as the automatic angle cock) was one-third of the hardware cost for new cars and two-thirds of the hardware cost for retrofitted cars. To this we have added $70 per retrofitted car for the cost of two new angle cocks.

FIGURE 7. AUTOMATIC AIR LINE CONNECTOR [7]

This value comes from the AAR Manual of Interchange Rules for present angle cocks. Corrected to 1979 dollars this amounts to $556 for new cars and $765 for retrofitted cars.

Benefits

The benefit of this system is clearly to reduce or eliminate the man hours that are now spent coupling hoses, and perhaps more importantly to increase freight car utilization. The majority of the time that can be saved is in yard operations,but some time can also be saved in local pick-ups and deliveries. In the yard, the most time consuming portion of coupling air lines is not the actual coupling of hoses, but rather it is the time spent walking from car to car.

The benefits of automatic air line connectors in yard were found by using the yard simulation model [1]. These savings amount to $43.5 million per year for labor, and $75.6 million per year in equipment utilization time.

To estimate savings for local delivery and pick-up, we assumed that in most cases a man would be present at the cars that are being coupled, and hence the only time savings would be the time it takes to couple air lines. That is, for local trains we did not include any time for walking to cars that need their air lines coupled since the man would have to be there anyway to supervise the coupling or adjust the angle cocks. Based on a 26 day load cycle, an average local train consist of one locomotive and 20 cars, and the assumption that the average number of cars delivered per siding is two, the labor savings amount to $2.7 million per year and the equipment utilization savings amount to $1.9 million.

Maintenance costs for the automatic air line connector are estimated to be 4% of purchase price. Based on the new car price of $556 per car, and a future fleet of 1,444,000, this amounts to $32.1 million per year. One would also expect reduced maintenance costs for present air hoses which are often damaged during uncoupling. Presently, approximately 650,000 hoses are sold annually as replacement parts [8]. Of these, approximately 355,000 are used to replace 8 year old hoses, as required by the AAR interchange rules. We may assume that this routine change-over will continue because of normal hose aging, and also make the upper bound assumption that all remaining hose replacements could be avoided. From the Office Manual of the AAR Interchange Rules for 1978 and Table 1 of this report, the following savings could be achieved per hose:

Material	$15.25
Labor	4.57
Scrap Credit	(4.12)
	$15.70
Inflation Factor	x1.1
Total (1979 dollars)	$17.27

Applying this per-hose cost to the upper bound estimate of 295,000 hoses per year gives a potential $5.1 M of annual savings.

Accounting for all of the above costs, the total savings are therefore $100.7 million after the system is fully compatible,but zero until compatibility is reached.

Cost/Benefit analysis

The above benefit information was used in the financial model, and cash flows were computed for 50 years. It was assumed that none of these benefits would be subject to labor union reimbursements. This analysis predicted that the allowable expense per car would be $318 for an internal rate of return of 20 percent. The ratio of the estimated cost to the allowable cost for automatic air line connectors is

$$\frac{\text{Estimated Cost}}{\text{Allowable Cost}} \doteq \frac{\$765}{\$318} = 2.4$$

Since this ratio is greater than one, we conclude that this is not a good system for further consideration. Once the system was fully implemented the yearly benefits would exceed the costs by a factor of three, but the initial ten years of investment would not show a satisfactory rate of return.

2.2.3 Incompatible couplers

Many of the virtues of the basic knuckle coupler are also its limitations. Type E and F couplers in wide-spread domestic use are assembled from loosely fitting unmachined castings. The built-in clearances in all degrees of freedom between couplers allow for considerable misalignments between coupled cars, and for acceptable performance, even after components incur a considerable degree of wear. The result is that couplers are generally inexpensive and long lasting. However the inherent slack in couplers (and draft gear) also leads to high in-train dynamic loads which contribute to derailment tendencies and failure of draw gear components. Moreover, imprecise coupler alignment makes it difficult to integrate air or electrical connectors which require precision alignment. When the knuckle concept is abandoned, couplers can be developed which overcome these functional limitations, but with the penality of higher cost and mechanical incompatibility.

Description of system

Of the numerous incompatible couplers that have been designed or developed, the UIC (spread claw) type stands out for its attractiveness for potential use in freight systems, and will be described and evaluated here.

In 1956, the International Union of Railways (UIC)* began to study the feasibility of replacing the European hook and screw coupler with an automatic coupler. Their primary reasons have been to

- o Make operations increasingly automated and more effective
- o Reduce personal accident rates
- o Become less dependent on a limited labor supply.

One of the major constraints faced by the UIC was the need for a coupler that would be compatible with future Eastern European couplers, which, in turn, had to be compatible with the Willison type SA-3 coupler used in the USSR.

The coupler developed by the ORE (the UIC's Office of Research and Tests) and a consortium of coupler manufacturers** is shown in Fig. 8. The coupler is comprised of skewed mechanical alignment claws or arms, two air line connectors and a 10-pin electrical connector. The two air lines are included for possible eventual use of electropneumatic brakes. Four of the 10 electrical contacts are for air brake operation, two for communication, and four for the supply of low voltage electric power. To protect the system from snow, rain, dirt, and other contaminants, both air

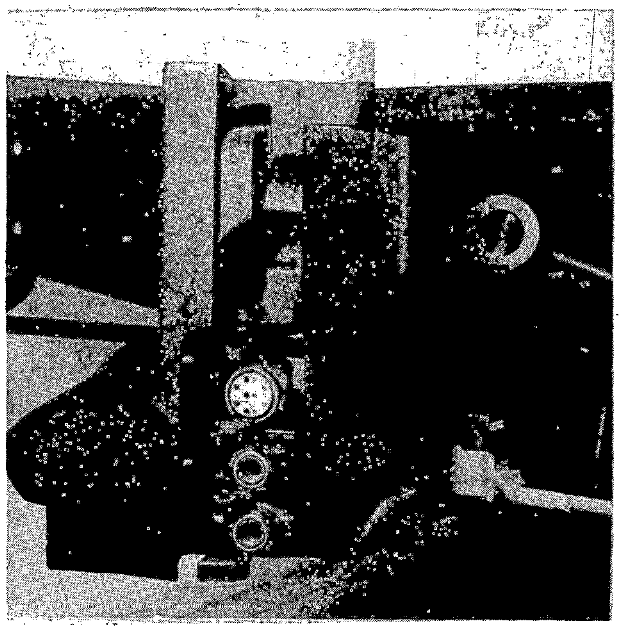


FIGURE 8. UIC AUTOMATIC COUPLER [PHOTO COURTESY OF THE UIC]

* Union Internationale des Chemins de Fer.

** Unicoupler/Boirault - Sambre and Meuse/National.

lines and the electrical connector may be fitted with caps that move away upon coupling.

The operation of the UIC coupler, shown schematically in Fig. 9, consists of three stages: engagement, penetration, and coupling. During engagement, two opposing couplers contact and align themselves as the cars to which they are attached close on one another. As shown in Fig. 9a, if the couplers are initially offset to the left, the oblique faces of the bottom arms engage and slide along each other. If the couplers are offset to the right, the top arms and lugs engage and slide the couplers into alignment. As the couplers swing into alignment, the lugs penetrate the opposing pockets and depress the check locks, pushing the locks partially back within the coupler heads (Fig. 9b). When the motion is complete (Fig. 9c), the locks fall back into position under their own weight and prevent separation of the coupling heads. During this action, the check locks activate a securing catch which prevents the lock from recoiling and inadvertently uncoupling the couplers.

Uncoupling is performed by retracting the lock of one coupler by means of a control lever. Unlike the levers used on U.S. knuckle couplers, the control lever extends to <u>both</u> sides of the car. Once the coupler heads separate, the check locks fall back into position and the couplers are again ready to couple.

The UIC coupler is designed to meet the following specifications [9]:

- Yield strength (at 0.05%) in draft: 1000kN (225 kips)
- Yield strength (at 0.05%) in buff: 2000kN (450 kips)
- Ultimate strength in draft: 1500kN (337 kips)
- Weight (coupler body, locking system, and pipe coupler): 250kg (551 lb)
- Coupling speed range: 1.5 - 15 km/hr (0.93-9.3 mph)
- Vertical gathering range: 140 mm (5.5 in.)
- Horizontal gathering range: ±220mm (±8.7 in)

<u>Implementation scenario</u>

Converting the fleet from knuckle to incompatible couplers such as the UIC coupler requires years of planning and preparation, followed, in all likelihood, by a very rapid change-over. Throughout the preparation period, annual costs associated with manufacture of new components and modifications to existing cars would probably be fairly uniform. Costs would undoubtedly become quite high during the actual conversion. As a reasonable approximation to this scenario, we assume retrofit costs are uniformly distributed over a 10 year period. Beyond, only incremental costs for new components and their maintenance cost are incurred.

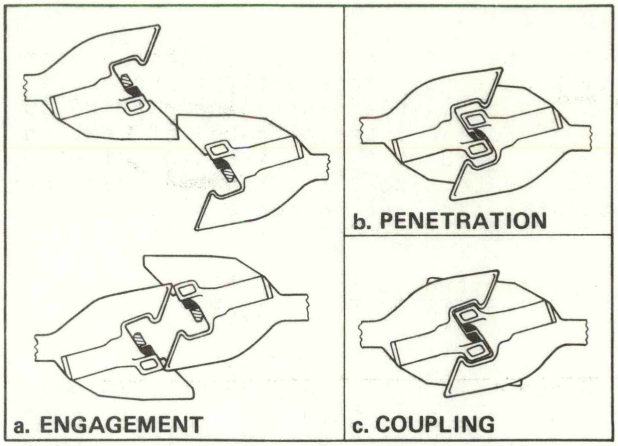


FIGURE 9. OPERATION OF UIC AUTOMATIC COUPLER

<u>Unit costs</u>

In evaluating the costs of converting the Western European railroad fleet of 834,000 cars from the hook and screw to a spread claw type automatic coupler, the UIC estimated the cost of coupler and draft gear components at 4.6 billion German Marks (DM) in 1972. In addition, 2.9 billion DM would be needed for the structural modifications, 3.7 billion DM for early car retirement and 1.3 billion DM of additional costs associated with car preparation and conversion, bringing the total to 12.5 billion DM. To apply the results of the UIC study to American railroads, we perform new-car and retrofit estimates. The new-car estimate simply extrapolates the UIC coupler price to 1979 dollars and assumes there is no incremental labor or hardware needed for installation. The estimate for retrofit extrapolates the total UIC figure to the U.S. railroads.

<u>New Car Costs</u>

The 4.6 billion DM for hardware applied to 834,000 cars results in a per car cost of 5516 DM. In 1972 1DM =

$0.31364 U.S. Using this and the 2.18 inflation factor from Table 1 results in an estimated cost of $3771 per car set in 1979. Incremental costs are estimated by subtracting costs of conventional components:

	UIC coupler pair	$3771
Less:		
	two E60CHT couplers at $320	<640>
	two Y40 AHT yokes at $150	<300>
	two M901 draft gear at $400	<800>
	Net	$2031/car

Retrofit Costs

Allocating the total 12.5 billion DM among the 834,000 cars requiring retrofit results in a cost of 14,988 DM per car. Converting to U.S. dollars, as above, and escalating the results from 1972 to 1979 gives

Retrofit cost: $10,248/car

It should be recognized that this is an upper bound estimate. Conversion of U.S. rail cars to the UIC type coupler would probably be easier because they are already configured to carry all of the longitudinal loads through center sill couplers. European cars, in contrast, have side buffers which would have to be removed.

Maintenance Costs

Using the 4% of purchase price to estimate maintenance expenditures gives an incremental annual maintenance cost of $81.24/car. For the 1,444,000 car population projected to the year 2000, the total incremental maintenance cost would be $117.3 million.

Gross benefits

The UIC type coupler has the potential for significantly improving yard and road operations and reducing equipment damage. The following is an estimate of the potential annual benefits.

Yard Operations

The yard operations model is used to estimate the labor and equipment utilization savings that can accrue from the use of a UIC type coupler. Benefits are assumed to accrue from the elimination of

- Manual air hose coupling
- The need to cross in front of a car and open a knuckle
- Trimming of miscoupled cars on a classification track prior to pull down

The resulting savings are shown in Table 3.

TABLE 3. ANNUAL BENEFITS FROM UIC TYPE COUPLER

	Time	Cost (mil)
Road Crew	227 man-yr	$ 6.67
Yard Crew	5,981 man-yr	173.80
Yard Inspectors	1,025 man-yr	30.20
Total Labor	7,233 man-yr	$210.67
Road Locomotives	0.34 mil Loc hr	$ 8.16
Switch Engines	5.90 mil Eng hr	124.03
Freight Cars	260.42 mil car hr	234.38
Total Equipment	–	$366.57
Grand Total		$577.24

Road Operations

The non-compatible coupler simplifies road operations by eliminating the need (and associated times) to couple hoses and open and close angle cocks. Based on the model described in Ref. 1, the time savings shown below in Table 4 occur for each pick-up and delivery.

TABLE 4. TIME SAVINGS FOR ROAD PICK UP AND DELIVERY WITH UIC TYPE COUPLER

Operation	Time Per Occurrence (Min)	No. of Occurrences	Total Time (Min)
Couple Hose	0.125	2	0.25
Open Angle Cock	0.357	2	0.714
Close Angle Cock	0.200	2	0.40
			1.364

The average load-to-load cycle for a freight car in the U.S. is approximately 26 days. Assuming that each pickup and each delivery involves two cars, the projected number of pick-ups and deliveries is $(1.444 \times 10^6$ cars) $(365/26)(0.5) = 10.1$ million. Thus, the labor savings potential for a four-man crew is $4 \times 10.1 \times 10^6 \times 1.364/60 = 918000$ man hrs. At an average rate of $14.16/hr. the potential labor savings is $13.00 million/year. Equipment utilization time is reduced by $10.1 \times 10^6 \times 1.364/60 = 230000$ hrs. Assuming further, that a local delivery train is comprised of one locomotive and 20 cars, the equipment utilization savings would be (230000 hrs)($15.89/locomotive-hr + 20x$0.78/car hr) = $7.24 million.

Other Benefits

Three other benefits that could be realized, but are very difficult to quantify, are (1) a reduction in derailments and broken train collisions caused by coupler failures, (2) a reduction in equipment damage due to coupler bypasses, and (3) a reduction in the incidence of undesired angle cock closing through tampering or human error. For the first, almost all of the $12 million annual costs are attributable to derailments. It is difficult to project how much of this would be saved by a UIC type coupler. On the one hand, the interlocking feature should keep couplers from falling to the track (and subsequently causing a derailment) in the event of a shank failure. On the other hand, the couplers are larger and heavier than present E type knuckle couplers and would presumably increase the likelihood of a derailment if they did in fact fall. Because of this uncertainty and because $12 million is small compared to the potential of approximately $600 mil of savings in yard and road labor and equipment utilization, we neglect accident cost reductions.

The magnitude of coupler bypass damage is unknown and extremely difficult to estimate. Interviews with yard supervisors have suggested that, while it occurs, it frequently is sufficiently low to make data gathering at a variety of yards an inordinately time consuming task. Moreover, current records (e.g. the AAR car repair billing system) would not be sufficiently detailed to enable one to discriminate repair costs associated exclusively with coupler bypasses from costs due to other causes. We judge that these potential savings are sufficiently low and difficult to determine to be neglected in comparison with the potential yard and road operational savings.

Data do not appear to be available on accidents associated with undesired angle cock closing.

Summary

The following is a summary of the estimated benefits that maybe realized from the UIC type coupler:

Yard Labor	$210.67 M
Road Labor	13.00
	223.67
Yard Equipment	366.57 M
Road Equipment	7.24
Total Equipment	$373.81 M
Grand Total	$597.48 M

Clearly, the major savings would occur in yard operations.

Cost/Benefit analysis and recommendations

When the above parameters are used in a financial analysis, it is estimated that $1,717 are available to retrofit each car and $340 are available for new cars. These figures compare very unfavorably with the estimated retrofit cost of $10,248 per car and the incremental cost of $2031 needed just for the coupler components.

2.3 Improved Dynamics: Alternate Mechanical Components

The mechanical systems evaluated in this section have the potential for enhancing train dynamics and/or reducing component wear and failure.

2.3.1 Truck mounted brakes

Truck mounted brakes have been in service for several decades and provide an alternative to conventional brake rigging. Their primary feature is the development of a symmetrical brake force application on trucks and wheels, resulting in lower wheel wear than for conventional systems.

Description of System

Truck mounted brake systems use the same brake valve, reservoir, and associated piping as conventional systems. However, instead of supplying air to a single body-mounted cylinder, air is supplied to four truck mounted cylinders. Fig. 10 illustrates a truck mounted brake assembly in relation to a standard three-piece truck. Opposing cylinders, supplied with air through flexible hoses leading to the car body, generate forces directly on brake beams. These beams force shoes against wheel treads.

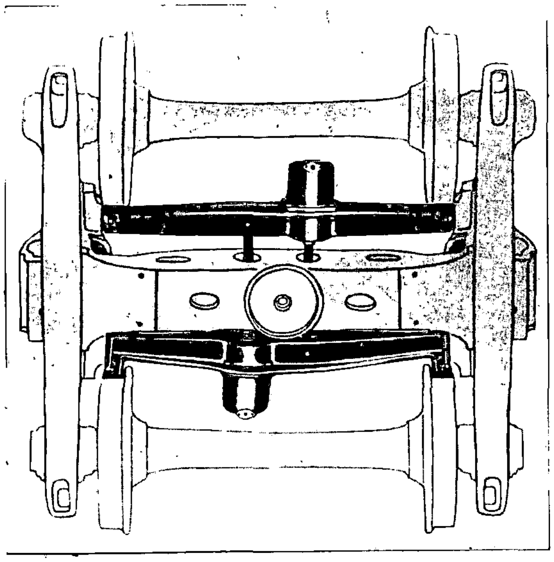


FIGURE 10. TRUCK MOUNTED BRAKE ASSEMBLIES [3]

Application of truck mounted brakes would only be considered for new or rebuilt cars. Cars in fleet service would almost surely never be retrofitted with truck mounted brakes because of the high cost of scraping the present conventional brake hardware. Moreover, since there are no car-to-car compatibility issues, benefits of using truck mounted brakes begin to accrue with each car that is equipped.

Unit Cost

The cost of either a truck mounted or conventional brake system depends on the specific configuration and the practices of the car manufacturer installing the system. WABCO has estimated the costs for three truck mounted systems, with several hand brake arrangements. The results are shown in Table 5 and escalated (by means of Table 1) to 1979 estimated prices.

Implementation Scenario

Estimating the cost of conventional brake systems is not a straightforward matter. Car builders purchase components from a number of different suppliers and install them on cars without necessarily keeping careful track of the labor required for this installation. The costs of conventional rigging are reputed to be approximately the same as for truck mounted brakes but will vary among manufacturers. Accordingly, we shall leave the price differential (between conventional and truck mounted brakes) as a variable in the cost/benefit evaluation.

Maintenance cost

The following chart summarizes the comparison of scheduled maintenance costs for truck mounted brake and conventional rigging.

	Truck Mounted Brake Assembly		Conventional Rigging	
	Total	Per Year Based on 14 Year Cleaning	Total	Per Year Based on 14 Year Cleaning
COTS[1]	$481.03	$34.36	$315.06	$22.50
Slack Adjuster Maintenance[2]	*	*	80.54	5.75
Total	$481.03	$34.36	$395.60	$28.25

*Slack Adjuster Not Needed.

[1]Based on AAR Job Codes 1050 and 1061 for 1978, escalated to 1979.

[2]Based on AAR Job Code 1556 for 1978, escalated to 1979.

TABLE 5. PRICE ESTIMATES FOR TRUCK MOUNTED BRAKES.

Handbrake Arrangement	Two 8½ in. Truck Brake Assemblies	2nd Set* Levers "A" End	Handbrake (Approximate Price Each)	Components & Labor Connecting A & B End (Estimated)	Total	Estimated Total Price 1979
1	$940	$123.50	S63-2 Mechanism & AAR 66 Bell Crank $240.00	$ 90	$1393.50	$1547
2	$940	$123.50	S63-2 Mechanism & AAR 66 Bell Crank $240.00	$190	$1493.50	$1658
3	$940	$123.50	H.P. Mechanism & Sheave Wheel $300.00	$300	$1663.50	$1846

*Source: WABCOPAC Brake Assembly, WABCO, January 1978.

While truck mounted brakes eliminate the slack adjuster maintenance cost, there are four brake cylinders to service instead of only one resulting in a higher yearly maintenance cost of $6.11.

Use of truck mounted brakes, especially on low mileage cars, is very sensitive to maintenance costs. Careful consideration should be given to the possibility of unforeseen nonscheduled maintenance costs. Points of concern are:

1. Four brake cylinders to fail instead of one. (cylinder reliable.)(# of cylinders) = System Reliability. Thus, to achieve the same system reliability, the truck mounted brake cylinder reliability would need to be higher than for a conventional system.

2. The harsher environmental conditions of the lower truck mounted brake cylinder mounting position in terms of vibration, dirt and water, temperature extremes.

3. Relative ease of maintenance when a problem does occur.

Gross Benefits

The primary benefit of truck mounted brakes is the reduction of wheel wear due to a symmetrical application of brake shoe forces. Fig. 11 shows how a turning moment is generated on a truck equipped with conventional rigging with the top rod over the bolster and dead lever anchored to the truck bolster. The moment is resisted partly by center plate friction and partly by lateral forces on diagonally opposite wheel flanges. These flanges will tend to wear faster than the companion wheels on the same axles, and require changing sooner than if this effect did not occur. Turning moments are virtually absent on trucks equipped with conventional brakes with a bottom rod under the bolster and the dead lever attached to the car body. However, empirical data suggest that wear occurs primarily on the two wheels located on the same side of the car as the top rod and anchor rod. The reason for this is not clear.

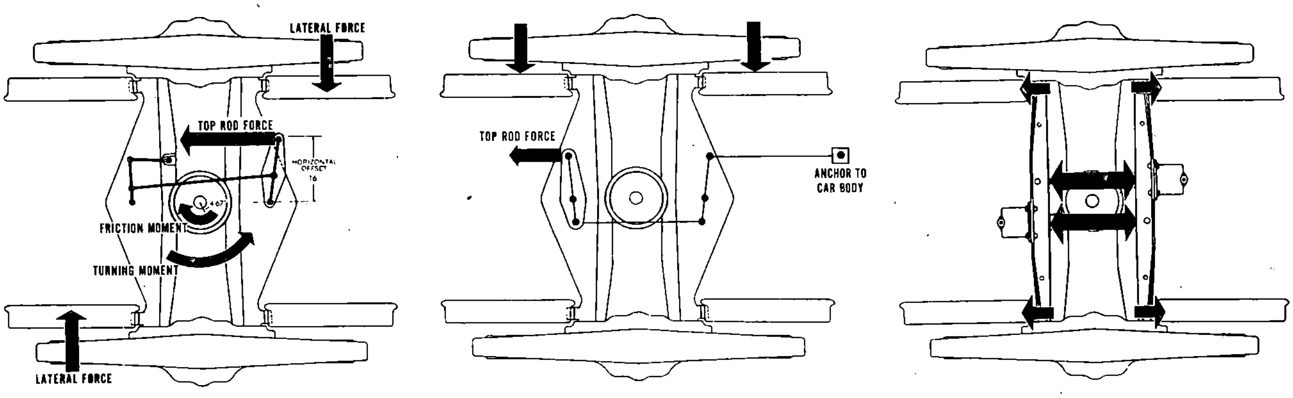
The Westinghouse Air Brake Co. monitored the wear on the wheels of 450 new 100 ton cars used in unit coal train service for a four year period [10]. A total of 215 cars were equipped with truck mounted brakes and 235 cars with conventional rigging with the bottom rod under the bolster and the dead lever attached to the car body. All cars were equipped with 36 in 1W wheels and travelled about 100,000 miles per year. The average wheel life was 325,000 miles for cars equipped with truck mounted brakes and 245,000 miles for cars equipped with conventional rigging.

The cost of changing a pair of wheels is given in the following table:

2 new 36" wheels (AAR Job Code 3075)	522.54
Labor (AAR Job Code 3160)	212.04
Scrap Value of two worn wheels (AAR Job Code 3075)	(52.61)
Total	681.97

Source: Office Manual of the AAR Interchange Rules for 1978, escalated to 1979.

The total cost for a car is $2727.88. This reduces to 0.839 cents/mi for cars with truck mounted brakes and 1.113 cents/mi for cars with conventional



Top Rod Over Bolster   Bottom Rod Under Bolster   Truck Mounted Brakes

FIGURE 11. ILLUSTRATION OF FORCES GENERATED BY THREE BRAKING SYSTEMS [10]

bottom rod under bolster brakes, or a 0.274 cent/car/mi difference.

Cost/Benefit

Fig. 12 shows the cost/benefit relation as a function of annual mileage based on the values presented above. For this unique situation, the maximum allowable incremental investment is estimated as the per car benefits divided by the desired 20% return. If truck mounted and conventional rigging cost the same, the break-even point would occur at a car utilization of only a few thousand mi/year.

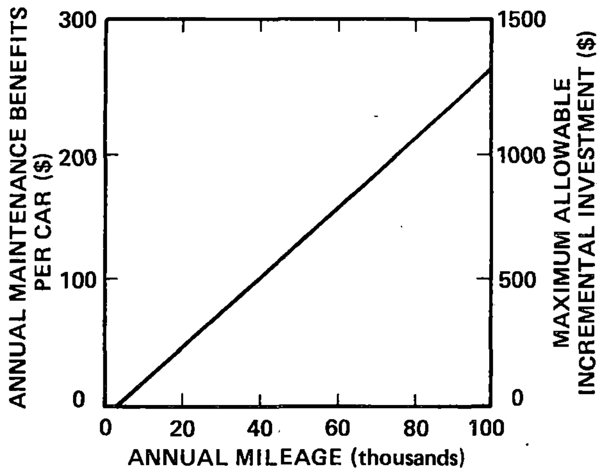


FIGURE 12. ANNUAL MAINTENANCE BENEFITS PER CAR($)

We would caution the reader that these results do not include unscheduled maintenance. They are also based on empirically determined differences in wheel wear rates, the cause of which is not completely understood.

2.3.2 Disk Brakes

Freight cars operating in North America and much of the world rely primarily on tread brakes to develop retardation forces.* These forces are generated by the tangential friction forces occurring when a composition or cast iron brake shoe is applied to the tread (i.e. running surface) of each wheel. Several effects inherent in the interaction between brake shoes and wheels are the wear, temperature rise, and possible failure of each component.

* Braking forces are also developed by locomotives which use traction motors to generate electrical currents (dynamic braking) or use tread brakes.

In fact, wear rates of brake shoe and pad materials increase exponentially as temperatures rise. As wheel surfaces are heated, internal wheel stresses are generated that can ultimately contribute to failure. If surfaces become very hot and are quickly cooled, localized metallurgical changes occur which also contribute to failure.

In contrast to these deleterious properties, tread brakes have certain beneficial effects. Brake shoes clean contaminants from the tread, thereby enhancing the wheel/rail adhesion. Moreover, brake shoes will tend to restore the profile of a hollow-worn wheel to one more nearly resembling that of a new wheel.

A heavy demand on brake systems occurs when trains are stopped from high speeds. As shown in Figure 13 [11], tread temperatures can reach 500-600 °F when 100 ton cars are stopped from 80 mph. These temperatures are generally not regarded as damaging. Unless deceleration rates become significantly higher, it appears that tread brakes are adequate for stopping.

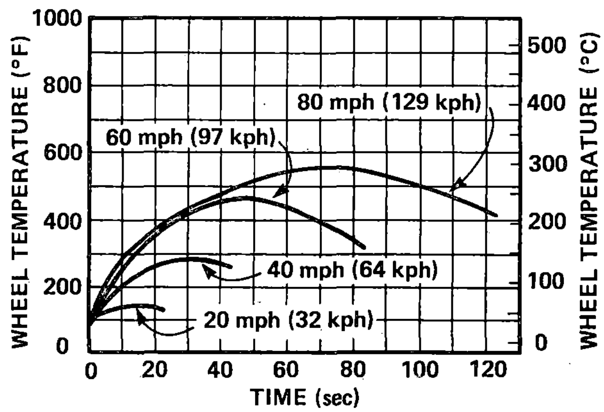


FIGURE 13. WHEEL TEMPERATURES DEVELOPED DURING STOP TESTS [11]

Probably the most severe demand for braking occurs when brakes are applied for extended periods of time to maintain a controlled speed on a long down grade. Disk and tread brakes together could absorb about twice the energy of tread brakes alone. This would enable trains to proceed faster down those grades for which brake heating is a limitation. However, heavily graded track is often accompanied by numerous curves which limit speeds. Therefore, it is not clear how much additional speed could be achieved in practice.

Description of system

Figure 14 shows a representative disk brake assembly. The disk is mounted to an axle and consists of two plates separated by a set of radial vanes. As the disk rotates the vanes pump air outward through the intervening spaces, providing some cooling for the plates. The plates are heated during braking by a pair of brake pads pressed against each side of the disk to generate tangential forces. Brake pad forces are developed by a pneumatic cylinder acting through a pair of tongs pivoted at the bridge.

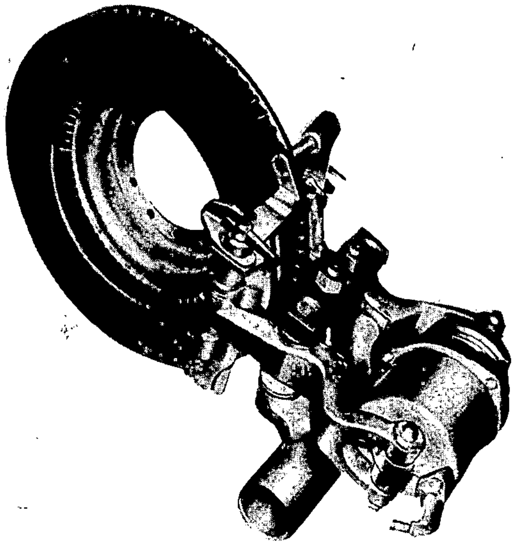


FIGURE 14. WABCO DISC BRAKE [3]

Disk brakes are able to absorb about the same power as a pair of wheels [11]. Therefore, little would be gained by merely replacing tread brakes by disk brakes. To achieve greater braking capacity, it would be necessary to supplement tread brakes with disk brakes.For cars equipped with conventional brake rigging, this implies the addition of flexible air lines from the brake valve to the disk brake cylinders which would have to be mounted on the trucks. Because of the volume of air required by the disk brakes, beyond that needed for the tread brake cylinder, an additional or enlarged reservoir would also need to be added to the car.

Implementation scenario

It is unlikely that disk brakes would be used in general service. They could conceivably be used on unit trains in mountainous regions or on future high speed trains with rapid stopping requirements. Therefore, in contrast to comparable sections of this report, we will consider the installation of disk brakes on a specific train, but will not attempt to evaluate a fleet-wide conversion to disk brakes.

Unit costs

The following are the costs of a single brake system used for passenger cars:

Disk and Hub	\$ 725
Brake Assembly	\$2125
Total	\$2850

In addition, a reservoir and piping are estimated as

Reservoir	195	(1978 price of AB reservoir escalated to 1979 from Table 1)
Piping	\$100	(est)
	\$295	

Therefore, the total costs for a four axle installation is approximately \$11,700.

Gross benefits

Three potential benefits are 1) less wheel damage due to thermal cracks, 2) faster speeds down grades, and 3) lower wear rates of composition shoe or pad materials accompanying lower temperatures. Here we will consider each factor at the order-of-magnitude level of analysis.

Wheel damage due to thermal cracks is virtually negligible. The cost of removing wheels because of thermal cracks was 0.2% of all wheel repair costs accounted for in the AAR car repair billing system [12]. The cost of accidents associated with broken wheels (FRA Cause Codes 460, 461, 462, 463) and damage from thermal/flat causes (Cause Code 466) is also negligible. In 1977 there were 153 such accidents [13]. Using an average cost of \$42,912 per wheel-related accident [13], escalating the result by a factor of 2 to account for unreported costs and by 1.2 to account for inflation, results in an upper bound estimate of $15.8 mil/yr. Using the 1977 fleet size of 1,667,000 cars, this amounts to $9.45 per car or less than 0.1% of the cost of disk brakes. Therefore, savings associated with both maintenance and accidents may be neglected.

The degree to which a dual tread/disk system allows higher speeds may be illustrated by means of an example. We will follow the example presented in Sec. II of Ref. 14* of a 100 car train descending a 1.5 percent grade with dynamic brakes applied. Assuming that 20 hp per wheel is selected as an upper limit, the train is able to descend at 30 mph. We will further assume the grade is 30 miles long, for an altitude change of 2375 ft.

From Figure 15 it may be seen that the brake shoe temperature will reach approximately 500°F (260°C). Figure 16 indicates that the wear rate for composition materials at this temperature is 0.204 cc/MJ. Since the total friction braking energy consumed per wheel is 20 HP-hr or 53.69 MJ, the material lost per brake shoe is about 11.0 cc. By splitting the braking energy between disk and tread brakes, wheels would be required to absorb only 10 HP. Figure 15 indicates that shoe temperatures would reach about 340°F (171°C) and Figure 16 indicates a wear rate of 0.048 cc/MJ at this temperature, or 1.29 cc/wheel plus 2.58 cc/disk. The difference between the amount of brake shoe material lost at the higher temperature and amount of shoe and pad material lost at the lower temperature is 67 cc per car.

A 2 in. brake shoe contains about 1000 cc of material that may be worn before the shoe must be replaced. The cost of replacing a shoe is $7.19 for material and $5.64 for labor (AAR Job Code 1840), or a total of $12.83 for the 1000 cc of material. Therefore, the extra 67 cc per car identified above represents 86 cents in brake shoe wear.

Instead of conserving friction brake material, the train illustrated in the above example equipped with tread and disk brakes could descend the grade at

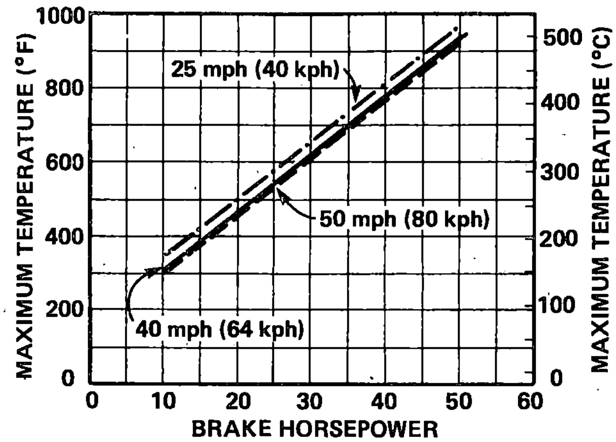


FIGURE 15. EFFECT OF BRAKE HORSEPOWER AND SPEED ON WHEEL TEMPERATURES [11]

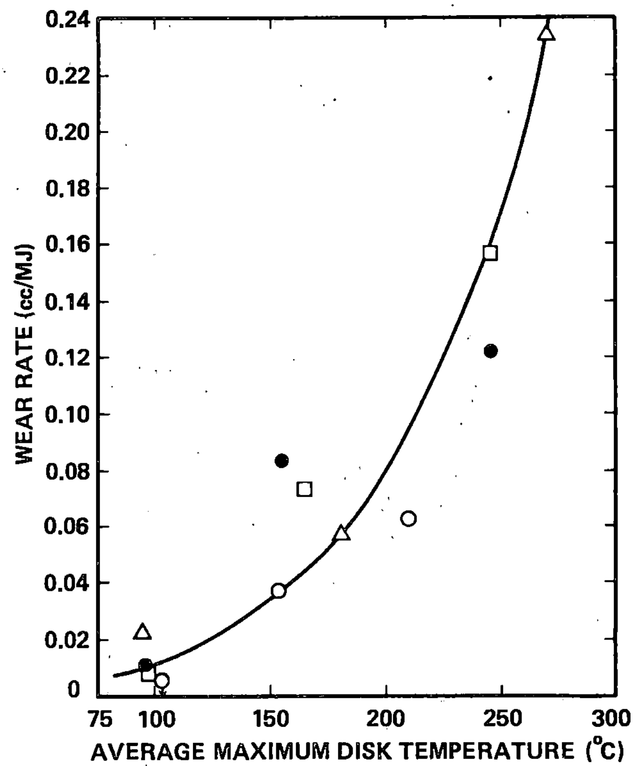


FIGURE 16. VARIATION OF WEAR WITH TEMPERATURE FOR FOUR EXPERIMENTAL COMPOSITION MATERIALS [15]

up to 55 mph, and still limit brake power to 20 hp per wheel. At this speed the train would save about 27.3 minutes. This amount of time for an "average" 68 car train discussed in Ref. 1 would be worth $84.46 or $1.24 per car. This is a somewhat more cost-effective strategy than operating slowly to conserve brake shoe material.

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* The train is assumed to consist of 100 cars weighing 125 tons each, four six axle locomotives at 195 tons each, and one 20-ton caboose. The total train weight is 13,300 tons

Cost/Benefits

Whether disk brakes would be used to save material or time, it is unlikely that their initial expense could be justified for freight application. The initial $11,700 per car investment would have to save several thousand dollars per year. This requires several thousand brake applications per year (or about 10 per day) of the type discussed in the above example. This is extremely unlikely with present freight trains or for those anticipated in the foreseeable future.

2.3.3 E couplers with bottom or top shelf

A coupler with a shelf provides a vertical interlock which prevents broken couplers from falling on the tracks and causing derailments. A top shelf coupler keeps itself from falling if it breaks, but does not help if the coupler to which it is mated fails. Alternatively, a bottom shelf coupler will hold a broken mated coupler, but will not catch itself it if fails. Figure 17 shows a standard E coupler, and an E coupler with a bottom shelf.

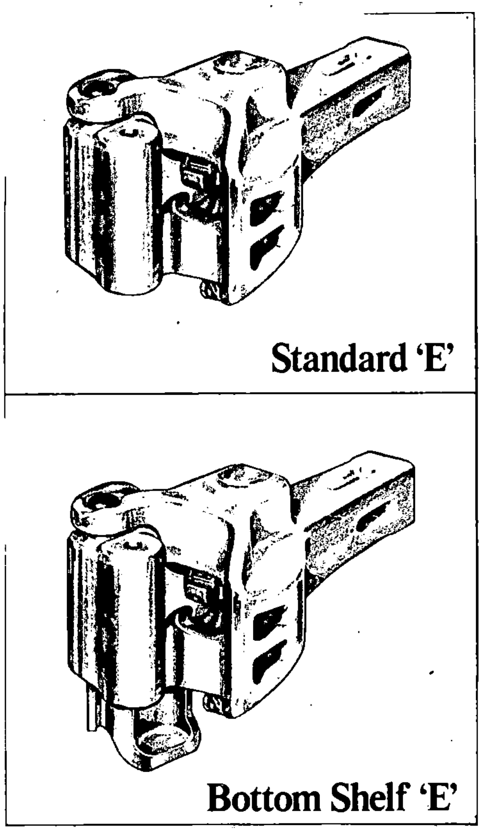

FIGURE 17. STANDARD E COUPLER, AND E COUPLER WITH BOTTOM SHELF [16]

If all couplers had bottom shelves (or if all had top shelves), then no broken couplers would be able to fall on the tracks. An alternative design would be to use interlocking side lugs and pockets similar to those on F couplers; this would be the preferred alternative if the guard arm extension discussed in Sec. 2.2.1 were adopted because it would leave room under the coupler for the addition of an air line coupler or an electrical coupler.

Implementation Scenario

Three implementation scenarios were considered. The first scenario is simply to phase in shelves on new cars, and cars that need replacement couplers. Based on present values for coupler life this would take approximately 25 years.

The second scenario would be to accomplish the change-over in ten years in conjunction with coupler change-over suggested by Section 2.2.1. The third scenario would be accomplished by independently welding shelves onto existing couplers as was done recently as a safety precaution for all tank cars that carry dangerous liquids or gases.

Unit cost

The 1979 incremental value for couplers furnished with bottom shelves is $112 per car. It is expected the cost for either a top shelf alone or a side pocket (if the guard extension design of Sec. 2.2.1 were adopted) would be similar. If the shelves had to be welded to existing cars, the price would probably increase by approximately two man-hours, or roughly $28 per car. The increased maintenance costs are expected to be negligible.

Benefits

The benefits for couplers with shelves are discussed in Section 4.6.3 of the Methodology report [1]. If all couplers had shelves, the savings would be as much as $12 million per year. If shelves are phased in over a period of time, then the savings would be proportional to the number of couplers with shelves in service, with 100% implementation corresponding to $12 million.

Cost/benefit analysis

The three scenarios described above were used with the financial model to determine the allowable cost for each scenario. The results are summarized in the following table.

In all cases the yearly expense of supplying new cars with shelf couplers was equal to approximately one-half the yearly savings, but only after almost all couplers were equipped. The early expenses, incurred before the system is fully equipped with shelf couplers, are not justified by the long term discounted returns on the investment.

Scenario	Ratio of Estimated Cost to Allowable Cost
1. Phase in on new cars for a period of 25 years.	$\frac{112}{58} = 1.9$
2. Phase in on retrofitted couplers over a period of 10 years.	$\frac{112}{52} = 2.2$
3. Weld shelves to existing couplers and phase in on new couplers such that system is compatible in 10 years.	$\frac{112+28}{56} = 2.5$

2.3.4 High strength draw gear

When draw gear components (knuckles, couplers, yokes) break or wear out they generate maintenance expenses and can cause train delays or accidents. Over recent years there has been an improvement in the quality of steel used to manufacture these components and an increase in their useful lives. The once-common Grade B steel is no longer used, with most components presently made of Grade C steel and some of Grade E steel.

Implementation Scenario

The efficiency of an implementation strategy depends on the expected life of each component. The life of existing components is estimated from failure rate data as shown in the following table.

TABLE 6. FAILURE RATES AND LIFE EXPECTANCIES FOR MAJOR DRAW GEAR COMPONENTS

Component	Breakage Failures Per Year [1] (Thousands)	Failure Rate*	Mean Life (Yr)
Knuckles	373.5	0.109	9.19
Couplers	136.2	0.0397	25.2
Yokes	69.6	0.020	49.3

*Failure per year divided by 1,716,937 cars x 2 couplers/car. The data apply to 1972.

From Table 6, and a recognition that the life of a freight car is about 30 years, one may conclude that knuckles will have to be replaced several times during the life of a car, most couplers will be changed once, and most yokes will never be replaced. Accordingly, it may be most appropriate to replace broken knuckles with longer lived (and more expensive) components made of Grade E steel but to replace broken couplers and yokes with conventional Grade C components. In summary the strategy that we will evaluate is as follows:

- Replace broken knuckles with Grade E knuckles.
- Replace broken couplers and yokes with Grade C components.
- Equip all new cars with Grade E draw gear components.

Unit Costs

The incremental unit costs for using Grade E rather than Grade C steel in major draw gear components are

Knuckles	$ 8.90
Coupler Body	15.25
Yoke	5.75

Gross Benefits

As illustrated in Fig. 18 of Ref. 1, Grade E steel has better fatigue properties than Grade C steel. It also has better wear properties, but these are difficult to quantify due to lack of available data. To estimate the increased life of components cast with E Grade, rather than C Grade steel, consider the load histogram and fatigue curves shown in Fig. 18. The figure shows that the component experiences n_i cycles/year at load F_i. The cumulative damage is $n_i/N_{i,c}$ for C Grade material and $n_i/N_{i,E}$ for E grade material. The total damage D in a year accruing from the complete spectrum is

$$D = \sum_i n_i/N_{i,c}$$

The load/cycle relationship for each material is given by

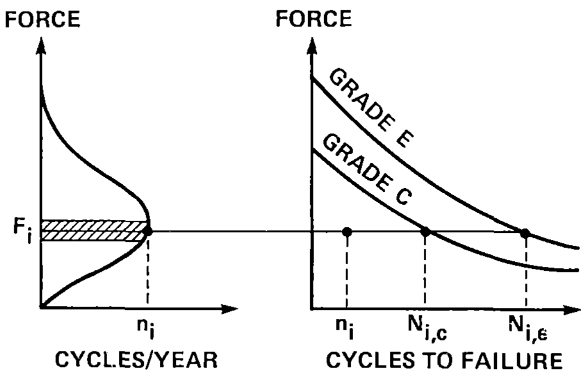

FIGURE 18. LOAD HISTOGRAM AND FATIGUE CURVES FOR TWO GRADES OF STEEL (NOT TO SCALE)

$$F^{\alpha_C} N = B_C$$

$$F^{\alpha_E} N = B_E$$

where the α's and B's are empirically determined constants. The life L of a material is simply D^{-1}. Therefore, the ratio of fatigue lives for E and C grade materials is

$$\frac{L_E}{L_C} = \frac{B_E \Sigma n_i F_i^{\alpha_C}}{B_C \Sigma n_i F_i^{\alpha_E}}$$

As shown in Ref. 1, $\alpha_C \simeq \alpha_E$. Also, $B_E \simeq 2.5 B_C$. Therefore, $L_E \simeq 2.5\ L_C$.

That is, the fatigue life of E grade steel is about 2.5 times that of C grade steel. Based on this information and the data in Table 6 we assume that grade E components will last the life of the car.

Benefits accruing from improved draw gear components are (1) reduced maintenance expense, (2) reduced train delay, (3) fewer derailments. Benefits (million dollars per year) are shown in Table 7:

TABLE 7. BENEFITS OF USING HIGH STRENGTH DRAW GEAR COMPONENTS

	Maintenance[1]	Train Delay[2]	Derailment[3]	Total Estimated Benefits[4] for a 1,444,000 Car Fleet
Knuckles	18.16 mil/yr	1.94 mil/yr	1.4 mil/yr	18.1 mil/yr
Couplers	31.92 mil/yr	3.31 mil/yr	5.6 mil/yr	34.5 mil/yr
Yokes	12.62 mil/yr	0.85 mil/yr	2.2 mil/yr	13.2 mil/yr

Notes:

1. Ref. 1, Table 24
2. Delay time from Ref. 1, Table 19 projected to a full year and multiplied by \$185.82/train hr.
3. From Table 23 of Ref. 1. Direct costs to apply to Cause Code 430 for knuckles, Cause Codes 432 and 436 for coupler, and Cause Code 434 for yokes. Cost data are multiplied by 2.0 to account for unreported costs associated with loading damage and cleanup.
4. Maintenance and train delay costs apply to a 1972 fleet of 1,717,000 cars and are scaled to the smaller projected fleet of 1,444,000 cars by the ratio 1444/1717. Similarly derailment costs are for a 1977 fleet of 1,667,000 cars and are scaled by 1444/1667.

Cost/Benefit

For knuckles, the stream of investment and benefits would appear like that illustrated in Fig. 1 at the beginning of this report. Initially, there would be little benefit from using Grade E knuckles since failure rates are very low during the beginning of a component's life (see Fig. 21 of Ref. 1). In fact, we assume benefits would begin to accrue after the 9.19 years required on the average to replace all existing knuckles. However, there would be an incremental cost of \$8.90 for each knuckle used as a replacement part or on new freight cars.

To estimate the costs and benefits, consider the projected baseline population of 1,444,000 cars and the failure rate given in Table 6. This requires a retrofit of 1,444,000 x 2 x 0.109 = 314,792 knuckles per year at an incremental cost of \$2.8 million. In addition, assuming a 30 year car life (or a replacement rate of 48,133 cars per year) an incremental cost for new cars of \$0.86 mil will be incurred. Beyond the 9.19 year replacement period the railroad industry will realize annual benefits of \$18.1 mil as indicated in Table 7.

When this stream of costs and benefits is evaluated, the resulting allowable costs per car are \$20.71. Thus the cost/benefit ratio is

$$(C/B)_{knuckles} = 0.43$$

which is a favorable ratio.

For couplers and yokes, there would be an incremental investment in new components only. Benefits would begin to accrue after 25.2 years for couplers and after 30 years (the life of a car) for yokes. From the financial model and the above data, we find that an incremental investment of $6.43 is allowable for a coupler and $1.77 for a yoke. The cost/benefit ratios are:

$$(C/B)_{coupler} = 2.37$$

$$(C/B)_{yoke} = 3.25$$

Neither is favorable.

2.3.5 Zero Slack System

It is well known that when brakes are applied to a train in a draft condition, the rear cars run in, accompanied by a rising buff force between cars. If the force becomes high enough, a derailment can occur. It is believed that the development of this force is exacerbated by the slack in couplers and draft gear. It has been claimed that trains typically run with 4 to 6 in. of slack per car, or up to 50 ft for a 100 car system [17]. It is also believed that if this slack could be eliminated, through tight fitting couplers and a draft gear lock-out device, dynamic inter-car forces could be reduced, resulting in better train braking. Here we will consider the constituents of coupler and draft gear slack and the effects of their reduction.

Sources of Slack

Coupler slack is the free travel between draft and buff conditions for a pair of mated couplers. The following shows the minimum and maximum slack for new and worn couplers [3].

	New	Worn to Condemning Limits
Type E	25/32 in.	2-17/32 in.
Type F	3/8 in.	1-3/8 in.

New draft gear preloads couplers and should not allow for additional slack. However, such slack may develop as the draft gear ages and wears. There do not appear to be data available on this effect. Draft gear compression under load may range from about 2-1/2 in. for friction draft gear to 15 in. for end of car hydraulic cushioning equipment.

Dynamic Analysis

The effects of eliminating coupler slack and locking out draft gears in trains under way must be analyzed to determine if the desired benefits in fact occur. This system is sufficiently straight forward that a simple analytical model may be postulated and evaluated. First we consider the development of braking forces under worst-case conditions, namely the application of emergency brakes. Then we determine the resulting inter-car force levels.

Fig. 19 shows the buildup of brake cylinder pressure in each car as a result of an emergency application. The pressure vs time wave form is identical for each car, but delayed by about 50 msec from one car to the next. Because of this, the spatial distribution of brake cylinder pressures among cars at any given time is simply the reverse of a wave shown in Fig. 19. This is illustrated in Fig. 20.

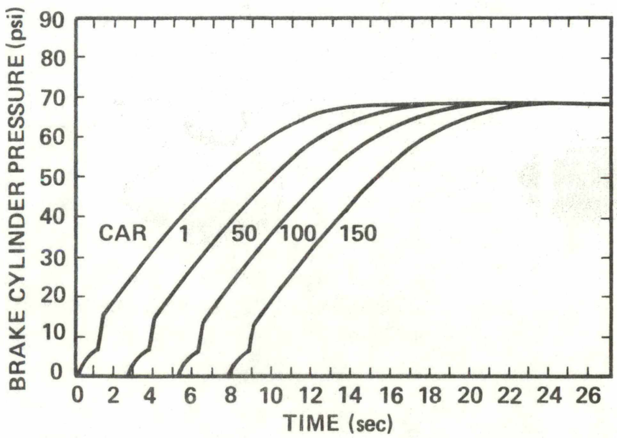

FIGURE 19. BRAKE CYLINDER PRESSURE BUILDUP ON A 150 CAR AIR BRAKE TEST RACK [18]

The relationship between brake cylinder pressure and retardation force may be estimated for a specific example. Consider a train made up of 100 fully loaded 100 ton capacity cars, each of which would have a gross weight of 130 tons. If each car satisfied the AAR minimum net braking ratio (NBR) requirement of 6.5% at a 50 psi brake cylinder pressure, the total normal force on all wheels of a car would be 16,900 lb. For a friction coefficient

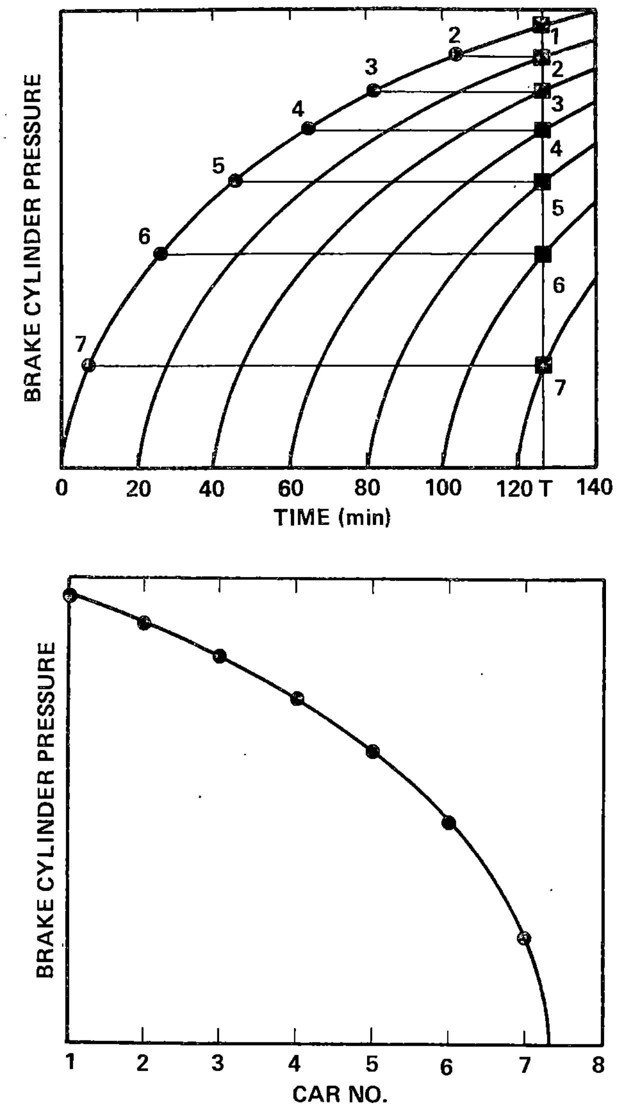

FIGURE 20. ILLUSTRATION OF THE DEVELOPMENT OF THE DISTRIBUTION OF BRAKE CYLINDER PRESSURES AMONG CARS

of 0.3, the retardation force would be 5000 lbs. That is, the retardation force (in lb) is about 100 times the brake cylinder pressure (in psi).

Using the above relationships, the distribution of retardation forces over a 100 car train may be readily determined and is shown in Fig. 21.

The inter-car buff force, P, generated by the retardation force, may be estimated with the aid of the free body diagram shown in Fig. 22. F_1 is the total external force applied to the front segment of the train of mass m_1 and F_2 the total force applied to the

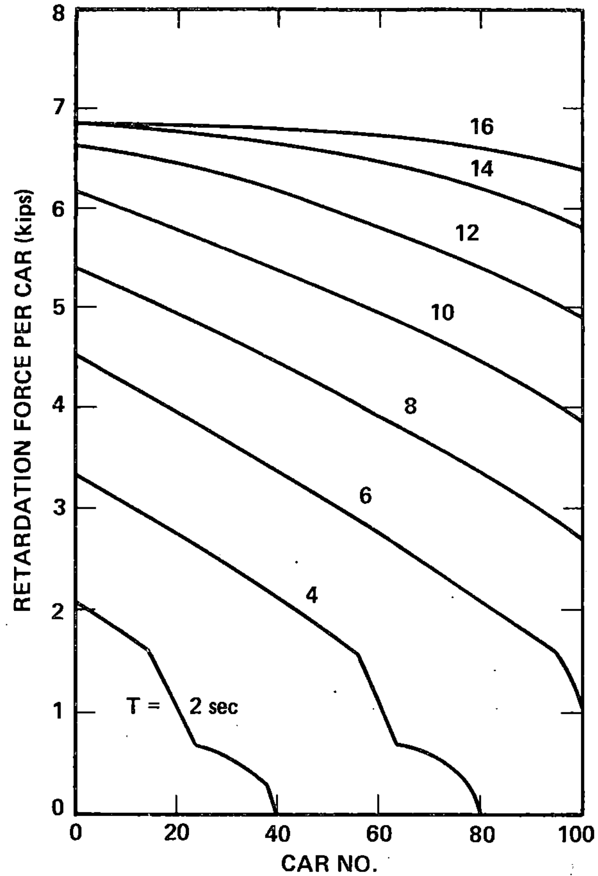

FIGURE 21. DISTRIBUTION OF RETARDATION FORCES ALONG A TRAIN MADE UP OF 100 CARS WEIGHING 130 TONS EACH

rear segment m_2. The equations of motion are

$$F_1 - P = m_1 d \quad (1)$$

$$F_2 + P = m_2 d \quad (2)$$

where d is the deceleration of the train. Solving for P gives

$$P = \frac{F_1 - (m_1/m_2)F_2}{1 + m_1/m_2} \quad (3)$$

If we evaluate the force in the middle of a train made up of uniformly loaded cars, $m_1 = m_2$ and

$$P = 0.5(F_1 - F_2) \quad (4)$$

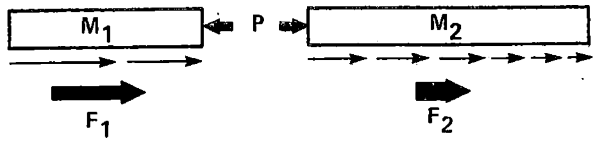

FIGURE 22. FREE BODY DIAGRAM OF TRAIN SEGMENTS

The force F_1 may be thought of as the area under the force distribution curve in Fig. 21 from 0 to the 50 car point and F_2 the area from the 50 to 100 car point. From this, and Eq. 4, the curve in Fig. 23 is generated, which shows some interesting effects. The buff force starts at zero and rises to a peak of 52 kips after about 5 sec. It is at this point that the force difference reaches its maximum. The force level drops rapidly as the set-up transient passes from the last cars and then more slowly during about 7 to 12 sec intervals when force levels rise at nearly the same rate. After that, the force difference declines more rapidly as brake forces in front cars approach asymptotic levels and forces in rear cars are still developing. Finally, the buff force gradually approaches zero as the retardation force distribution becomes uniform.

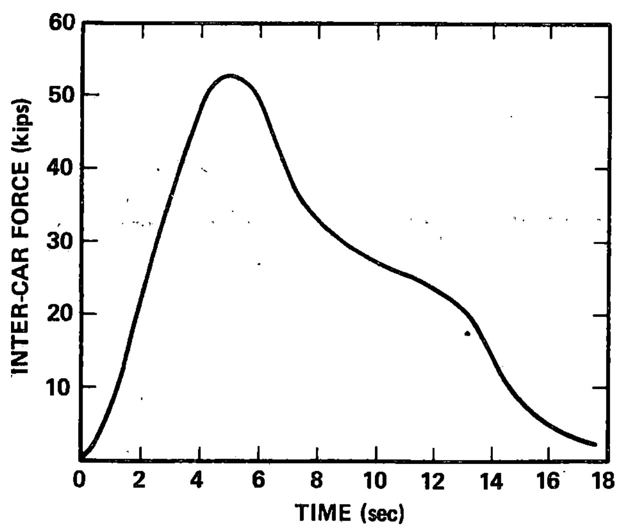

FIGURE 23. INTERCAR FORCE VS TIME WHEN EMERGENCY BRAKES ARE APPLIED TO A ZERO-SLACK TRAIN

The 52 kip buff force level reached in Fig. 22 is rather small compared to measured [19] and simulated [20] force levels determined for comparable train consists. Moreover, draft gear lockout prevents the development of draft forces, for which draw gear components will fail more readily.

Benefits

The above analysis shows that a zero slack system could be beneficial in that it would likely reduce loads over those that are presently experienced. As an upper bound, we may assume that such a system would prevent draw gear failures, with the following benefits:

Train delay	-	$ 6 M
Derail & Broken train coll.	-	12 M
Coupler, yoke & knuckle maintenance	-	12.5 M
		$30.5 M/yr

To develop an upper bound estimate of the allowable costs for implementing a zero-slack system, we assume:

- The fleet would be retrofit over a 10 year period with cash flow indicated in Fig. 24
- All of $30.5M indicated above could be eliminated.
- Retrofit costs are twice the costs of equipping new cars

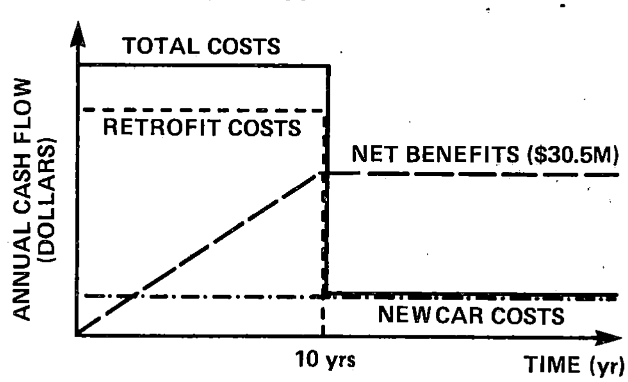

FIGURE 24. POSTULATED CASH FLOW FOR ZERO SLACK SYSTEM

Using this scenario and our financial model yields the following results

	Per Car	Per Draw Gear
Retrofit	$91.00	$45.50
New	$45.50	$22.75

Keeping in mind the upper bound nature of these results, we conclude that it is highly unlikely that it will be feasible to develop a draft gear lock-out and zero slack coupler for $22.75.

2.3.6 Mechanical load sensors

For the vast majority of existing freight cars, braking forces are independent of the load carried. Accordingly, a train made up primarily of empty or lightly loaded cars is capable of stopping in considerably shorter distances than an otherwise identical train which is fully loaded. In mixed consists with loaded and empty cars, large inter-car coupler forces can be created due to under-braked loaded cars and relatively over-braked empty cars.

The AAR specifies the following limits on net braking ratios (NBR)* for new cars:

	Composition	Cast Iron
Max.NBR (empty)	30%	53%
Min.NBR (loaded)	6.5%	13%

It is felt that higher values of NBR would result in increasing occurrences of wheel lock-up and skidding for empty cars. Lower values of NBR for loaded cars would reduce braking effectiveness. The NBR range indicated above allows for a car gross to tare weight ratio of 4.6 for cars equipped with composition shoes and 4.1 for cars with cast iron shoes. This accommodates most - but not all - cars in existing service. Therefore, increasing the loaded car NBR would decrease allowable gross-to-tare weight ratios,which would adversely impact a train's freight carrying capacity.

Description of system

The following types of mechanical load-compensating systems may be installed on freight cars to permit the application of braking forces that are related to the weight of the car:

- o Load/empty
- o Load proportional

In addition, electrical load proportional systems could be developed and integrated with electrical brakes as will be discussed in Sec. 3.2.

One type of load/empty system is available that provides full braking force to a loaded car and 60% of full braking force to an empty car. A schematic diagram of this SC-1 system is illustrated in Figure 25. In addition to the normal brake system, it consists of a load sensor valve, a load proportional valve, and an equalizing reservoir. The load sensing valve is typically mounted on the car body bolster above the side frame. With each brake application the sensor arm is depressed to detect the distance to the side frame. For loaded cars, the arm will not travel far and the load/empty system does not affect braking. For empty cars*, the arm moves sufficiently far to activate the load proportional valve. This is a pressure reducing valve that maintains the brake cylinder pressure at 60% of that in the supply line from the ABDW control valve.

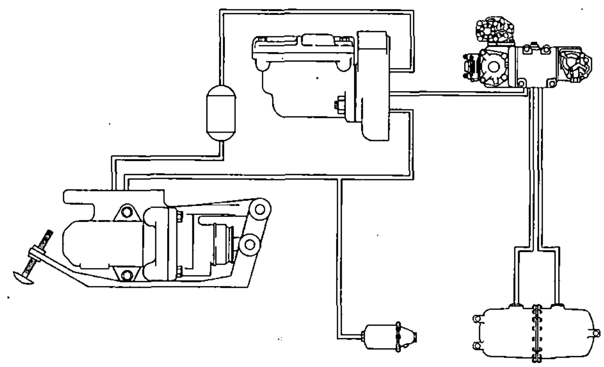

FIGURE 25. SCHEMATIC DIAGRAM OF SC-1 EMPTY & LOAD FREIGHT BRAKE EQUIPMENT

A version of a mechanical load-proportional system, supplied by the Swedish SAB Co., is shown schematically in Fig. 26. A pneumatic weighing valve uses air from the auxiliary reservoir to develop an output pressure proportional to force generated by a suspension spring. This pressure is transmitted to a pneumatic cylinder where a spring-loaded piston is activated. The piston positions a fulcrum which determines the mechanical advantage of the brake rigging. In practice, one weighing valve is used at each end of the car and the pneumatic cylinder is controlled by the valve generating the lowest pressure. Over-braking the car (with attendant wheel lock-up) is thereby avoided.

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* The NBR is the ratio of the sum of the normal forces on the brake shoes to the weight of the car.

* Actually, those for which truck spring deflection is not more than 20% of the total stroke.

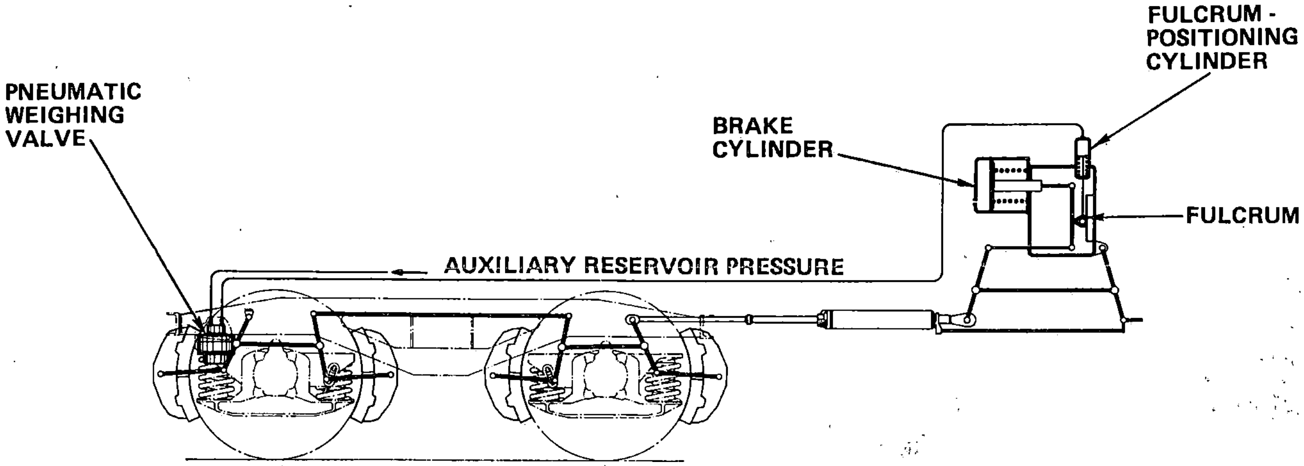


FIGURE 26. SCHEMATIC DIAGRAM OF A LOAD-PROPORTIONAL BRAKING SYSTEM

Implementation Scenario

We assume the SC-1 load-empty system is implemented uniformly during a 10 year period. Benefits will grow linearly with time, reaching a steady level after the 10 year implementation period. We also assume retrofit and new car costs are the same.

Unit Cost

The cost of SC-1 load empty equipment is as follows:

Components	$305
Installation (est)	100
Total	405

Benefits

An upper bound estimate of benefits is as follows.

Collision avoidance	$30.5M
Reduced Train Delay	6
Derailments and broken train collisions	12
Draw gear maintenance	12.5
Total	$61.0M
Maintenance*	(23)
Net Benefits	$38.0M

Cost/Benefit

The annual net benefits of $38M and the implementation scenario described above justify a per car investment of $51. The estimated/allowable cost ratio is 7.94 which is quite unfavorable for fleetwide application. However, load sensors may be a good investment for specialized cars with high gross/tare weight ratios or for specialized trains.

*Four percent of $405 per car applied to 1,444,000 cars.

3. ELECTRICAL SYSTEM EVALUATION

The application of electronics to freight train control holds enormous promise but will also require a considerable investment for system development and conversion from the present pneumatic system. In this section, we consider these systems at two levels. First, a basic system framework is postulated on which it would be possible to build various electronically controlled subsystems. Second, the cost and benefits of subsystems like electronic brake application and monitoring and uncoupling are evaluated. The costs of the basic framework and the electronic subsystems are estimated and reported separately.

3.1 Basic System Framework

The three major elements of an electronic system are (1) a monitoring and control system, (2) a set of reliable electrical connectors, and 3) an energy storage system. Here we will consider candidates for each.

3.1.1 Monitoring and control system

The following have been taken as working guidelines for the formulation of this, or the evaluation of other monitoring and control systems, suitable for railroad braking and coupling:

1. There may be a large number of monitored points; therefore, complexity and cost should be minimized at remote points, even at the expense of complexity and cost at the control center.

2. Design for very high reliability; the inclusion of redundancy or parity-type checks is desirable.

3. The number of interconnecting wires should be minimal.

4. The system should be capable of being broken and reconstituted easily; e.g., addresses should be easily set.

5. The system should be failsafe.

6. The system should be very adaptable, so as to be suited to very modest installation, yet capable of handling an almost unlimited number of points or complexities.

Consider the monitoring and control system to consist of "N" points. At each point we want to observe a state, and be able to command a change. The state at the monitored points has been limited to a binary description, as represented by an electronic high/low logic level, or a mechanical switch closure/nonclosure. This would apply directly to coupler open/close or brake application/release positions. If the monitored point is the closure/nonclosure of a micro switch representing a mechanical state under scrutiny, no preprocessing is necessary. However, as an alternative example, the monitored state might represent an appropriate/inappropriate vibrational intensity and preprocessing might include an accelerometer, preamplifier, filter, integrator, and threshold detector, in order, for example, to detect a local derailment.

The "observation point" can also be defined encompassing more than one functional block, thereby representing more than a single binary state. For example, a "state" represented by an analogue measure of brake cylinder pressure level could be accommodated by A-D conversion and assignment of each digital bit to an observation point. The usual limitations of bit size and sampling rate would, of course, apply, but the analogue waveform could be reconstructed at the control center.

Since we have assumed that the system may be called upon to monitor a very large, and perhaps variable number of points, we have considered only serial data transmission techniques. In order to understand better the types of problems which might be encountered and thereby to gain insight into potential solutions, a preliminary, or speculative, electronics design was done.

<u>A Speculative System Design</u>

Referring to Fig. 27, consider a monitoring and control system consisting of "N" points. Each point is interconnected to all others in a serial fashion, using six electrical lines: an interrogation line, a status line, an address line, a command line, common, and power. (Power might be carried on one of the other lines.)

Each monitored point has one electronic functional block which performs the following:

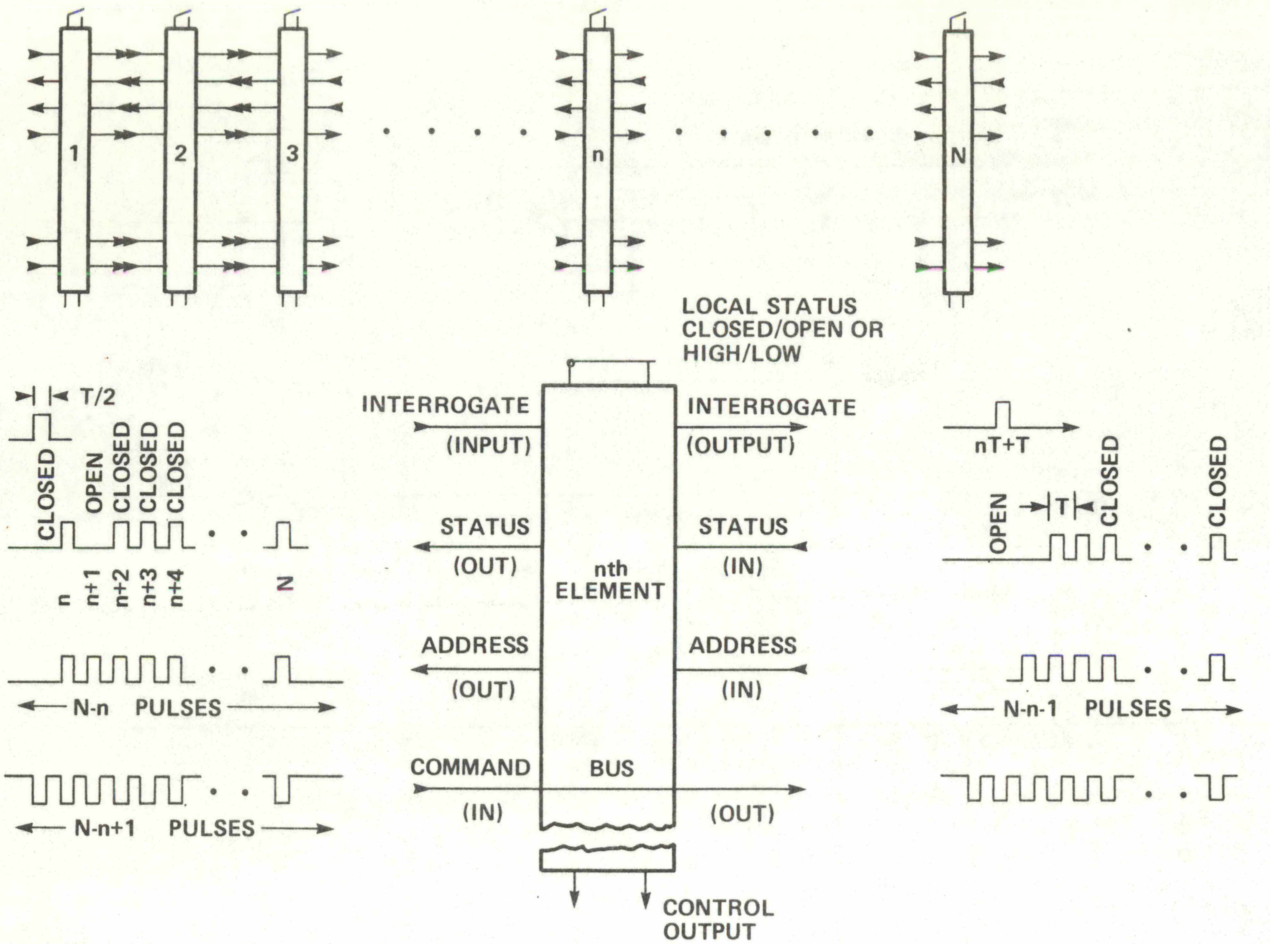


FIGURE 27. MONITORING AND CONTROL SYSTEMS

1. Upon receipt of a high-state pulse on the interrogation line:

2. It waits "T" seconds, and emits a delayed high-state pulse on the interrogation line to the next module;

3. if the local status under observation is high (closed switch), it returns this delayed high-state pulse to the previous module on the status line;

4. it transmits all pulses received on the status line from successive modules to previous modules, substantially unaltered;

5. it clears the local address register, then assumes as its address the sum of all pulses which follow on the address line; one for each following module and one for itself.

6. Upon receipt of its "address" on the command line, its output state goes high to accomplish a control function.

All of the logical functions can be accomplished relatively easily and inexpensively, using standard digital components. For example, in Fig. 28, there is shown one such circuit.

<u>A Bilateral Configuration</u>

As previously stated, a design goal is that the system should be capable of being broken and reconstituted easily. If one or more of the above modules are removed from the chain and reinserted <u>backwards</u>, they will not function properly. The ability to function in a bilateral fashion is necessary in railroad coupling and braking, since railroad cars do not have a "front"; i.e., they operate in either direction and might be coupled into the train in either orientation.

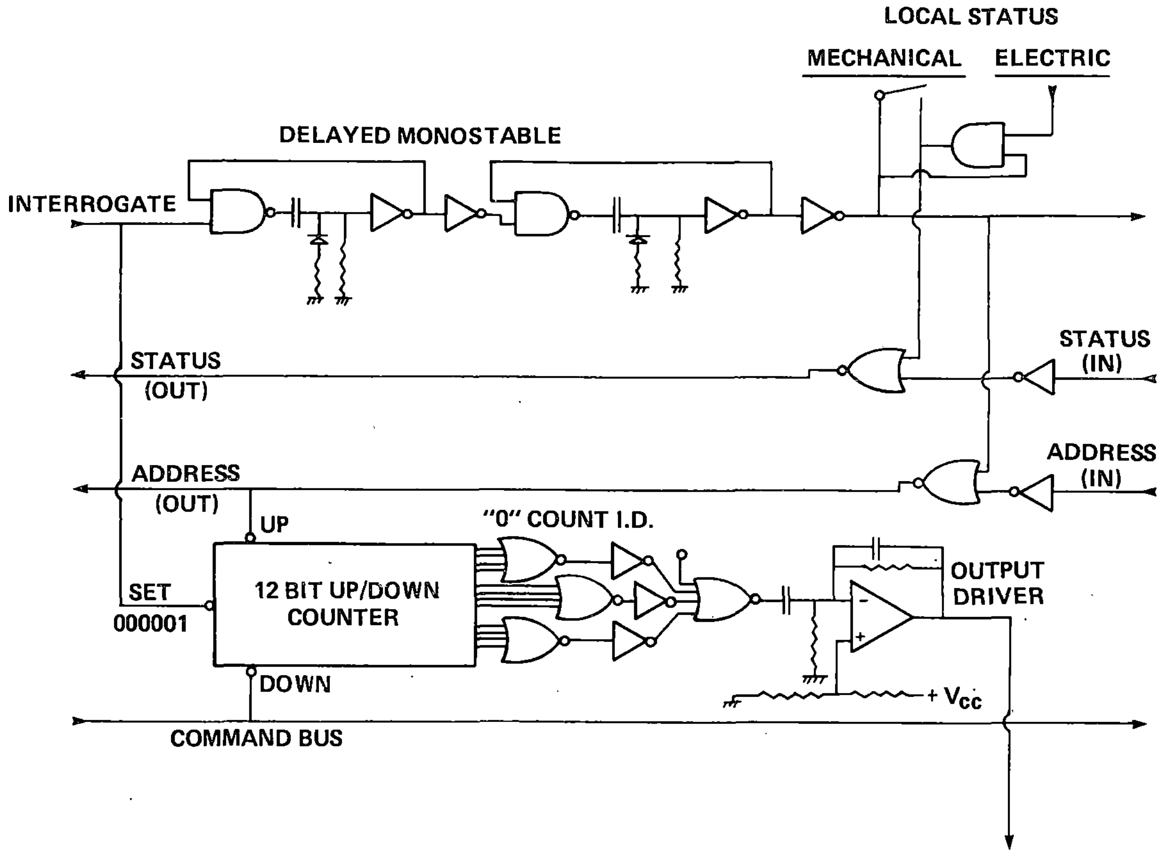


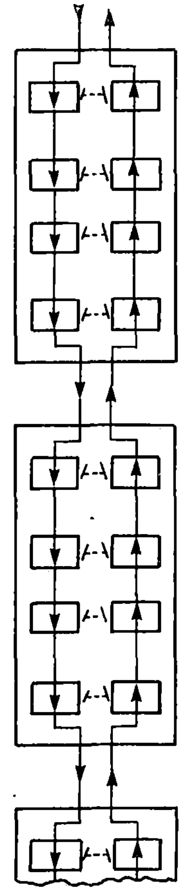
FIGURE 28. ELECTRONIC CIRCUIT FOR MONITORING AND CONTROL MODULE

In this case, a redundant system could be used, as shown in Fig. 29. The electromechanical couplings between cars could be offset from the centerline of the car. If the car is reversed, the systems employed simply reverse. An advantage of this approach is that a completely redundant system is accessible at the caboose or, if the intercar coupling is made to jumper systems in the absence of a coupling (as shown), then a redundant reading is accessible at the control center in the locomotive, since the two systems are in series, and the n^{th} element and the $2N+1-n^{th}$ element represent the same monitored function. Redundancy of control function addressing could also be provided, whereby a control function at the n^{th} point could not be accomplished without a virtually simultaneous 2N+1-n command as well, greatly reducing the likelihood of a false alarm.

Control Center Interpretation

A time serial system, as outlined above, could readily be interfaced to a micro- or minicomputer control and display. Interpretation of the "status" output is very straightforward. A status readout can be initiated at any time simply by applying a short duration high-state pulse to the interrogation input of the functional block. A bucket brigade reaction ensues, interrogating each successive position at an interval of T seconds. The presence of a high-stage status pulse might indicate a "normal" state. In this case, a dropout at interval mT would indicate an abnormality at position "m", etc.

Breaking the chain at any point merely truncates the series and limits operation to the interval between the control center and the break. Addresses represent position with respect to the end of the chain, and are refreshed with each interrogation. It follows that power intermittencies are not critical unless they occur during an interrogation avalanche (perhaps 0.5 seconds), in which case it is simply necessary to reinterrogate.



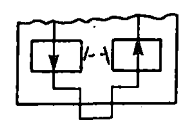


FIGURE 29. BILATERAL CIRCUIT

The price of monitoring and control modules is estimated as follows. First, the cost of each electronic component is obtained from manufacturers and summed. The result is scaled up by the following factors:

10% extra for miscellaneous

20% extra for engineering expansion (i.e. additions that become needed only as the system is developed)

3% line stock

3.3 empirical multiplier relating sale price to total material costs.

The result of multiplying the above factors together is a 4.46 factor on component costs.

The cost of the monitoring system is as follows.

Dual Monostable MM54C221	$1.00
Quad Switch CD4016E	.80
Resistors (2 at 10 cents ea.)	.20
Capacitors (2 at 15 cents ea.)	.30
Board	2.00
Cover	.40
Terminals	.70
Total Materials	$5.40
	x 4.46
Total Price	$24.00

The cost of a monitoring and control module is as follows:

Control components	
Converters (D4029 (3 at $1.00)	$ 3.00
Retriggerable Monostable CD4047	1.00
Driver Op. Amp (e.g. LM 318)	1.25
Resistors and Capacitors	.75
Board	2.00
Total Materials	$ 8.00
	x 4.46
Control Price	$36.00
Monitor Price	24.00
Total	$60.00

3.1.2 Connectors and wiring

There are at least three basic types of electrical connectors that may be used to transmit power and signals between cars: (1) spring-loaded pin, (2) pin and socket, and (3) inductive.

Spring-Loaded Pin

Most rail transit car couplers use spring-loaded pin connectors. As illustrated in Fig.30, a set of pins is typically attached to a coupler and butt against a companion set when couplers are connected. A cover is often used to protect pins against environmental contamination such as rain, snow, or dirt. The cover is pivoted in such a way that it is pushed aside automatically when coupling takes place.

Pin and Socket

Fig. 31 shows a pin and socket arrangement that is used for underwater connectors. Even though water may enter the connector, the electrical resistance from one pin to another is so much

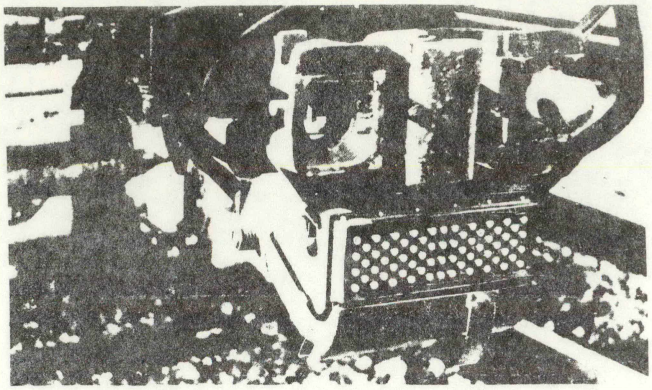


FIGURE 30. AUTOMATIC COUPLER SYSTEMS FOR RAPID TRANSIT CARS

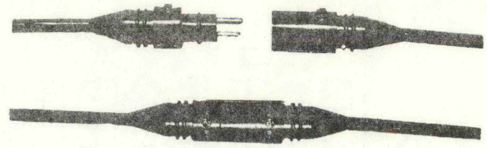


FIGURE 31. CABLE CONNECTORS

higher than the pin-to-mating socket resistance, the connector functions well. While the connector shown in Fig. 31 is designed for 1- to 4-contacts, there is no reason that a 6- or 12-pin connector appropriate for monitoring and control could not be developed.

The pin and socket connector presents many of the same advantages and disadvantages as the conventional air line connector. It would be relatively inexpensive, and would pull apart automatically. However, it would also require manual coupling. This would not be a particular disadvantage if conventional airline connectors were retained for the indefinite future. If automatic air line connectors were developed and implemented, an automatic pin and socket (or other type) of connector would also be required.

Inductive Connectors

Research conducted at the Canadian Institute of Guided Ground Transport has shown that information can be transferred through a series of inductive couplers. As shown in Fig. 32, these couplers can be built into a conventional air line-type of "glad hand" for ease of connection and separation.

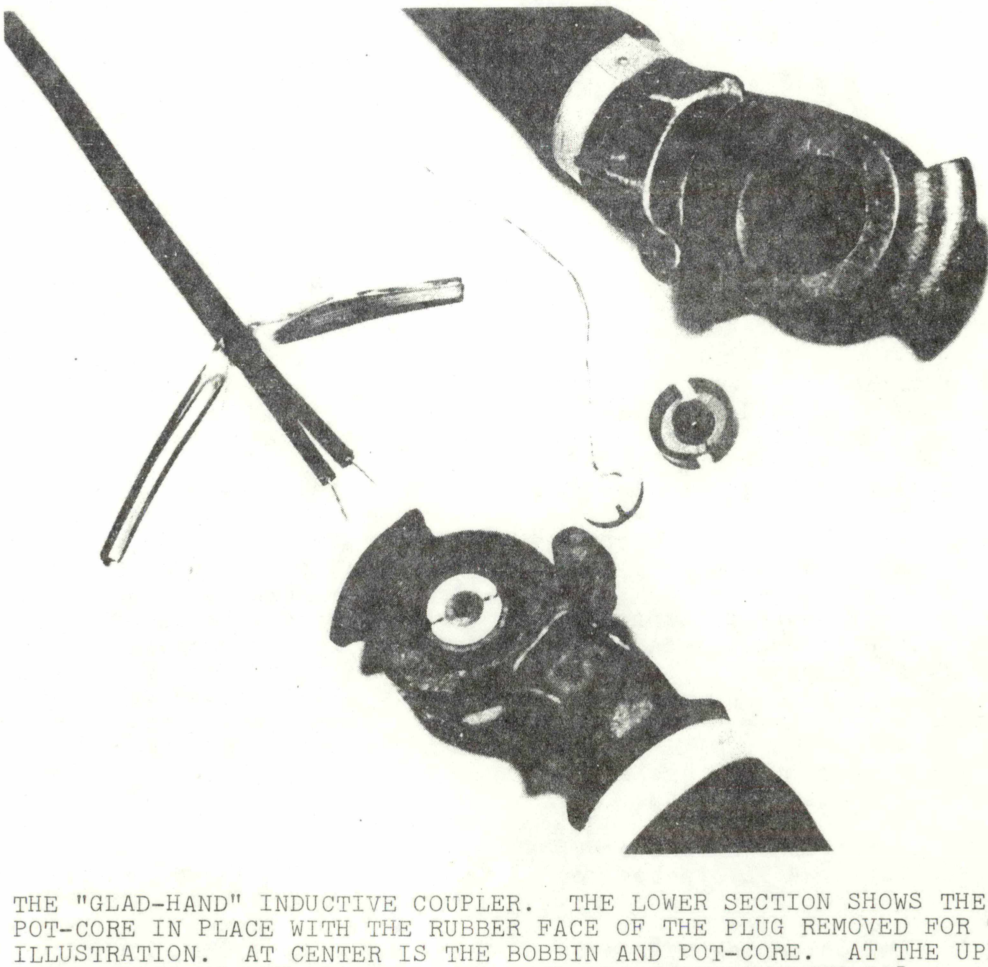


FIGURE 32. THE "GLAD-HAND" INDUCTIVE COUPLER. THE LOWER SECTION SHOWS THE COIL AND POT-CORE IN PLACE WITH THE RUBBER FACE OF THE PLUG REMOVED FOR THIS ILLUSTRATION. AT CENTER IS THE BOBBIN AND POT-CORE. AT THE UPPER LEFT-HAND CORNER IS THE SHIELDED TWO WIRE TRANSMISSION LINE [2]

However, at this stage of development, the inductive coil occupies the space normally used for air passage. Thus, this type of connector would supplement the airline connector.

The potential data rate for this system may be estimated from the transmission data shown in Fig. 33. The bandwidth of this system is about 10 kHz, which would allow up to approximately 20k bits per sec. This data rate would be adequate for the information requirements described in Sec. 3.1.1. However, multiplexing would be required to transmit, interrogate, status, address, and command signals over the single line.

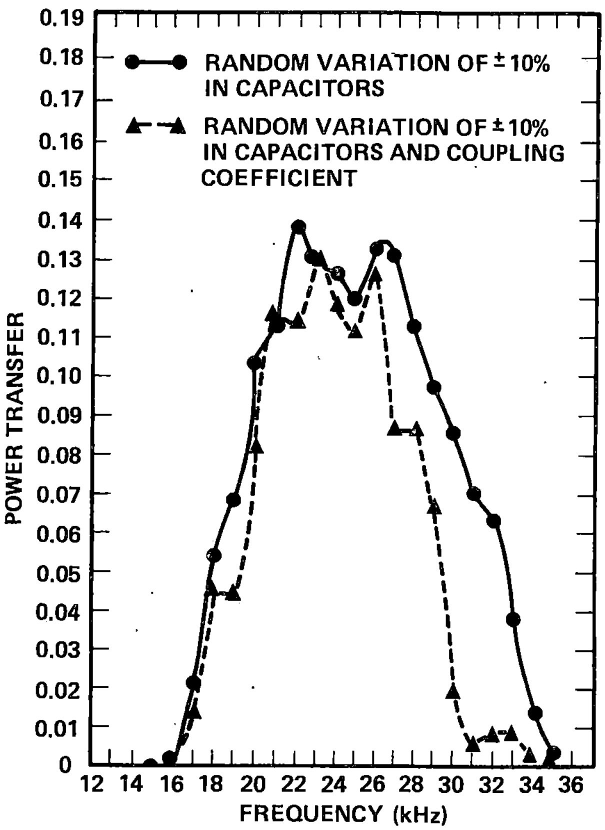


FIGURE 33. POWER TRANSFER AS A FUNCTION OF FREQUENCY FOR A STRING OF 100 COUPLERS [22].

Connector and Wiring Costs

The costs of the connectors described above cover a wide range. Spring loaded pin type connectors used on transit systems cost about $2000 per car set. This is for a 100 pin connector and costs for a 6- or 12-pin connector, manufactured in large quantity, would obviously be considerably less expensive. A pin and socket type of connector with six contacts cost about $30 when ordered in quantity. Two sets would be needed for the system depicted in Fig. 31 at a cost of $60. It is a bit premature to estimate the cost of the inductive connector.

The following is a lower bound estimate of connector and wiring costs:

Connectors (2 sets at $30 each)	$ 60
100 ft of six strand wire (at 25c/ft)	$ 25
Total Materials	$ 85
Installation	50
	$135

3.1.3 Energy storage

Most of the systems evaluated in this section are designed to function when cars are operating as part of a train. For these systems and conditions, pneumatic power is available from the brake pipe or the auxiliary reservoir, and electrical power may be supplied by wires extending from the locomotive through each car. However, many systems must also function when cars are disengaged from a train. Systems that allow remote uncoupling or brake application should also be readily activated at the car. For these, manual operation of solenoid valves would be feasible. Only two others, ultrasonic brake control and hybrid brake valves, require electrical power when cars are disconnected from trains. The ultrasonic brake control system needs electrical power only when the car is moving faster than about 4 mph. This power may be supplied by a wheel-mounted alternator. The hybrid brake system is the only one that requires electrical energy storage. An appropriate battery and charger will be discussed in Sec. 3.3.3 in conjunction with the evaluation of hybrid systems.

Probably the most obvious way to store pneumatic energy is simply to use the auxiliary air brake reservoir. It exists on every freight car and is kept charged by the brake line and control valve. Equally apparent, however, is the risk of using brake supply air to operate other systems. If these systems used a significant amount of air, less might be available for braking. In an extreme situation, a system that is piped into the auxiliary reservoir might fail and deplete the entire air supply in that chamber. But certain functions, such as remote uncoupling or the remote application of a brake lock, are not likely to be performed when auxiliary reservoir air is needed for brake application. Moreover, the emergency reservoir is always available for emergency brake application and the failure of the brakes on a single car will not influence stopping distance appreciably.

To determine whether a pneumatic actuator would materially affect the supply of air in the auxiliary reservoir, let us consider some size effects. At a nominal brake pipe pressure of 70 psi, a 2-1/2-in. diameter piston is capable of generating 344 lb of force which is probably adequate to apply or release a brake lock or uncouple a coupler. Assuming that a 3 in. stroke is adequate, the actuator would require 14.7 cu in. of volume. A 1/4 in. diameter air line running 30 ft (about half the length of a car) would require an additional 17.7 cu in. The total system requirement would be 32.4 cu in. or 1.3% of the auxiliary reservoir volume of 2500 cu in. A single stroke of this type of actuator would reduce the auxiliary reservoir pressure by about 1 psi. For systems of this size, we assume the auxiliary reservoir can be used with little risk. Clearly, this assumption would have to be re-examined in detail during the development of such a system.

Another important assumption that we make at the outset is that hardware would be redesigned to preclude the need to bleed air reservoirs. This would save time at the receiving tracks in yards where brakes are usually bled by car inspectors. Assuming that inspections would still be performed, but that the inspectors would not have to stop at each car to bleed brakes, we use our yard simulation model to determine the following annual benefits:

Labor savings	$ 7.2 M
EquipMent utilization	14.9 M
Total benefits	$22.1 M

3.2 Improved Operations

3.2.1 Remote-Controlled Coupler

One of the major limitations of contemporary couplers is that they must be uncoupled manually. Uncoupling requires a man to stand, walk, or run next to the car to be uncoupled and lift the uncoupling lever located on the right side of the car. Unless the coupler is in buff (or only slightly in draft), he will not be able to exert sufficient force to release the lock.

An electrical train line presents the opportunity to uncouple cars remotely from the locomotive cab. During flat switching, the engineer or a trainman in the locomotive could command couplers to open at the proper time. In hump yard classification, all couplers could be simultaneously opened while a string is pushed forward, thereby eliminating the need for a pin puller at the crest of the hump.

Description of System

A conceptual layout of a remote-controlled coupler system is shown in Fig. 34. Communication with a control center in the locomotive (or other remote station) is carried over the six-wire bus that extends the length of the train. The electronic controller serves both a monitoring and command function. When one of the couplers is to be opened, the controller opens the appropriate solenoid. Air flows from the auxiliary resevoir to the uncoupling mechanism, which activates the lock lift assembly. A micro switch senses when the coupler is open and signals the controller which, in turn, closes the solenoid, venting the uncoupling mechanism. Fig. 34 also shows electrical push buttons located at each corner of the car to facilitate uncoupling by a man standing adjacent to the car.

An example of an uncoupling mechanism, suitable for retrofit to E or F couplers, is shown in Fig. 35 [23]. It involves primarily a fluid-activated piston coupled to a rack and pinion mechanism. When air is introduced to the cylinder, the piston extends, rotating the pinion. The pinion rotates the lock lift assembly in much the same way as the rotation imparted by the uncoupling lever.

Implementation Scenario

The system is capable of being entirely added as a modular retrofit to cars equipped with a basic electronic framework (see Sec. 3.1). As with many of the other systems considered in this report, we assume that implementation takes place over a 10 year period. During that time, new cars are equipped with the system and cars that are expected to be operational by the end of the period are retrofitted.

Unit Costs

The following components would be required at the corresponding cost

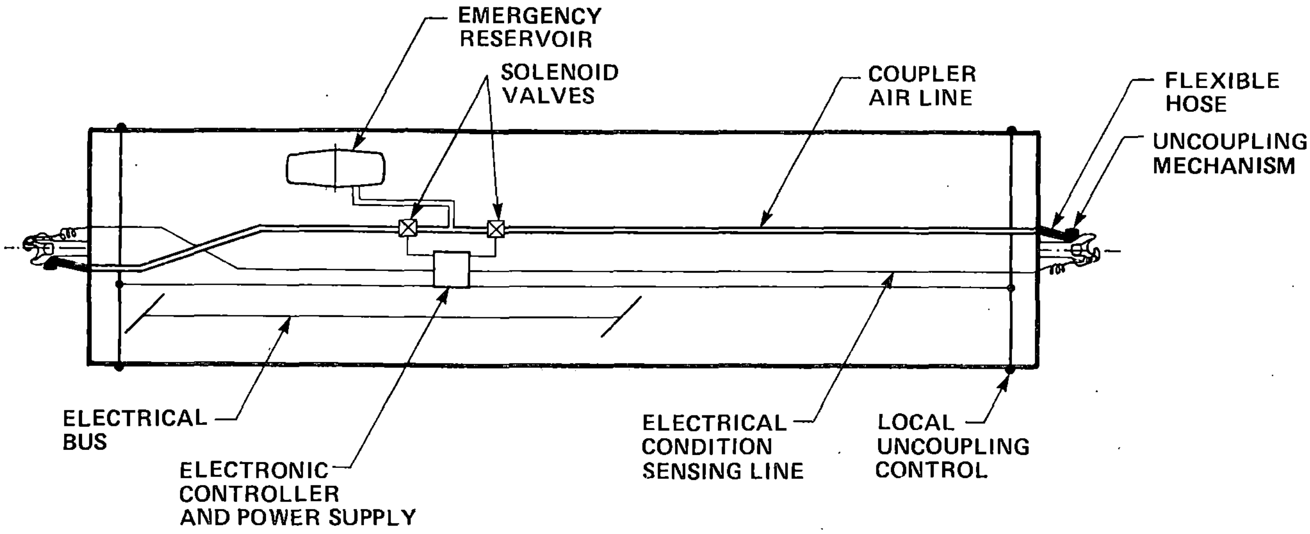


FIGURE 34. SCHEMATIC OF REMOTE-CONTROLLED COUPLER SYSTEM

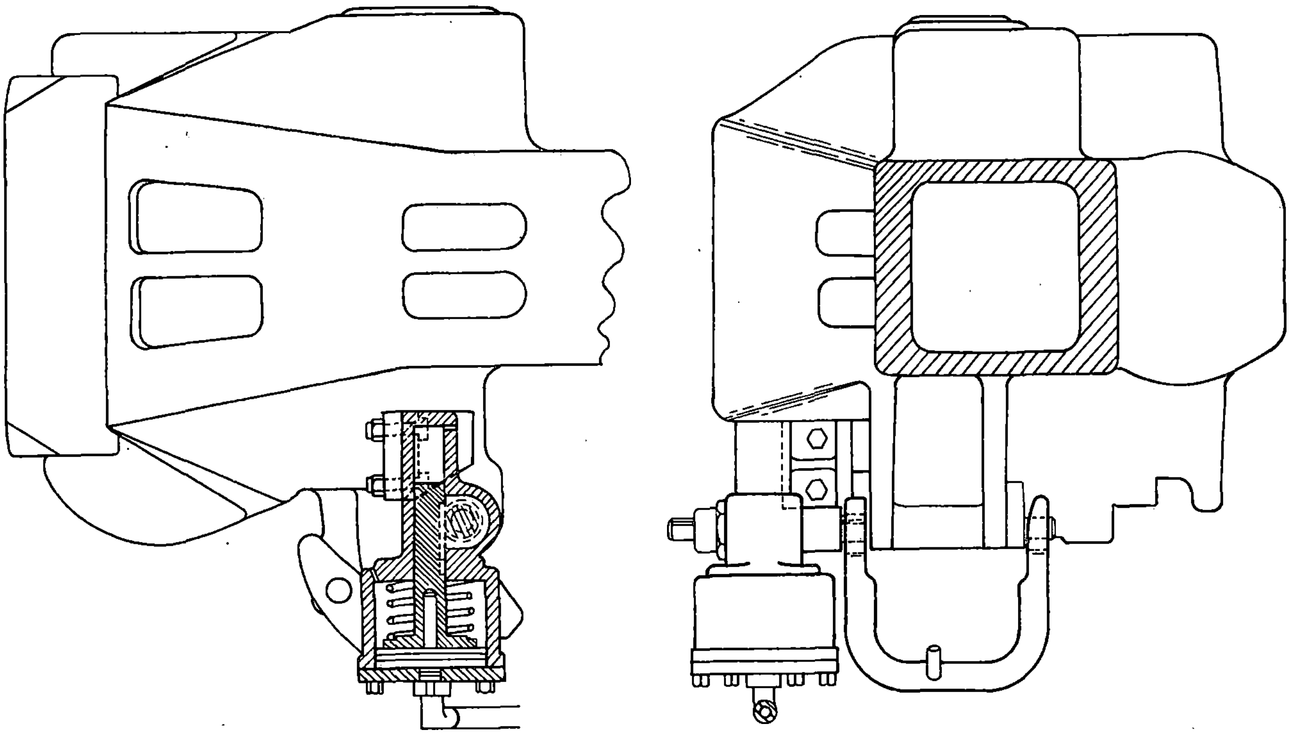


FIGURE 35. TWO-VIEW ILLUSTRATION OF AN UNCOUPLING MECHANISM ATTACHED TO AN E-TYPE COUPLER [23]

2 solenoid valves ($ 19 ea)	$ 38.00
4 electronic control modules ($68.5 ea)	274.00
2 uncoupling mechanisms ($200 ea)	400.00
150 ft. of 2 strand wire ($.08/ft)	12.00
60 ft. of 1/4 in. ID tubing ($.05/ft)	3.00
10 ft rubber hose ($.26/ft)	2.60
4 switches ($5 ea)	20.00
2 knuckle position sensing switches ($5 ea)	10.00
Total Equipment	759.60
Estimated Installation Cost	300.00
Total Cost Per Car	$1059.60

Benefits

Benefits are evaluated for two scenarios. For Scenario #1, benefits are assumed to accrue by eliminating the time required to

- open and close angle cocks (they would automatically close when the coupler is deliberately opened and open when the coupler is closed).
- open knuckles by manually activating the operating lever
- cross over and open knuckles

Gross Annual benefits are:

Labor	$23.5M
Equipment Utilization	58.9
Total	$92.4M
Maintenance Costs*	(61.2)
Net Annual Benefits	$31.2M

For Scenario #2 the above benefits are assumed to accrue and the crew size is reduced to three because there is no need for a pin puller. Gross annual benefits are:

Labor	$495.2M
Equipment Utilization	58.9
Total	$554.1M
Maintenance Costs*	(61.2)
Net Annual Benefits	$492.9M

When all cars are equipped with remote-controlled couplers, another important benefit accrues which has not been quantified here: the speed of classifying cars in a hump yard may be materially increased. Petracek, et al [24] point out that, in new hump yards, car classification would be performed at eight to ten cars per minute, instead of the present six cars per minute. This corresponds to an increase in hump speed to about 4.5-5.75 mph from the present value of 3.5 mph which is limited by the walking pace of the pin puller. This increase would significantly improve yard productivity and reduce labor and equipment utilization costs.

* Maintenance costs are estimated at 4% per year of $1059.60/car times 1,444,000 cars

Cost/Benefit

Assuming that retrofit costs are the same as new car costs, the following maximum allowable costs and estimated/allowable cost factors are developed

	Allowable Cost	Estimated Cost / Allowable Cost
Scenario #1	$ 86.92	12.2
Scenario #2	$1373.09	0.77

Clearly, this system is potentially viable only if it permits a reduction in crew size.

3.2.2 Remote Controlled Brake Lock

When cars are left standing for extended periods of time, it is often necessary to apply hand brakes to prevent them from rolling. This application consumes labor directly for the actual hand brake application, and indirectly for personnel to walk to a car or move from car to car if several brakes are to be set.

It is not enough to apply air brakes and leave them for any length of time. Air will gradually leak from cylinders, brakes will release, and cars can roll away with dangerous consequences. Accordingly, a device is needed that enables the remote application of brakes in a way that does not allow for a gradual release.

Description of System

Fig. 36 shows a conceptual design of a system that would allow brakes to be locked for an indefinite period. The system relies on the conventional brake cylinder and auxiliary resevoir. In addition, it uses an electronic control module, a four-way valve, a locking cylinder, and a ratchet and pawl mechanism.

During a running brake application, the pawl is held in the release position by a spring. The four-way valve vents to the atmosphere both the "lock" and "release" sides of the locking cylinder. This spring and valving arrangement is designed to minimize the likelihood of an unintentional brake locking during a running application. The over center spring provides a positive retaining force and the valve position prevents possible leakage from building up cylinder pressure.

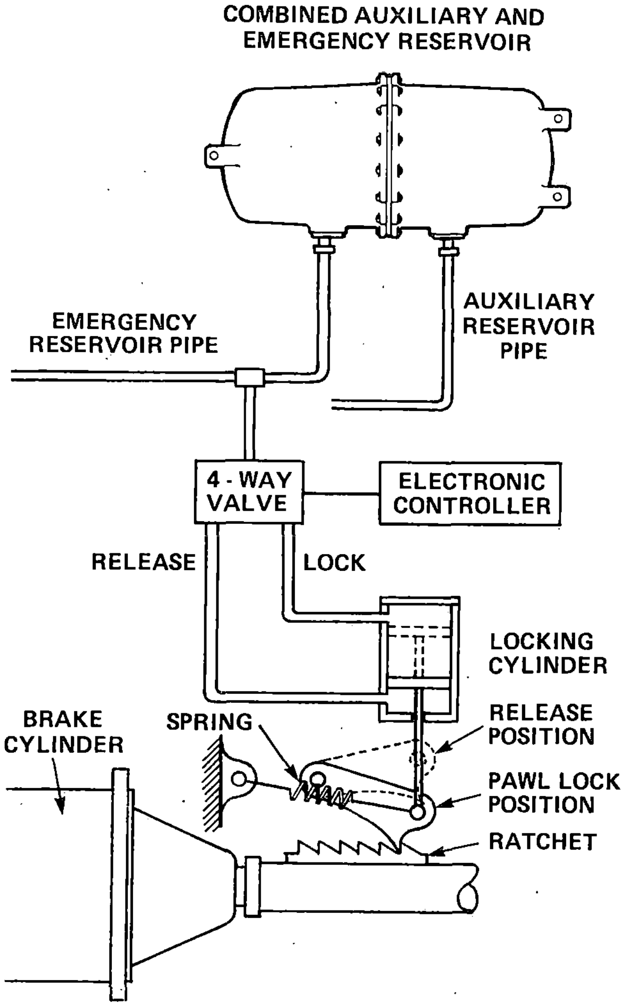


FIGURE 36. CONCEPTUAL DESIGN OF A BRAKE LOCKING MECHANISM

When the controller commands the brakes to lock, the valve supplies air to the top of the locking cylinder, engaging the rachet and pawl. The spring now holds the pawl in place. Even if air were to leak from the locking cylinder or the brake cylinder, the pawl could not release. When a brake release is commanded, the top of the locking cylinder is vented and compressed air is supplied to the bottom of the piston.

As for the remote controlled uncoupler, this system is amenable to control by means of electrical switches at convenient locations on the car.

Implementation Scenario

The system is capable of being entirely added as a modular retrofit to cars equipped with a basic electronic framework (see Sec. 3.1). We assume that implementation takes place over a 10 year period. During that time, new cars are equipped with the system and cars that are expected to be operational by the end of the period are retrofitted.

Unit Costs

The following components would be required at the corresponding cost

Locking cylinder*	$ 38.75
Four-way valve*	25.13
Pneumatic tubing* (20 ft at 60 cents/ft)	12.00
Electronic controller (2@$60)	120.00
Ratchet and pawl	50.00
	$245.88
Installation	100.00
Total	$345.88

Benefits

Time to set and release hand brakes with associated walking from car to car is eliminated.

Gross annual benefits are:

Labor	$259.1M
Equipment Utilization	463.1
Total	$722.2M
Maintenance Costs**	(20.0)
Net Annual Benefits	$702.2M

Cost/Benefit

Assuming that retrofit costs are equal to new car costs, the maximum allowable cost is $1957. The estimated/allowable cost ratio is then 0.18, which is extremely attractive.

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* Manufacturer's quote

** Four percent of $345.88 times 1,444,000 cars

3.2.3 Ultrasonic Brake Control

During the 1960's, R.F. Trevisin developed and field-tested 20 cars equipped with ultrasonic brake control systems [25]. The purpose of these "Sonicars" was to reduce car impact speed to 4 mph or less during classification and thereby avoid damage to landing. As shown in Fig. 37, the system appeared to have functioned quite well in controlling impact speed.

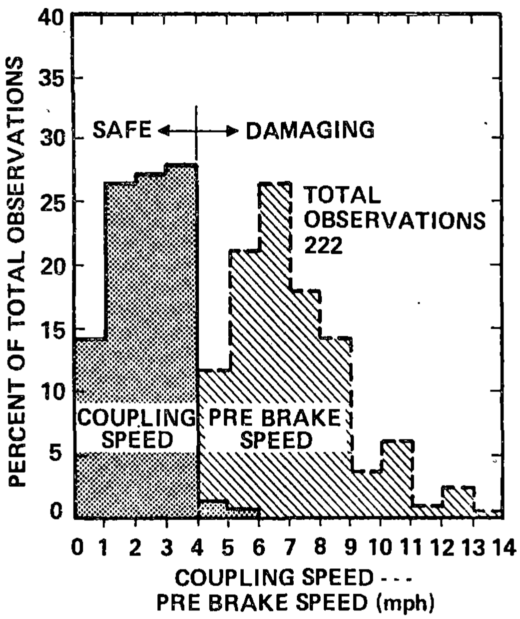

FIGURE 37. EFFECT OF SONICAR ON IMPACT SPEED [25]

Fig. 38 illustrates the main components of Trevisin's system, which replicated many of the standard brake system components. The ultrasonic system had its own reservoir and electronically controlled brake valve. In addition, it incorporated an alternator that was powered by one of the car's wheels, an electronic control box, and one ultrasonic transducer at each end of the car. The system is activated only when the brake pipe is at atmospheric pressure and several other conditions are met.

Trevisin reported several developmental problems associated with his Sonicars, but none that appeared to be fundamentally insurmountable. Slippage (and excessive wear) of the rubber drive wheel for the alternator, solenoid valve leakage, and separation of cables from transducers were among these.

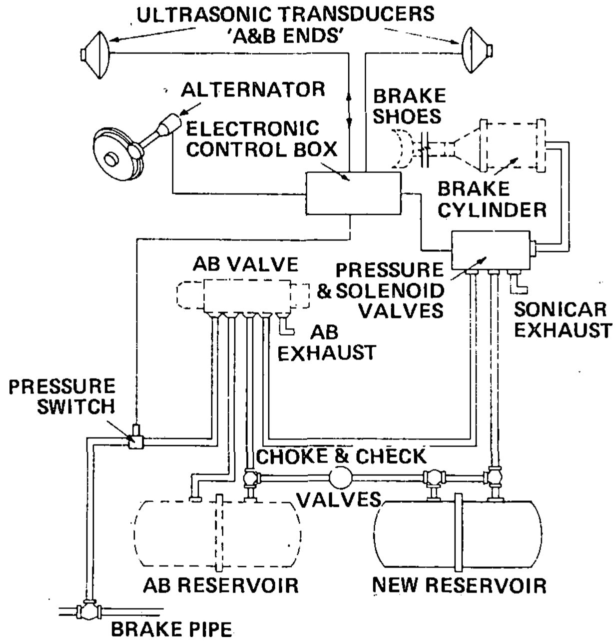

FIGURE 38. FUNCTIONAL BLOCK DIAGRAM OF SONICAR SYSTEM [25]

Description of System

Any ultrasonic brake control module, added to an already electrified car, could have several advantages over the Sonicar system. It would not require an independent wheel-driven alternator and could probably use the auxiliary air reservoir and the brake control valve. It would still require the addition of ultrasonic transducers and an electronic control system.

Implementation Scenario

Ultrasonic brake control systems are assumed to be installed only on cars that presently require special handling. If the system were sufficiently reliable, these cars could then be humped or kicked during classification with minimal risk. We assume that an ultrasonic brake control system would be incorporated only in these new cars that would otherwise be designated for special handling. The investment and benefit cash flows are illustrated in Fig. 39.

Unit Cost

The estimated cost of the Sonicar system in 1970 was approximately $2000 per car [26]. Because of inflation this value would approximately double when extrapolated to 1979. However, efficiencies of hardware utilization, as discussed above, and advances in the state-of-the-art of electronics and ultrasonic transducer technologies would

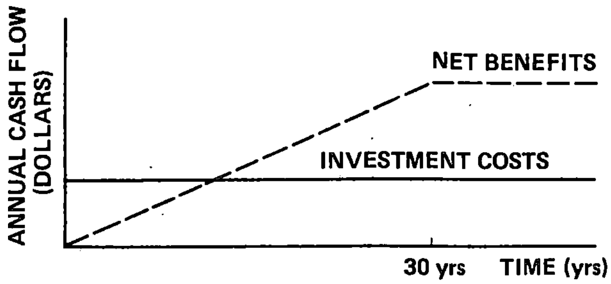

FIGURE 39. CASH FLOWS ASSOCIATED WITH AN ULTRASONIC BRAKE SYSTEM

probably offset this inflation. Accordingly, we assume the incremented cost of the system would remain at $2000.

Benefits

The benefit for this system is the reduction in time to special handle a car from 10 min to 0. This applies to 5% of cars classified. Benefits are as follows:

Labor	$ 74.3M/yr.
Equipment Utilization	129.3
Total	$203.6M
Maintenance Costs*	(5.8)
Net Annual Benefits	$197.8M

Another potential benefit of ultrasonic brake control to the railroads is the ability to carry fragile cargo with less likelihood of damage. This would not only reduce the loss and damage payments for which Trevisin originally developed the Sonicar, but would attract additional freight to the railroads with concomitant benefits. Unfortunately, insufficient data are available to quantify this effect with an adequate level of confidence.

Cost/Benefit

Based on the above information, the maximum allowable cost per car is $5340. Thus, the estimated/allowable cost ratio is 0.37, which is very attractive.

*Four percent of $2000 applied to 72,200 cars

3.2.4 Train Condition Monitor

Parameters which might be monitored on a train include wheel vibration (bearing failure and/or derailment), drawbar forces, wheel slip, and power brake variables (brake pipe pressure and piston travel). Each of these has some appeal, and might be the subject of a detailed cost/benefit analysis. Monitoring the brake test is particularly attractive in view of the fact that this test is a classification yard critical path, and its elimination or speedup would represent immediate payback with regard to operating efficiency. Accordingly, we have taken a brake application monitor as the subject for evaluation.

System Description

During the power brake test, several conditions are inspected that are amenable to electrical monitoring. These include brake pipe pressure at the caboose, piston travel, angle cock position, and brake shoe application.

The power brake law requires pressure at the caboose to be no less than 60 psi and within 15 psi of feed valve pressure. A valve and switch arrangement could be developed for which the operating railroad would "dial in" the feed valve pressure used on its trains. When the criteria at the caboose are met the valve could then close a switch, a condition that would be detected by the interrogation system.

To minimize the complexity of the electronics on each car, switches could sense piston travel and angle cock position and be wired in series. A candidate system is illustrated in Fig. 40. An extension of the brake piston rod with a linear cam, or raised portion, would close a switch when the rod is extended the required 7 to 9 inches, or some other appropriate distance. Rotary cams on angle cocks would close switches when they are in the fully opened position. A malfunction of any system would be sensed in the locomotive when brakes are applied and a man could be dispatched to

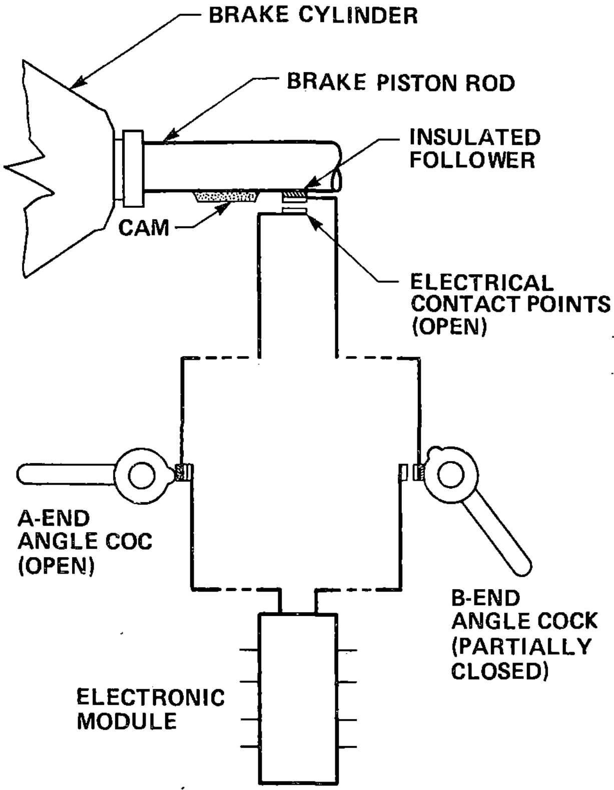

FIGURE 40. SCHEMATIC DIAGRAM OF A BRAKE CONDITION MONITORING SYSTEM

the appropriate car to determine the source of the problem.*

Other functions are performed during the power brake test that defy instrumentation. Fouled rigging or components that have nearly worn or corroded to failure can only be detected by visual inspection. Even brake shoe wear, while capable of being measured with transducers, is most easily checked by a visual inspection. We assume for purposes of this analysis that such functions will be performed at an inbound inspection or by a roll-by outbound inspection and that neither will materially affect inspection costs.

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* One drawback of this system is that it effectively places equal importance on angle cock position and piston stroke. In fact, a closed (or partially closed) angle cock on a single car could have catastrophic effects whereas an incorrect piston travel on one car would have minimal effects. An alternate system could be designed which monitors angle cocks and piston travel separately and provides this information to the engineer.

On the other hand, the monitoring system described here could also be operational while a train is underway. This has the potentially very valuable feature of providing the train crew with information on malfunctioning cars in a timely fashion. These cars could be designated for repair at the next convenient shop facility.

Unit Costs

Costs per freight car for this system are estimated at

Microswitches: 3@$5.	$ 15
Mechanical activators on piston and angle cocks (est)	$ 50
Electronic modules 2@$24. ea.	$ 48
Wire: 100 ft@$.08/ft	$ 8
Total Materials	$121
Installation	$100
Total	$221

Benefits

The time required to perform a power brake test is assumed to be reduced to zero. Benefits annual are:

Labor:	$197 M
Equipment Utilization:	$295
Total:	$492
Maintenance Costs*	(13)
Net Annual Benefits	$479 M

Cost/Benefit

Assuming the new-car cost of this system is the same as retrofit cost, that the fleet will be converted in 10 years and that benefits will begin to accrue only after conversion takes place, the maximum allowable cost is estimated at $1334. This results in an estimated/allowable cost ratio of 0.17 which is very desirable.

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* Four percent of $221 times 1,444.000 cars

3.3 Improved Dynamics: Electrical Systems

In addition to their capacity for improving operations, electrical systems may significantly improve braking dynamics. Here we develop conceptually an electronic brake system capable of bridging the gap from present pneumatic to ultimate electrical braking. We also evaluate the potential of electrical load-proportional braking and consider the costs and benefits of passenger train type of electro-pneumatic brakes.

3.3.1 Electronic brakes

There are several inherent limitations associated with conventional air brake systems that could be overcome with electronically controlled brakes. The subsonic propagation speed of brake signals through the brake pipe restricts the speed with which brake cylinder pressure may be allowed to build up. (More rapid build up is technologically feasible but would cause unacceptably large intercar forces.) Also, brakes must be released completely and cannot be reduced in level of application. A graduated release could enhance train control during stopping and on grades. Electronic brakes, commanded simultaneously on all cars from the locomotive, could allow for quick application and release without lengthy time delays.

Implementation Scenario

For other systems evaluated in this report, it was convenient to conceive of and describe the systems first, and then think of how they might be implemented. They usually could be implemented as add-ons to existing cars (e.g., remote controlled coupler) or changed when existing parts wore out (e.g., wide range couplers). Air brakes are different, and we found it necessary to think about an implementation strategy prior to conceiving of a system. Brake valves are intended to last the life of the car and are too expensive to scrap and replace with an alternate electronic system. Moreover, there is a particularly strong need for the brake components to function together as a system. For example, the brake pipe serves a dual role as a signal transmission line and as a source of pneumatic energy. Contemporary ABDW valves bleed some air from the brake pipe to enhance the propagation of brake application signals and restore air to the brake pipe to facilitate a running release. Electronic brakes would not have to do this. However, in a train in which some cars were equipped with pneumatic brakes and others with electronic brakes, failure of the electronic system to adjust brake pipe pressures could significantly affect the dynamic response of the pneumatic brakes.

Two options for converting the present fleet from pneumatic to electronic brake systems are illustrated in Fig. 41. In Option 1, all cars in the fleet would be wired (during a 10 year period) with an electrical train line, as discussed in Sec. 3.1. Then electronically controlled valves would be installed on new cars. These valves would be controlled from the locomotive and would have the capacity for fast application of brakes. However, their electronic control circuitry would be configured initially to simulate the response of ABDW valves, in order to avoid generating high intercar forces as discussed above. After conversion has taken place, their electronic control module would be converted to control brakes in a fast-acting mode.

In Option 2, cars would also be wired during a 10 year period. Simultaneously, hybrid valves would be installed on new cars. These valves would use transducers to sense brake pipe pressures and electronics to develop valve control signals. After 10 years, all cars would be wired and it would no longer be necessary to install hybrid valves. Instead, new cars would be equipped with electronic valves that are controlled from a remote location such as the locomotive or the caboose. As with Option 1, the hybrid and remote controlled valves would initially replicate the dynamic response of ABDW valves and later be converted to fast-acting configurations.

Each of these options has inherent advantages and disadvantages with respect to the other. Option 1 bypasses the need for an intermediate hybrid system and uses only remote controlled brake valves. However, this option requires the longest time period - 40 years - to achieve conversion. By developing and instrumenting a hybrid system, the conversion period may be reduced by 10 years as illustrated by Option 2. Whether or not this is an economically justifiable course of action depends on the cost of the various systems. These costs can only be estimated once a system is postulated. Accordingly, we start with a hybrid system and investigate its properties and costs.

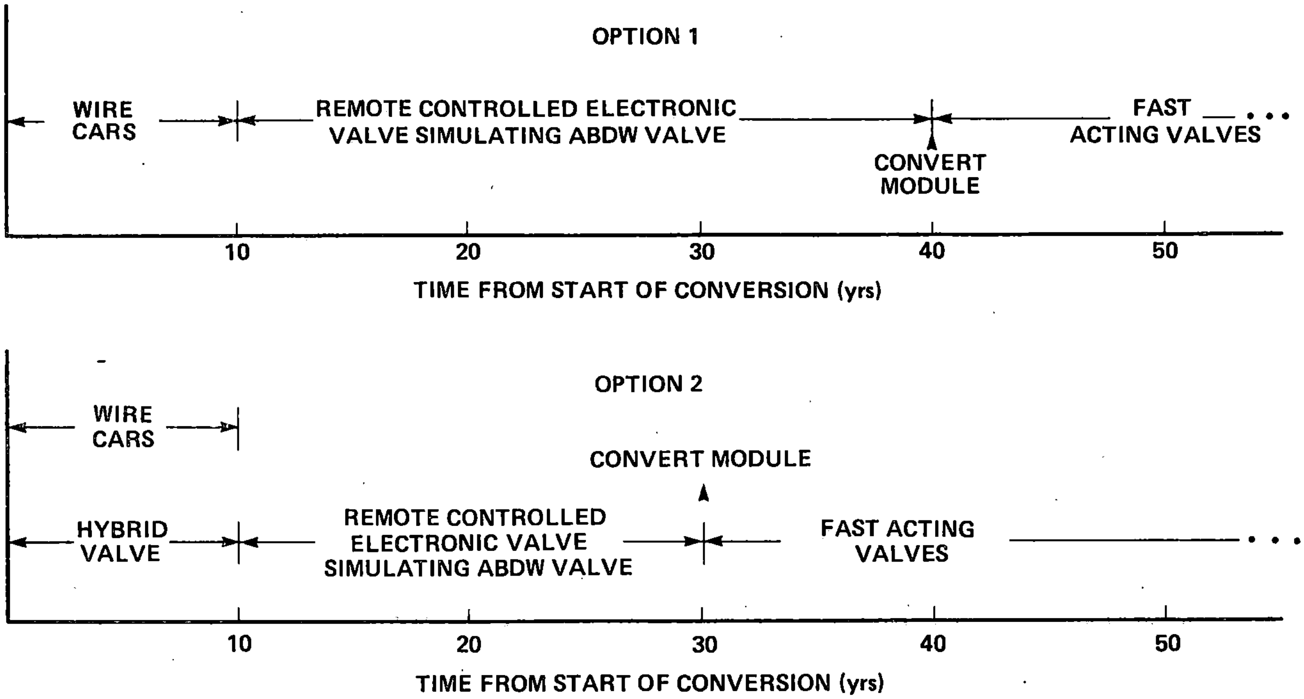

FIGURE 41. TWO OPTIONS FOR CONVERSION OF THE U.S. RAILROAD FLEET FROM PNEUMATIC TO ELECTRICALLY CONTROLLED BRAKES

Systems description

As shown in Fig. 42 , the electronic braking system under consideration is a virtual replacement for the present ABDW control valve. It is designed to perform functionally as a mechanical control valve, using electropneumatic control rather than purely mechanical control. The electronic valve applies pneumatic brakes in response to pressure reductions in the brake pipe, vents air locally upon receipt of an emergency brake pipe reduction, etc.

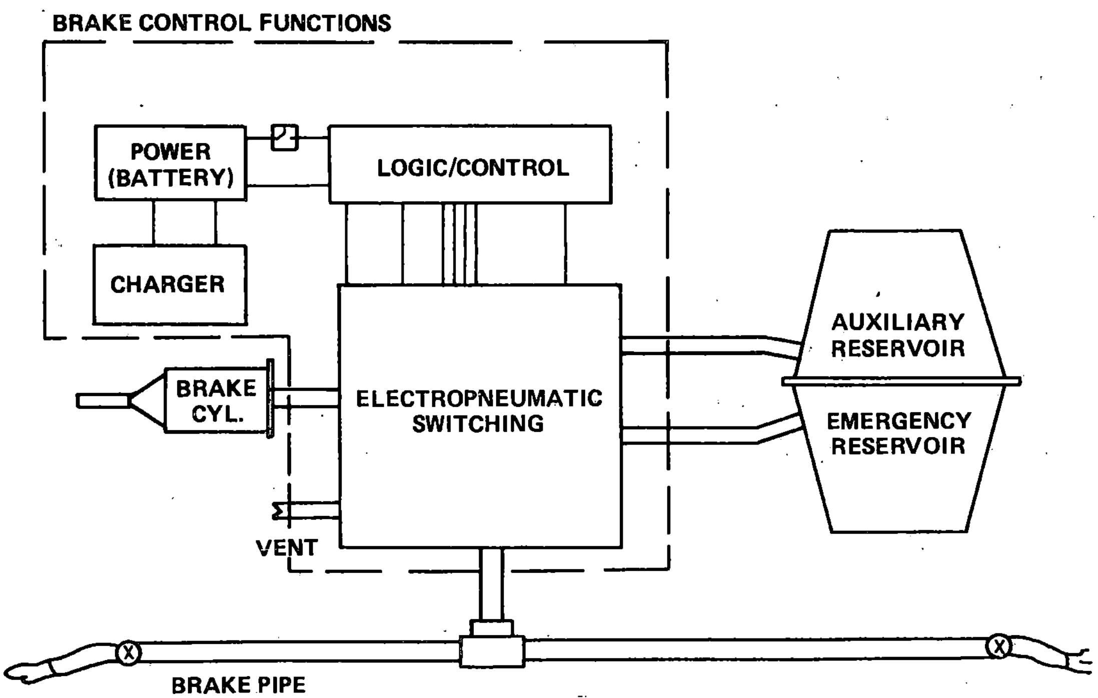

FIGURE 42. OVERVIEW OF A HYBRID ELECTROPNEUMATIC BRAKING SYSTEM

We will treat the electronic brake control valve as a five-port device, with interconnections to the (1) brake pipe, (2) brake cylinder, (3) auxiliary reservoir, (4) emergency reservoir, and (5) outside air. We will also treat the control valve as consisting of three functional parts:

1. Logic, or electronic control
2. Pressure sensing
3. Electropneumatic control valves.

Item 1, electronic control, activates electropneumatic control valves (Item 3), controlling air flow to apply or release the brakes, in response to measured pressures (Item 2) and a preprogrammed valve response characteristic. Initially this valve response characteristic will be chosen to mime the ABDW valve. Eventually it might be modified in order to improve performance, or respond to different requirements or inputs. For example the system might be modified to respond to intercar forces, or react to wheel skid.

In order to explore the technical and economic viability of this concept, we have designed a first generation device using off-the-shelf, currently available hardware. The purpose is to size components, both physically and economically, as well as uncover problem areas, i.e., to test the feasibility.

Pressure reductions sensed at the brake pipe port will be the basis for internal valve operations directing auxiliary or emergency air into the brake cylinder. Flow to the brake cylinder will be stopped when an appropriate pressure is achieved. Similarly, air will be vented from the brake cylinder upon receipt of an appropriate rise in brake pipe pressure. Air could also be valved into the brake pipe in this instance to speed the propagation of a release signal.

The equipment needed to accomplish these functions is shown in Fig. 43. There must be at least two pressure sensors, one on the brake pipe port, and one on the brake cylinder port, and five electropneumatic control valves. We have also provided for two check valves, five chokes to control flow, and a pressure activated power switch.

Valve S is used to control air flow from the auxiliary reservoir to the brake cylinder, during normal brake application. Switches S2 and S4 are used for emergency application, i.e., switch S4 vents emergency reservoir air into the brake cylinder if an emergency state exists, while S2 vents the brake pipe to the air, locally assisting the pressure drop. Check valves V1 and V2 allow reservoir pressurization without reverse flow into the brake pipe. Choke pairs C1-C3 and C2-C4 limit flow from the brake pipe and reservoir in order to restrict the rate of brake cylinder pressure buildup, as well as isolate

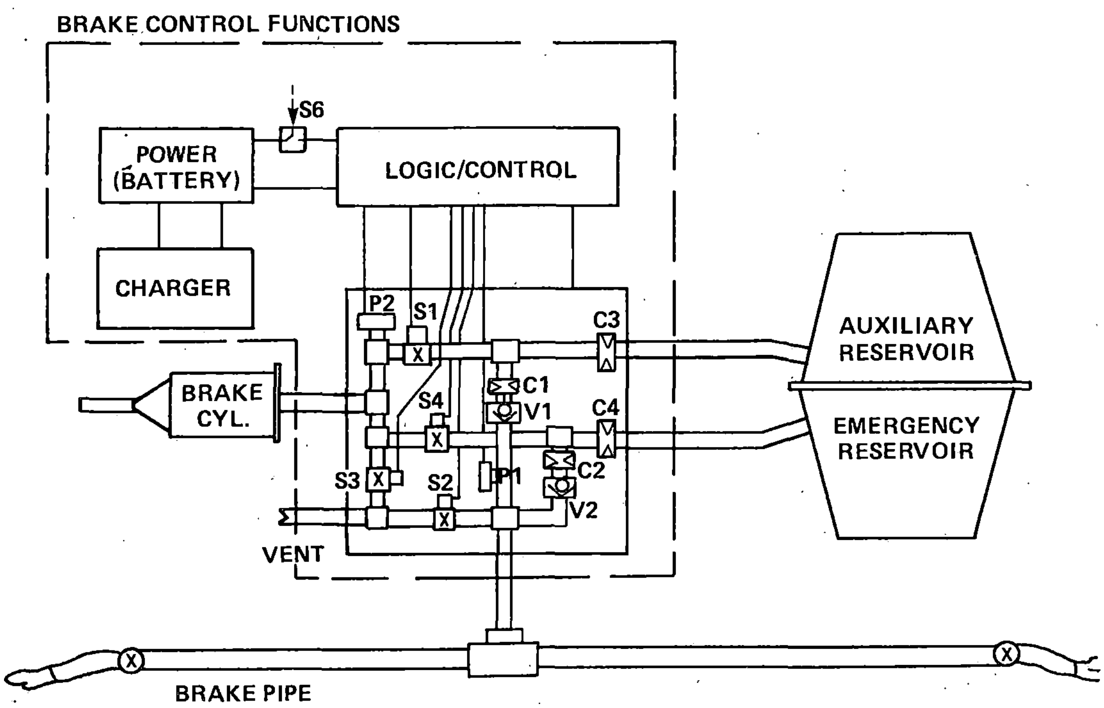

FIGURE 43. HYBRID ELECTROPNEUMATIC BRAKING SYSTEM SHOWING DETAILS OF THE ELECTROPNEUMATIC SWITCHING SUBSYSTEM

the brake pipe pressure sensor from brake cylinder pressures.

These choke pairs can also be chosen so as to vent internally an appropriate volume of brake pipe air into the brake cylinder, thereby helping sustain a brake pipe pressure reduction on a long train.

Switch S6 is a pressure activated, normally open switch which activates the system as soon as brake pipe pressure achieves a few psi, thereby minimizing unnecessary battery drain.

The electronic control receives power from the battery, activates the control switches as appropriate, and receives information from pressure switches P1 and P2.

Each of the system components will be discussed below in more detail.

Electronic Control

Electronic control is the least defined of the functional blocks. It was not judged necessary at this point to design the electronics (perhaps a substantial task) but rather to characterize the complexity of such a circuit, so as to estimate the cost, reliability, and power requirements.

Similarly, rather than diagnose, piece-by-piece, the inner workings of an ABDW valve, and construct an electrical equivalent, we have chosen to treat the control valve as a multi-port device with input-output relationships which can be empirically characterized and electropneumatically reproduced, or mimed.

Characterization of the ABDW Control Valve Response

The first question we address is that of linearity. In order to determine empirically whether the input-output relationships of a mechanical brake control valve are at least piecewise linear, we construct a test of the linearity of the relationship between brake pipe pressure reduction and brake cylinder pressure. Our test consists of comparing the input-output relationships for valves which have different input waveforms. If the brake control valve is linear, the output should be related to the input waveform. The convolution integral is

$$y(t) = \int_0^t h(a)\, x\,(t-a)\, da \qquad (5)$$

where x(t), y(t), and h(t) are the input, output, and impulse response functions, respectively.

Using data provided by Westinghouse Air Brake Co., recorded during their Test 80 of February 7, 1979 of an ABDW valve, we numerically deconvolve the output (brake cylinder pressure) with the input (brake pipe reduction) using waveforms recorded at car number 1. The linearity test then consists of convolving this calculated valve impulse response with a different input waveform, namely the brake pipe pressure at car 50, and then comparing the computed output (brake cylinder pressure) with the actual, experimentally recorded waveform. If both values are the same, and each has a linear input-output relationship, then the computed waveform should correspond closely to the experimentally observed, actual waveform.

The brake system response is shown in Fig. 44. It can be seen that the waveforms at car 1 and car 50 are substantially different due to brake pipe filtering. Our first step is to numerically deconvolve the waveforms at car 1 to obtain an impulse response function for the control valve at car 1. This control valve impulse response function, h(t), is then convolved with the brake pipe reduction at car 50 to derive a calculated brake cylinder pressure at car 50.

Our results are shown in Fig. 45 which is a superposition of the brake cylinder pressure at car 50 and our computed waveform. Since the two waveforms agree closely we conclude that, at least in this instance, an assumption of a piecewise linear system is valid.

Figure 46 shows the integral of the brake valve impulse response, which is the so-called step response. An interpretation of Fig. 46 is that an instantaneous brake pipe reduction of "R" psi results in an initially rapid rise in brake cylinder pressure, a slight overshoot, and gradual asymptotic rise to a brake cylinder pressure of 2.35 R psi.

Complexity of Circuitry: Control Valve Response Function

Our next goal is to estimate the complexity of the circuitry necessary to generate an impulse response function of this sort, and perform the necessary convolution. There are, of course, several approaches, the most general of which is to store sample values of the impulse response function in a read only

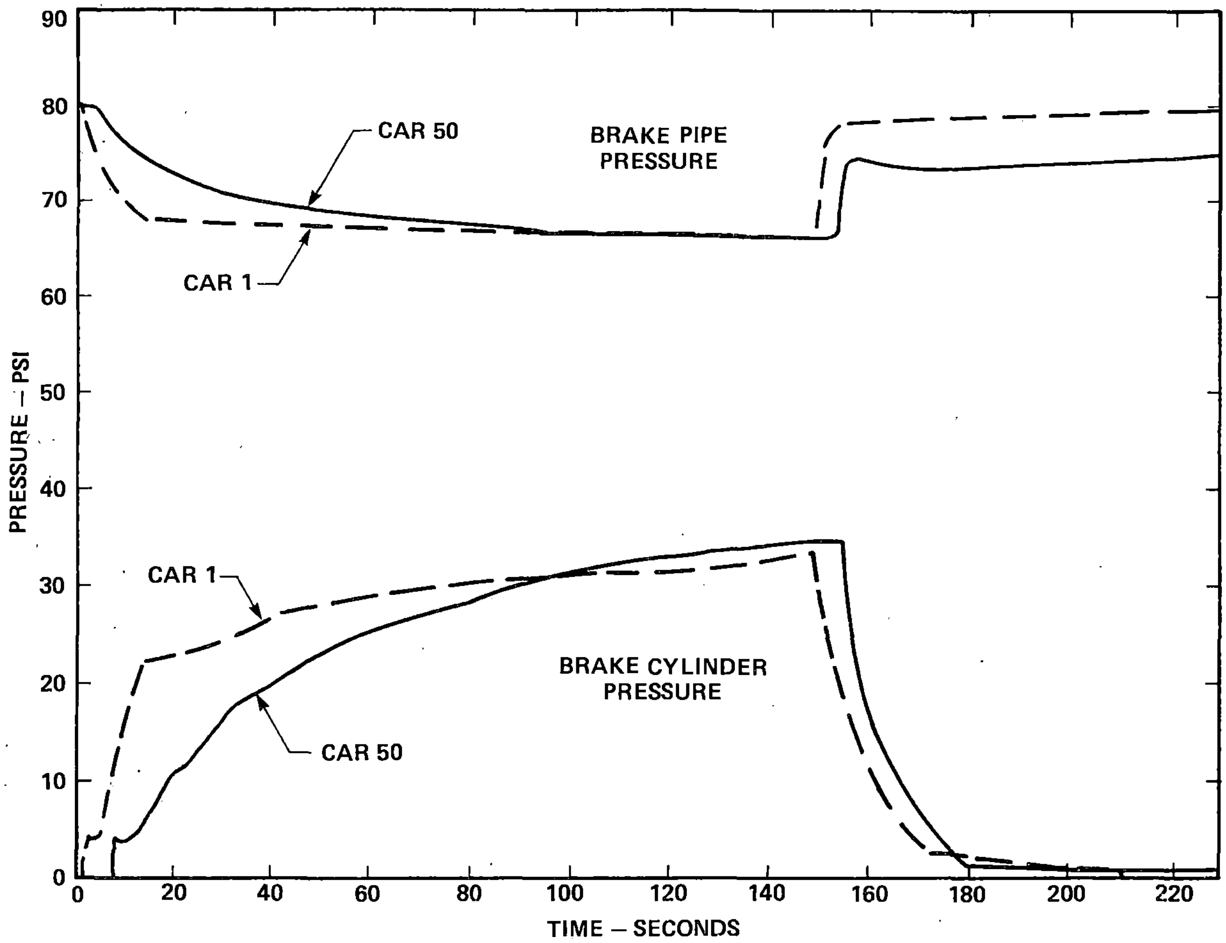

FIGURE 44. BRAKE PIPE AND CYLINDER RESPONSE TO A 15 PSI BRAKE PIPE REDUCTION [18]

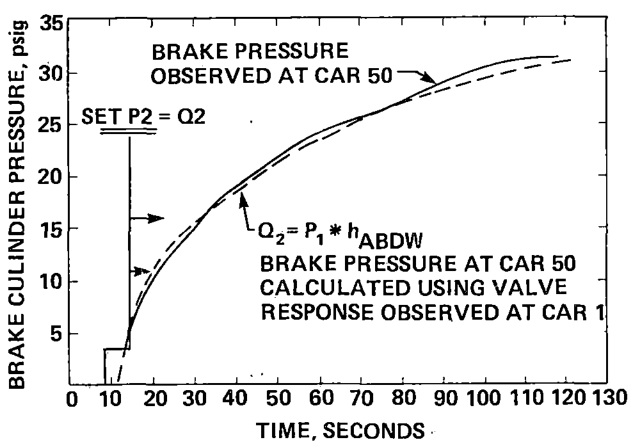

FIGURE 45. COMPARISON OF CALCULATED AND OBSERVED BRAKE CYLINDER PRESSURES AT CAR 50

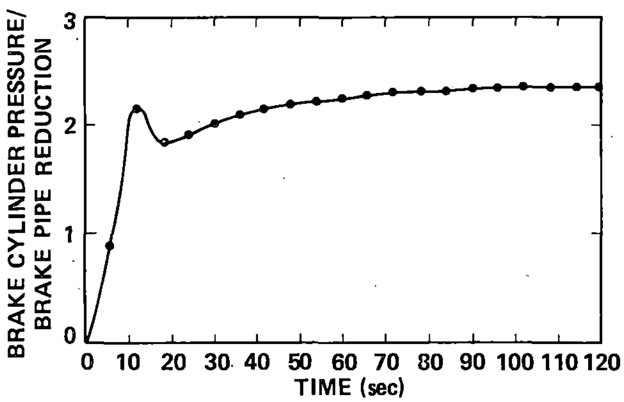

FIGURE 46. STEP RESPONSE OF ABDW VALVE

memory, and numerically compute the convolution waveform using a microprocessor to perform the calculation, just as the hand computed waveforms of Fig. 45 were generated. Alternatively, we might use a circuit which has an impulse response sufficiently similar to the control valve impulse response to be an adequate representation. If achievable, this would probably be the simplest and least costly alternative.

In order to test this possibility, we compared the Fourier transform of our computed impulse response function to the bandpass characteristics of readily fabricated, physically realizable analog circuits. Figure 47 is a comparison in the frequency domain of the amplitudes of the calculated impulse response function and a second order Butterworth low pass filter. The correspondence is sufficiently close to lead us to conclude that there is a high likelihood that simple electronic analog circuits can be constructed which will be adequately close piecewise representations of the response of a mechanical brake control valve. If this should prove not to be the case, more sophisticated filter shaping would have to be used, perhaps including digital filtering, or direct time domain convolution.

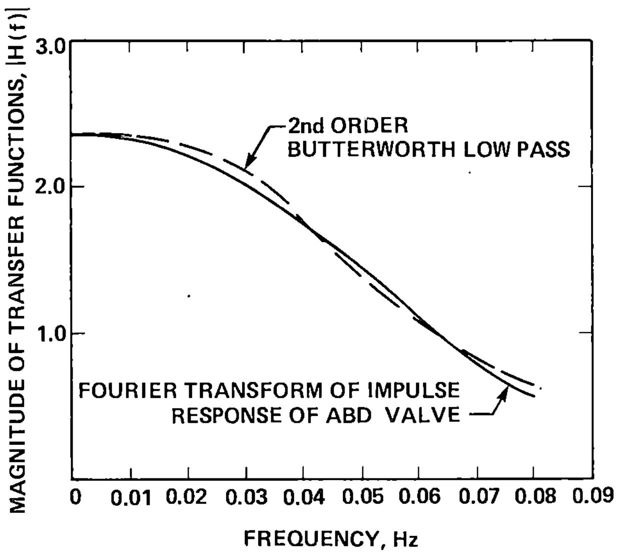

FIGURE 47. COMPARISON OF THE FOURIER TRANSFORMS OF THE COMPUTED VALVE RESPONSE AND A SECOND ORDER LOW PASS FILTER

Hybrid Control Valve Logic

In the preceding section we discussed techniques for generating a "desired brake cylinder pressure" waveform which could serve as a reference for the regulation of flow. However, the desired response is more complex. For example, there is a need to charge the brake cylinder quickly and "set" the shoes with perhaps a 5 or 6 psi cylinder pressure. Mechanical control valves then wait several seconds before proceeding to apply a regulated pressure in response to brake pipe pressure reductions.

Similarly, if the brake pipe pressure reduction is large and rapid, the control mechanism must recognize this as an emergency application and apply emergency reservoir pressure to the cylinders as well as vent the brake pipe to assist in rapid transmission of the emergency signal.

There must also be provision to recognize a brake pipe pressure rise as an instruction to release brakes, perhaps to a partial release if a retainer is set or alternatively, to a complete release. Air may also be vented into the brake pipe in this instance to assist propagation of the release signal.

Each of the above steps is summarized in Table 8 with a set of criteria. This tabulation may not be a completely accurate representation of an ABDW valve; however, the goal at this point is not to model the actual operating parameters quantitatively, but rather simply to ensure that we have provided for a sufficiently complex representation, since particular threshold levels and charge rates are not likely to affect substantially the cost or reliability of the device.

In Fig. 48 we have reduced the requirements of Table 8 to a speculative electronic design. Each of the functional rectangular blocks of this circuit can be constructed using an operational amplifier and a few resistors and capacitors. In the case of the monostable oscillators, either digital gates with resistive and capacitive load or analogue circuitry can be used.

Threshold detectors can be thought of as comparators with preset reference biases. If we assume that both brake valve response functions can be adequately represented with a second order filter, then the circuitry of Fig. 48 can be made using a total of about 18 operational amplifiers, 60 resistors, 30 capacitors, 12 VMOS FET switches, and 25 CMOS logic gates.

In order to minimize power drain and battery size, we have assumed a magnetically latched solenoid valve, such as those manufactured by Skinner Corporation. As a result, the electronic output to each solenoid has two ports, which provide a 1 amp surge for about 10 milliseconds to open the valve, and a similar pulse on the other port for closing. A speculative switch/driver design is shown in Fig. 49. Each open/close event requires a total energy of about 0.3 joule which is less than 10^{-5} ampere-hours. The current drain of the above is approximately 5 milliamps while the train is underway (brake pipe pressure up).

TABLE 8. FUNCTION OF ABDW CONTROL VALVE

Function	Logic Test/Criterion	Action
Partial or Full Service Brake Application	If $-\overline{P}_1 > 1$ psi	S_1 open 1 sec (Sets $P_2 = 5$ psi)
Quick Service Venting Controlled by Internal Chokes C1, C3		
Emergency Stop	If $-\overline{P}_1 > 30$ psi & $d\overline{P}_1/dt > x$ psi/sec	Open S_2 30 sec (Quick Action Open S_4 0.1 sec (Sets $P_2 = 5$ psi) Closes S_4 1.0 sec Opens $S_{4\neq}$ so that $P_2 = P_1 *h_{EMER}^{\neq}$
Release	If $\overline{P}_1 > .8$ psi "Retainer" set	S_3 open until $P_2 = 10$ psi
	"Retainer" not set	S_3 open 30 sec S_5 open 5 sec (represures brake pipe)

*≠Indicates convolution. P1 and P2 filtered to enhance stability and minimize false alarms.

<u>Pressure Sensing</u>

Our requirements for a pressure sensor include frequency response from dc to perhaps a few Hertz, modest power drain, a range from zero to 150 psi, a package suitable for hostile environment, extreme reliability and modest cost. There are several pressure sensors currently available which meet these requirements. Some specific examples include the Kavlico model P609, Data Instruments Model AB-100, and Viatran Model 108. The first of these, the Kavliko Model P609, is a capacitance gauge, while the others are strain gauge bridge sensors. Each manufacturer's design offers its own design advantages and weaknesses. It is sufficient at this point to note that there are several appropriate devices from which to choose.

The Kavlico sensor has been extensively tested and is currently being introduced into large volume automotive service [27], as well as the Canadian metro. Similarly, the data instruments AB series is in service in high reliability application such as heart assist machines and injection molding control. The Kavlico pressure sensor is compatible with circuit configurations like those shown previously. Its current drain is 10 milliamps. It is operable from 12 volt dc supply, and generates an output of 2 to 6 volts dc, proportional to pressure.

<u>Electropneumatic Control Valves</u>

There are five electropneumatic valves necessary for the functions described. Requirements for these valves are that they be extremely reliable, capable of handling the necessary rates of flow, and preferably operable with a modest electrical power drain.

<u>Quick Service Venting</u>

Control valve requirements are also affected by the sizing of supplementary chokes and check valves. We noted in Table 8 that chokes C1-C3 can be chosen to control quick service venting.

Referring to Fig. 50, assume that volumes V1, V2, and V3 represent the brake pipe, brake cylinder, and auxiliary reservoir volumes respectively. Let S1 represent an electropneumatic valve with a flow capacity large compared to either C1 or C3. Our goal is to select C1 and C3 so that (1) the brake cylinder (V2) is limited to an appropriate maximum rate of brake application, and (2) the quantity of air flowing out of the brake pipe into the brake cylinder (V2) is equal to that which was originally withdrawn by the previous car in order to signal a brake application.

The withdrawal of air from the brake pipe speeds the propagation of a pressure reduction signal on a long train, and is generally referred to as quick service venting.

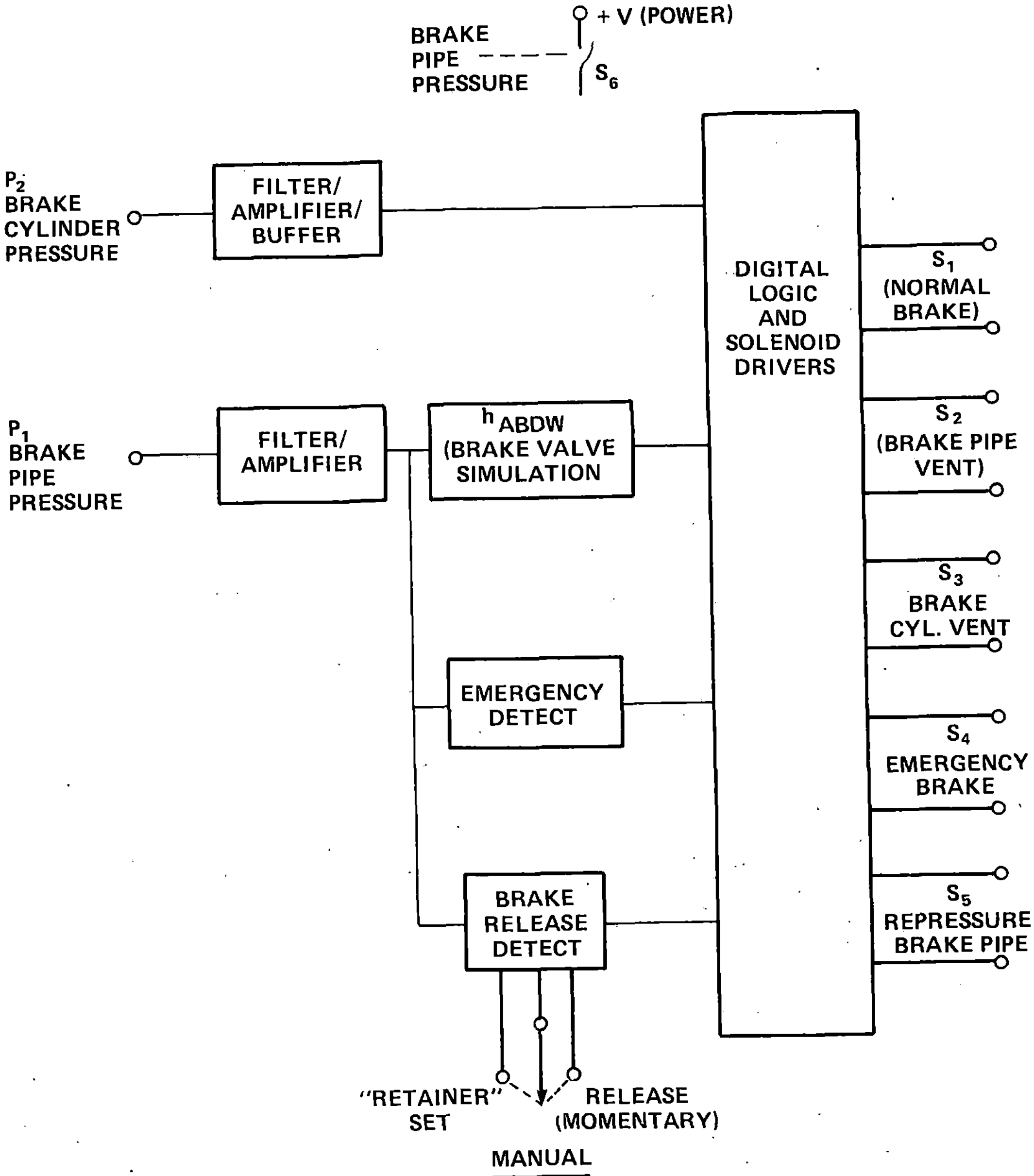

FIGURE 48. SPECULATIVE DESIGN OF BRAKING CONTROL LOGIC

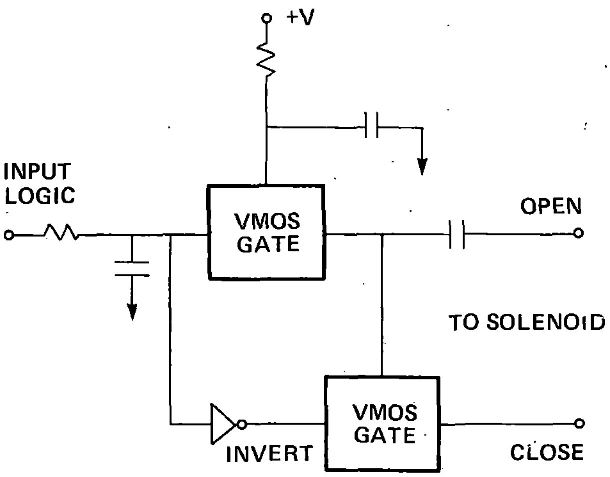

FIGURE 49. A SWITCH/DRIVER DESIGN FOR USE WITH A MAGNETICALLY LATCHED SOLENOID VALVE.

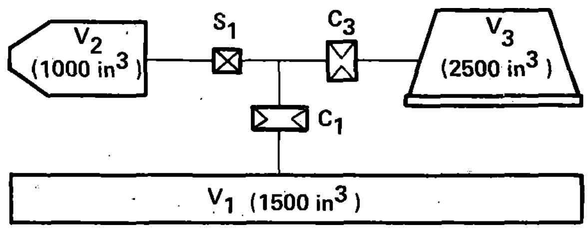

FIGURE 50. SIMPLIFIED MODEL FOR CHOKE SIZING TO ACHIEVE QUICK SERVICE ACTION

If electropneumatic switch S1 has a large capacity compared to the sum of chokes C1 and C3, then the total flow rate to V2 is given by

$$q_1 + q_3 = q_{total} \qquad (6)$$

where q_1 and q_3 represent flow in C1 and C3, respectively. The brake cylinder rate of pressure rise is given by

$$\frac{dp_2}{dt} = \frac{(q_1 + q_3)}{V_2} \qquad (7)$$

Similarly, in the absence of flow from the ends of the brake pipe its rate of pressure drop would be

$$\frac{-dp_1}{dt} = \frac{q_1}{V_1} \qquad (8)$$

We have previously established that a normal brake application results in a brake cylinder pressure rise which is approximately 2.4 times the brake pipe pressure reduction. It follows that in order to vent internally all of this brake pipe air (which of course now comes from the adjacent cars) we let

$$dp_2/dt = -2.4\, dp_1/dt$$

or

$$q_1/q_3 = V_1/(2.4V_2 - V_1) \qquad (9)$$

if $V_1 = 1500 \text{ in.}^3$ and $V_2 = 1000 \text{ in.}^3$, then $q_1/q_3 = 5/3$. Similarly, if we limit our service application time rate of rise of brake cylinder pressure to 5 psi/sec, then from Eq. 7 , $q_1 + q_3 = 12$ SCFM, or q_1 7.5 SCFM, $q_2 = 4.5$ SCFM which corresponds, at 80 psig upstream pressure, to flow coefficients of 0.14 and 0.08, respectively.

We can conclude that passive internal chokes can be used to achieve a quick service venting action. Inclusion of these chokes should not be deleterious to reservoir charging, since 4.5 SCFM corresponds to an initial rate of charge of more than 4.5 psig/min.

Chokes C2 and C4 can be chosen based on a different rationale (see Fig. 43). In the case of emergency brake application electropneumatic switch S2 will be activated, locally venting the brake pipe. Choke C4 can be used to limit the rate of pressure buildup in the brake cylinder during emergency application, and C2 used to control the rate of pressurization. For example, if we limit emergency brake pressurization to 50 psi/sec we need 120 SCFM, which at 80 psi upstream corresponds to a flow coefficient of about 2.

<u>Electropneumatic Valve Criteria</u>

It is clear from the preceding that in addition to other requirements, control valves S1, S2, and S4 need to have a flow coefficient, C_v of 2.0 or more. In the event this system were to be produced in the necessary quantities, a specific custom design would be warranted; however, as in the case of the electronics, we can estimate the cost and complexity by identifying presently available commercial products which satisfy our technical requirements. There are many suppliers of appropriate electropneumatic solenoid valves. Examples include Skinner, Asco, Atkomatic and others. For purposes of illustration, we will cite specific Skinner valves which represent technical capability and will later prove useful for cost estimating.

The Skinner model LC2DA3150 has an operating range of 5 psi to 150 psi and a flow coefficient of 2.4. It is a pilot piston design operable from 12v dc power. It satisfies our requirements for valves S1, S2, and S4.

Valves S3 and S5, which require a much more modest flow rate, might be the equivalent of the model V52DA1100 which operates from 0 to 100 psi with a flow coefficient of 0.28.

These and similar valves from other suppliers are typically capable of more than 10^6 operations, particularly if the air is a least periodically misted with oil, or an alternative lubricant or agent chosen to preserve the seals of both the solenoids and the glad hand couplers.

The power requirements outlined above are 300 milliwatts, or 25 milliamps at 12 volts D.C., <u>whenever the brake system is pressurized</u>. During braking each solenoid activation requires an additional 10^{-5} ampere-hours of energy.

Unit costs

Costs for an alternative braking control valve should include an estimate of initial cost (purchase), implementation costs for both new and retrofit installations, and maintenance costs. In the following each of these will be addressed.

Estimated Initial Cost

Because the electropneumatic brake is a significant design departure, its cost estimation is of necessity somewhat speculative. However, it is an electromechanical component, similar in complexity and manufacture to equipment presently in use in the automotive, electronics, and industrial control industries. The scope of our project does not permit a detailed manufacturing cost estimation, which involves estimate of direct labor hours for each step of the fabrication, assembly, inspection, and test, as well as allowances for overhead, production engineering, earnings, etc. [28]. However, there are concept cost estimating guidelines which are helpful and first-order accurate. The guidelines to be used for estimation of the cost of a hybrid electropneumatic brake are:

1. The cost of standard components ("material") can be quantity discounted at a rate of 0.9 to 0.95 per quantity doubled, depending on the nature of the product [29].
2. Line stock items (miscellaneous) represent an additional 3% of material [29].
3. Engineering change and design growth represent an additional 20% [29].
4. Manufactured price (cost to user) of the system will be 3.3 times the above total component "materials" costs [30].

Our approach will be to cost the components, assuming procurement in lots of 10,000 systems, based on our preliminary design and published prices discounted as above, with a 23% increment for engineering change during detailed design, then apply a 3.3 multiplier to account for labor, overhead, and profit, which assumes a product of "average" labor intensity, and may be overly conservative given the very substantial quantities involved.

A summary of these estimates is given in Table 9.

Implementation Costs

In the above analysis we assumed that the hybrid valve is designed and packaged as a virtual replacement for the ABDW valve. As a result the incremental implementation costs for labor, materials, facilities and time are nil for new installations.

TABLE 9. HYBRID ELECTROPNEUMATIC BRAKING ESTIMATED COSTS

Component	Estimated Materials Costs
Electronic Control[1]	$ 38
Pressure Sensory[2]	42
Electropneumatic Switching & Plumbing[3]	145
Sub Total	$225
Misc. (3%) and Eng. Growth (20%)	53
Total Materials	$278
Manuf. Price Factor	x 3.3
Total Estimated Cost	$917

[1] Estimate from parts cost
[2] Kavlico 100 GPA or equivalent, .945/decade
[3] (2) V52DA1100 +
(3) 6C20A3150 + Latch + discounted at .94/decade.

Estimated Maintenance Costs

Maintenance costs depend upon the frequency of service and repair, which in turn depend upon the acceptable failure level. We have adopted a criterion that at any instant at least 90% of the brakes must be expected to be functional.

Electronics with an average component failure rate of $1/10^8$ hours, and 100 gates and components will have a 90% reliability at 12 years. Substantially better performance may be achieved with integrated chip fabrication. However, a 12 year replacement should be taken as an initial estimate. Cost of the electronics is estimated above as $38 (1.23)(3.3) = $154.00.

Pneumatic valves are rated to more than 10^6 operations, which at 1,000 per average 26 day trip is substantially in excess of our estimate for the valve mechanism of 12 years between major overhaul.

Pressure sensors, dual electronic converters, and rudimentary dual constant current generators are very reliable equipment which should also survive a 12 year maintenance overhaul and repair. We estimate that this might cost $100 to $200 every 12 years.

In summary, the 12 year maintenance cost is estimated at $254 to $354 or about $20 to $30 per year.

Benefits

The potential benefits of electronic braking are performance and cost. Ultimately, when the fleet is converted, brakes will be applied simultaneously throughout a train with faster stopping, graduated release, and lower intercar forces. Even during a lengthy conversion period after all cars are wired but before all cars are equipped with electronic brakes, intercar forces will diminish. In this case, braking is initiated at each electronically equipped car in the consist, resulting in considerably reduced drawbar forces.

The estimated price of the electronic valve is $917, which is $358 less than the $1275 per ABDW valve. The estimated maintenance costs of both valves are sufficiently close, and within margins of error, to balance. (The electronic valve maintenance costs is estimated at $254-$354 every 12 years, while the ABDW scheduled COT&S maintenance costs is $284 plus unscheduled maintenance cost of about $54 over the same period).

Cost/Benefit

The estimated to allowable cost ratio is 917/1275 = 0.72 which is favorable.

3.3.2 Electrical load sensors

As discussed in Sec. 2.3.6, load sensing systems are very useful for compensating for the wide range of loads found in freight cars. While mechanical load compensating systems may be appropriate for cars with very high gross to tare weight ratios, they appear to be too expensive for general use. Here we will determine whether electrical systems may be sufficiently inexpensive to pay off.

Description of System

An electrical load proportional system may be used efficiently with an electronic brake system. It could involve a single potentiometer placed in a spring nest at the end of a car to detect the car's static deflection. A better system would involve the use of two potentiometers, one at each corner of the car to account for nonuniform load distributions. Mechanical filtering would be desirable to prevent vibrations from wearing the potentiometer. Also, electrical low pass filtering might be needed to detect the d.c. portion of the signal and prevent car oscillations from constantly changing the indicated deflection.

Implementation Scenario

We assume that electrical load sensors are installed on new cars, and that these cars are equipped with electronic brakes. Since each car equipped with a load-sensing system will immediately enhance braking performance, there is an immediate payback. The investment and benefit profiles are illustrated in Fig. 51.

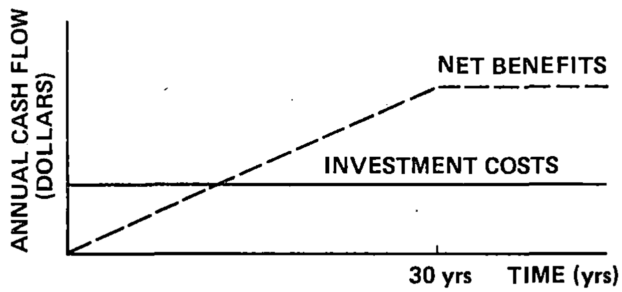

FIGURE 51. CASH FLOW FOR ELECTRICAL LOAD SENSORS

Unit Cost

A system would require the following major components.

Potentiometers (2@ $25 each)	$ 50
Wire (100 ft $.8/ft)	8
Misc. fasteners	12
Total materials	$ 70
Installation cost	50
Total	$120

Benefits

The gross benefits are identical to those presented in Sec. 2.3.6. Their upper bound value is $61 M. The annual maintenance cost is estimated at 4% of \$120 times 1,444,000 cars or \$7M. The maximum net annual benefit is therefore \$54M.

Cost/Benefit

A net annual benefit of \$54 M and a cash flow as illustrated in Fig. 1 justifies a per car investment of \$73. Thus the estimated/allowable cost ratio is 1.6 which is not favorable.

3.3.3 Electro-pneumatic brakes

Electro-pneumatic brakes used on passenger trains exhibit superior performance when compared to conventional freight train air brake systems, but at a substantially higher cost. Obviously, cost is of much less concern for passenger trains, in which a thousand lives may depend on the distance in which a high speed train may be stopped. The question addressed here is whether the benefits to freight trains of electro-pneumatic brakes would justify their additional expense.

Description of System

At its essential level, an electro-pneumatic brake system employs a locomotive brake valve and master controller, an electrical line running throughout the train, a straight air pipe and a brake pipe, both of which also run through the train (see Fig.52). The brake pipe supplies compressed air to each car; the electrical line provides brake application and release signals, and the straight air pipe feeds brake cylinder pressure back to the master control for comparison with the pressure commanded by the brake valve.

When a brake application is desired, the brake valve handle is moved to the application zone and air is introduced through the control pipe to the master controller. Within this controller, a diaphragm is deflected which closes electrical switches, activating valves in each car simultaneously. These valves allow air to flow to the brake cylinders in each car and to the straight air pipe. When the pressure in the cylinder and straight-air pipe build up to the level set in the master controller, the diaphragm moves towards its neutral position and the valve supplying brake cylinders closes, holding the cylinder pressure at the desired level. A graduated release may be made by reducing the pressure applied to the master controller.

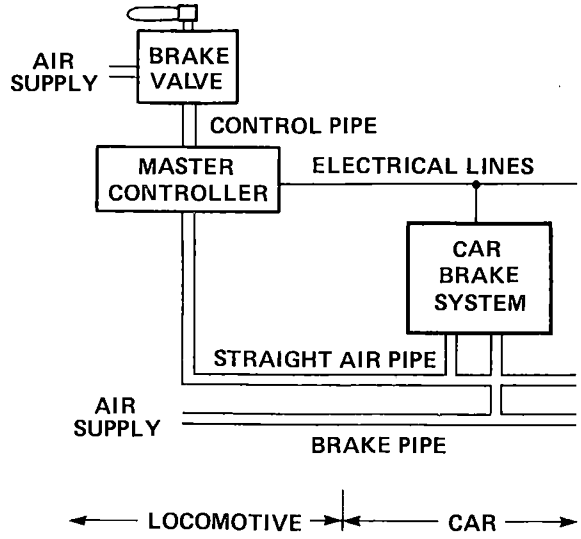

FIGURE 52. SCHEMATIC DIAGRAM OF ELECTRO-PNEUMATIC BRAKE SYSTEMS USED ON PASSENGER TRAINS

Implementation Scenario

The two-pipe electro-pneumatic system described above is fundamentally incompatible with single air line equipment used in freight brakes. Accordingly, significant development work would have to be performed to make this system compatible with freight brake systems. For purposes of completing the cost/benefit analysis we make the tentative assumption that this can be accomplished at no additional cost. We also assume that conversion would then take place in a 10 year period.

Unit Cost

The per-car cost of a CS-1 passenger brake system is about \$7500. Subtracting \$1275 for an ABDW valve results in a net increase of \$6225. This value would undoubtedly be reduced for large quantity production, but the reduced value is difficult to estimate without conducting a detailed cost study.

Benefits

The maximum upper bound benefits would be:

Collision avoidance:	$30.5M
Reduction of in-train forces train delay	6.0
Derailment and broken train collision	12.0
Maintenance	12.5
Gross benefits	$61.0 M
Maintenance *	360
Net benefits	(229)M

Cost/Benefit

As indicated above, the estimated maximum possible benefits do not even offset estimated incremental maintenance costs. This would still be the case even if CS-1 system costs were reduced by a factor of two and maintenance costs were only 2% of equipment costs.

* Four percent of $6225 applied to 1,444,000cars

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