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U.S. DEPARTMENT OF COMMERCE/National Bureau of Standards

Failure Prevention in Ground Transportation Systems

MFPG
31st Meeting

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MFPG
Failure Prevention in Ground Transportation Systems

Proceedings of the 31st Meeting of the Mechanical Failures Prevention Group, held at the National Bureau of Standards, Gaithersburg, MD, April 22-24, 1980

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FOREWORD

The 31st meeting of the Mechanical Failures Prevention Group was held April 22-24, 1980, at the National Bureau of Standards in Gaithersburg, Maryland. The program was organized by a special ad-hoc committee to bring together experts concerned with failures in ground transportation systems. H. R. Hegner of ManTech International Corporation served as Program Chairman. He was assisted by E. Passaglia, A. W. Ruff, C. G. Interrante and J. H. Smith of the National Bureau of Standards. Special appreciation is extended to K. L. Pierson, D. Dancer and J. Krugler of the Department of Transportation for providing speakers and contacts within the DOT.

Appreciation is extended to the following members of the NBS Fracture and Deformation Division: T. Robert Shives and William A. Willard for their editing, organization and preparation of the proceedings, and Joel C. Sauter for photographic work. Most of the papers in the proceedings are presented as submitted by the authors as camera ready copy. Some minor editorial changes were required.

Gratitude is expressed to the following members of the NBS Metallurgy Division: Marian L. Slusser for handling financial matters and Kimberly A. Smith for typing.

Special thanks are extended to Jo Ann Lorden of the NBS Public Information Division for the meeting, hotel and banquet arrangements.

HARRY C. BURNETT
Executive Secretary, MFPG

Center for Materials Science
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ABSTRACT

These proceedings consist of 18 submitted entries (16 papers and 2 abstracts) from the 31st meeting of the Mechanical Failures Prevention Group which was held at the National Bureau of Standards, Gaithersburg, Maryland, April 22-24, 1980. The theme of the symposium was failure prevention in ground transportation systems. Areas of interest included rail vehicles and structures, highway and road bridges, pipeline transportation systems, and motor carriers.

Key words: bridges; diagnostic systems; failure; failure detection systems; fracture; fracture control; ground transportation; motor carriers; pipelines; rail structures; rail vehicles; reliability; transportation systems.

UNITS AND SYMBOLS

Customary United States units and symbols appear in some of the papers in these Proceedings. The participants in the 31st meeting of the Mechanical Failures Prevention Group have used the established units and symbols commonly employed in their professional fields. However, as an aid to the reader in increasing familiarity with the usage of the metric system of units (SI), the following references are given:

NBS Special Publication, SP330, 1981 Edition, "The International System of Units."
ISO International Standard 1000 (1973 Edition), "SI Units and Recommendations for Use of Their Multiples."
IEEE Standard Metric Practice (Institute of Electrical and Electronics Engineers, Inc.), Standard 268-1979.

SESSION I

RAIL VEHICLES AND STRUCTURES

Chairman: D. Dancer, Federal Railroad Administration

COMPONENT RELIABILITY OF RAILROAD FREIGHT CAR TRUCKS

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Abstract: The modern freight car truck is a relatively simple structure which provides for weight transfer between the car and track, maintains the proper positioning of the wheels with respect to the rail, contains the suspension elements for isolating the effects of wheel/rail forces and acts as a support for the brake. The number of freight car truck components which fail and cause accidents is low with respect to the number of cars in the railroad fleet. Continued efforts for their improvement are warranted because of the potential seriousness of accidents. Most freight car truck components function as safe-life rather than fail-safe structures requiring a conservative approach to their design. The reliability problem is complicated by the practical difficulties of making adequate field inspections. Current efforts to improve reliability include gaining a better understanding of complex wheel failure phenomena, improving design and performance specifications, developing new systems for wayside inspection and developing new truck designs for the reduction of wheel/rail forces.

Key words: Railroad freight car; freight car truck; fatigue; derailments; railroad accidents; reliability; railroad testing.

Introduction: A number of important trends are evident in current railroad operations. First, there is the use of larger freight cars. Most new cars are of nominal 100 ton capacity, which allows a 32,875 lb wheel load. Larger wheel loads are not permitted in normal interchange service. As wheel/rail loads become larger there is more opportunity for developing damaging loads on both the track structure and vehicle. The second important trend is the increased movement of bulk commodities in unit trains. Unit train cars will all have similar dynamic behavior so that motions and forces excited by vehicle/track interaction will tend to be repetitive from one car to the next. These repeated loads will accelerate the development of damage at particular track locations. Third, there is the tendency for some railroads to allow track quality to deteriorate. This increases the dynamic vehicle/track interaction effects, hastens the onset of track deterioration, and increases the risk of derailments.

The increase in the derailment rate raises safety concerns. Derailments not only have the potential of causing serious casualties and inconveniences to the public, but they have an adverse affect on railroad operations as well. Of special concern is the probability that damage to cars containing hazardous materials might lead to a major catastrophe. In recent years there have been a number of accidents where widespread damage and numerous casualties were caused by the release of hazardous materials from tank cars. Although the loss of life has been relatively low from these events, one can conceive of accident scenarios which could lead to major catastrophes.

Insuring the reliability of the freight car truck is one aspect of safer rail transportation. This paper reviews the reliability characteristics of freight car trucks. Specifications used for their design are also reviewed. Actions being considered to improve reliability are mentioned. Finally, some indications are given of the kind of research and development which is likely to lead to further improvements in reliability.

Aspects of Reliability: One must recognize several aspects of the term "reliability". First, there is reliability with respect to accident free operation. Safety is equivalent to reliability in this context. Second, there is reliability with respect to the early removal of components from service because they are defective. A defective part may be functional, but it exhibits characteristics which are indicative of a premature failure. Removal of defective components is of economic concern because the desired life of the part is not obtained. Third, the relationship between reliability and design must be recognized. Reliability can be enhanced by design procedures which are based on an adequate definition of the environment in which the part functions as well as a complete understanding of the response of the component to that environment. Finally, the relationship of reliability to inspection practices must be considered. Since most failures are the result of defects which progress over a period of time, like a fatigue crack, reliability can be enhanced by the increased frequency and thoroughness of inspections. Adequate inspections are also required during the manufacturing process so that the quality of components introduced into service are maintained.

Description of Conventional Freight Car Truck: The principal features of the conventional four-wheel freight car truck are illustrated in Fig. 1. The weight of the car is concentrated by a structural element called the body bolster. This mates with the truck bolster at the center plate in a pivot connection which provides for the swiveling of the truck to accommodate yawing angular displacements between the truck and car. The truck bolster, a steel casting, distributes the load from the center plate to the two spring groups, one at each side of the truck. These are supported by the side frames. Two side bearings, one on either side of the center plate provide an alternate means of load transfer between the car and the truck bolster. These are engaged by a slight rolling motion of the car.

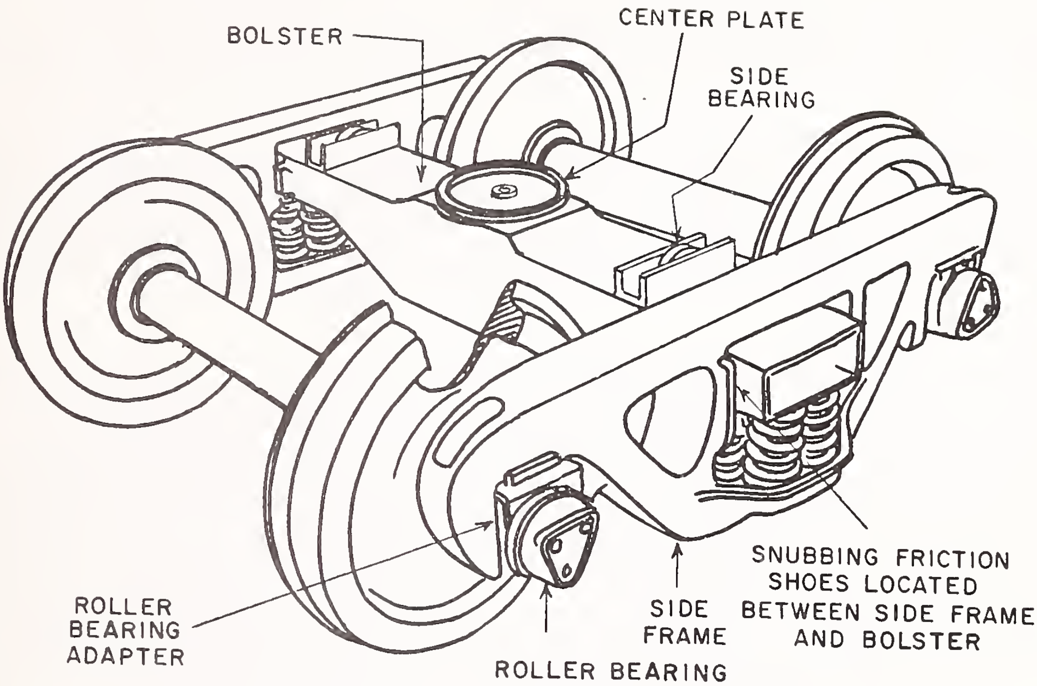

FIGURE 1 CONVENTIONAL FREIGHT CAR TRUCK

Conventional freight car truck designs utilize only one type of suspension element, namely, coil springs between the bolster and both the side frames. The side frames, which are also steel castings, distribute the load from the base of the springs to the two axles of the truck. The load is applied to the axles through bearings contained in housings which are independent of the side frame, but effectively connected to it through the jaws or pedestals at the ends of the side frame. The use of roller bearings has been required on all new cars built since August 1968. Over 60 percent of the freight car fleet is now equipped with roller bearings. The wheels are rigidly mounted to the axles so that a wheel/axle set turns as a unit.

Shock and vibration isolation is accomplished by relative motion between the bolster and the side frames. The primary freedom for relative movement is in the vertical direction; the total possible motion depends on the characteristics of the spring group. Relative motion is also permitted in the lateral direction, although to a lesser extent. The system must be damped to prevent excessive vertical motions, which would result in bottoming out the springs with accompanying hard solid blows to the bolster. Most trucks are damped by a shoe which provides frictional forces between the bolster and the side frame columns. The spring-loaded friction shoes also offer a resistance to skewing forces which tend to distort the rectangular array of the truck side frames and axles.

The freight car truck also supports the brake. The air cylinders which initiate the braking forces may be attached to the truck components or the car, but in either case the linkage which transfers the piston rod forces to the brake shoes is connected to the truck components. In this way the truck structure transmits the braking forces from the wheels to the car.

The overall reliability of the freight car truck is a function of the reliability of the components which go to make up the system. It is also influenced by the nature of the structural design philosophy, whether it is of fail-safe or safe-life design. Fail-safe design relies on the utilization of redundant structural members. The probability of a catastrophic system failure is small because failure of the system requires simultaneous defects in each of the redundant members. There are only a few examples of fail-safe design principles in the freight car truck. One is the truck-bolster/body-bolster connection where the center plate rim and the center pin provide redundant structural elements in the connection between the two bolsters.

Safe-life design requires maintaining the integrity of each structural element. Structural components must be designed for an extremely low probability of failure during intervals between inspections. Conventional freight car truck construction is based primarily on safe-life principles. Failure of one of the major structural elements, such as the side frames, bolster, wheels, or axles inevitably leads to derailment.

Changes in freight car truck design have taken place at a relatively slow pace. This is due in part to the large number of cars that are in the railroad fleet and the need for maximizing the useful life of this equipment. Changes in design must be compatible with existing operations, and a changeover of the entire fleet must be planned to occur only over an extended period of time. The economic factors are important too. The nature of freight car operations, wherein the average car moves only 60 miles per day, requires that it have a low capital value and this severely limits the alternate approaches that can be considered in freight car truck design.

Truck Component Loads: The vertical load is the principal load acting on the truck and is due to the weight of the car and lading. It is modified by transient factors, such as rail irregularities, suspension system oscillations, wheel flat spots, etc. Most of the fluctuations in the vertical load occur at frequencies below 10 Hz. Car rolling motions result in an alternating component of load on each side of the truck at a characteristic frequency of approximately 1 Hz for a loaded car. Car bouncing and pitching motions cause load fluctuations in the 3 to 5 Hz range (see Ref. 1). The line-of-action of the vertical load is normally at the center of the bolster, but this can be shifted because of load transfer to one of the side bearings. Load transfer to the side bearings is most severe on curves or during car "rock and roll" motions. Car rock and roll motions are enhanced by a high center of gravity, a truck center distance approaching rail length, and operation at critical speed on jointed rail.

Lateral truck loads are the result of load transfer from the car body, such as would be caused by the angularity of the draft force on the car, the superelevation of a curved track rail not commensurate with speed, guiding forces for the negotiation of curves, or flange contacts during the hunting motion of the wheel-axle set. Lateral loads are also associated with the internal truck forces which tend to maintain the truck in a rectangular configuration while traversing curved track. Lateral creep and slip forces are also built up at the wheel/rail contact points. These forces are normally directed toward the flange. Occasionally large transient forces can be directed in the opposite direction due to guardrail contacts. Severe lateral loads also can be developed at the wheel/rail interface under the self-excited vibrational condition commonly referred to as truck hunting. This condition is usually associated with a light weight car traveling at high speeds.

Horizontal twist loads between the side frame and bolster result from the tendency of the side frame to rotate into an out of square position with respect to the bolster during the traversal of curved track or due to truck swiveling motions. Longitudinal loads result from braking forces and the inertial forces accompanying acceleration of the truck. The most severe forces occur under car impact conditions when an unloaded car is coupled at high speeds into a standing string of cars. The longitudinal load is applied to the bolster at the center plate rim and is reacted at the side frame columns. The load transfer through the center plate rim has been shown to be one of the most severe loading conditions for the truck bolster (Ref. 2).

Wheels are subjected to thermal loading when tread brake shoes are applied. The deposition of energy in the rim leads to temperature gradients within the wheel. This causes an expansion and twisting of the rim relative to the plate. There are two distinct types of severe braking service which lead to significant thermal loads. The first is drag braking which is associated with the descent of a long grade. This might involve braking for over an hour at moderate rates of energy deposition in the tread of the wheel (e.g., 20 bhp). The relatively long duration of the brake application allows the heat to penetrate through the rim down into the plate (see Ref. 3). The second type of service is emergency braking. The relatively short time of brake application means that most of the heat is retained near the surface of the tread resulting in sharp thermal gradients within the wheel.

Failure Modes: Catastrophic failures of truck castings (side frames and bolsters) are usually due to the development of fatigue cracks. Because of the complexity of truck casting designs and the inherent characteristics of foundry practice, it is difficult to eliminate all conditions which may lead to the development of fatigue cracks. Fatigue cracks may originate at shrinkage cavities within the castings, the edges of holes, cracking strips, sharp surface corners, or geometrical discontinuities at core joints.

The development of a fatigue crack on a side frame is a relatively rare occurrence. The most common locations are through the tension member (see Fig. 2) or through the column. The tension member crack is the

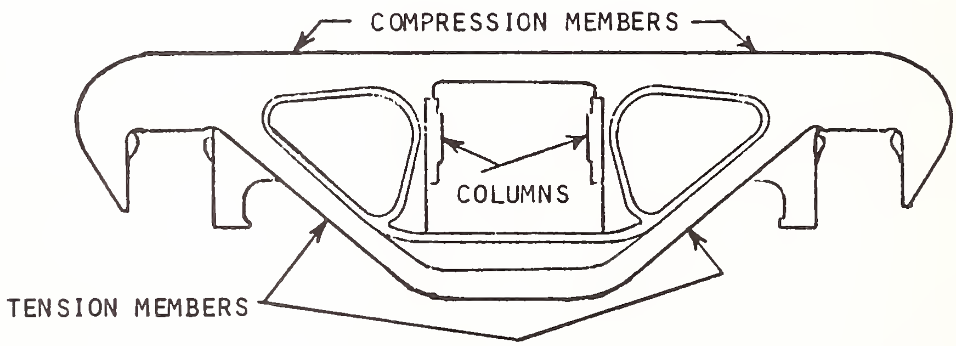

FIGURE 2 SIDE FRAME NOMENCLATURE

most critical because the propagation of the crack through the member can lead to the collapse of the side frame. On relatively rare occasions fatigue cracks develop in the compression member near the point where it joins the tension member above the pedestal opening.

Bolsters are also subject to the development of fatigue cracks. The most serious location of the initiation of a fatigue crack is on the bottom surface because these cracks eventually propagate up the side and lead to its collapse. These cracks often originate at one of the bottom lightner holes and propagate up to one of the side lightner holes (see Fig. 3).

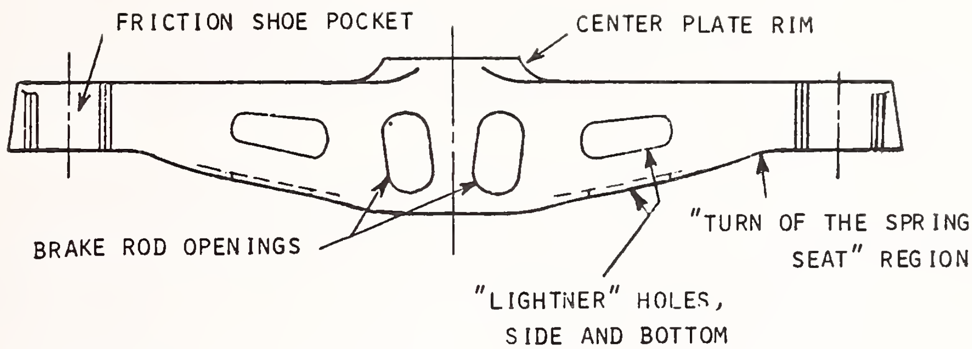

FIGURE 3 TRUCK BOLSTER NOMENCLATURE

Fatigue cracks also develop at the center of the bolster below the center plate and propagate up through the sides and inside ribs. Fatigue cracks can also develop on the bottom surface in the turn-of-the-spring-seat area, but this is a relatively rare occurrence.

The Association of American Railroads (AAR) Wheel and Axle Manual (Ref. 4) lists 22 types of wheel defects which require removal of wheels from service including excessively worn tread or flange, evidence of overheating, and the presence of cracks. From the standpoint of ensuring the structural adequacy of the wheel the most serious of these defects are thermal (radial rim) cracks, plate cracks and shattered rims because they are the type of defects which are most likely to lead to the catastrophic failure of the wheel.

Cracks which originate on the periphery of the wheel and propagate inward toward the hub are commonly referred to as thermal cracks. Once these cracks propagate through the rim they will often turn and propagate as a circumferential plate crack. Sometimes the crack will then reenter and propagate through the rim in a radial direction causing the separation of a large segment of the wheel. A thermal crack might be initiated and propagate over an extended period of time or the crack may develop rapidly causing a sudden failure of the wheel.

The thermal crack development is associated with modifications in the residual stress field within the wheel which is caused by severe tread braking. Under these conditions the periphery of the wheel is heated, but it is restrained from expanding by the colder plate and hub of the wheel. This causes the development of circumferential compressive stresses in the rim and radial tensile stresses in the plate. If the thermal gradient is large enough, plastic deformation will take place first in the highly stressed plate fillet regions and, with increasing severity of the thermal load, in the rim adjacent to the surface of the tread. If the thermal load causes inelastic deformation in the rim this region will develop residual tensile stress when the wheel cools. The existence of tensile stress in the rim will promote the initiation and growth of thermal cracks.

Cracks that initiate in the plate of the wheel and propagate circumferentially around the plate are commonly referred to as plate cracks. They normally develop and propagate as fatigue cracks from a point of initiation in the outside plate hub fillet. Plate crack development is believed to be caused by the effects of both thermal and mechanical loads. The highest stresses in the regions of crack initiation result from the thermal effect, but there are a relatively small number of cycles of high stresses over the life of the wheel. Mechanical wheel loads produce an alternating stress pattern within the critical fillet regions of the wheel once per wheel revolution, but the stress levels are of relatively low magnitude. The relative importance of these stress cycles has not yet been established.

The development of cracks in the rim slightly below the surface of the tread, and their propagation in a generally circumferential direction so that portions of the tread are broken off, is commonly referred to as spalling or shattered rim failure. It is caused by the fluctuations in the stresses surrounding the wheel/rail contact point as the load is repeated once per wheel revolution.

The most common cause of complete journal bearing failure is overheating. As a result of the loss of lubrication or some other defect in the bearing, the frictional forces increase causing the bearing to run hot (Ref. 5). This condition, if not detected, progressively becomes worse. Temperature in excess of 1500°F can be reached and the ability of the end of the axle to carry the bearing load is lost. As a result the end of the axle breaks off which often leads to a derailment.

Other modes of truck component failure include side bearings becoming loose and falling off the bolster, which leads to excessive sway of the car, and excessive wear in the components of the friction snubber system, which leads to uncontrolled oscillations of the car.

Accidents Caused by Failure of Freight Car Truck Components: Accidents caused by failures of freight car truck components are relatively rare events when one considers the large number of truck components in service. Table 1 presents a 5 year average of the number of derailments included in the Federal Railroad Administration (FRA) accident statistics which were caused by the failure of one of the principal freight car truck components. During this period there was an average of 29.4 billion freight car miles per year.

TABLE 1 DERAILMENTS CAUSED BY TRUCK COMPONENT FAILURES, 5 YEAR AVERAGE (1973-77)

Cause	Average Derailments Per Year
Broken or Loose Wheels	173
Worn Wheels	175
Broken Axles	15
Defective Journals	290
Defective Side Bearings	190
Broken Side Frames or Bolsters	63

The information in the FRA Railroad Accident/Incident Reporting System (RAIRS) data base can be processed to examine certain characteristics of derailments caused by the failure of freight car truck components. Of particular interest is the conditions under which a typical accident takes place and the resulting damage.

The results of such an analysis for 1977 data are summarized in Table 2. The first column shows the groupings of cause codes used to summarize categories of failure. The next column shows the total number of derailments that were analyzed for the year. The totals differ slightly from FRA accident summary tables because derailment cases were eliminated where certain items were omitted in the report.

The third column shows the median speed of derailment. The median value is used rather than the mean in order to get an indication of a typical derailment. Note that there is a large difference between certain derailment categories. For example, derailments caused by excessive wheel wear, improper truck swiveling, and defective side bearings occur most often at relatively low speeds. On the other hand derailments which are due to fractured components such as bolsters, side frames, wheels or axles tend to occur at moderately high speeds. This may be due to the fact that the final separation of the component is caused by a relatively high impact loading which is more apt to occur at higher speeds.

TABLE 2

ANALYSIS OF DERAILMENTS CAUSED BY TRUCK COMPONENT FAILURES

Defect Category (Cause Codes)	Number of Cases	Median Derailment Speed (mph)	Median Number of Cars Derailed	Derailment Damage (dollars)	
				Median	90 Percentile
Defective Side Bearings (440-2)	234	15	4	$ 7,900	$ 46,000
Broken Truck Bolster (443)	25	35	3	5,200	500,000
Broken Truck Side Frames (444)	9	30	1	10,000	270,000
Improper Truck Swiveling (445)	89	8	3	7,400	49,000
Defective Truck Springs or Snubbers (446-7)	11	17	3	12,500	80,000
Broken Wheels (460-3)	145	38	2	13,000	200,000
Excessive Wheel Wear (464-6)	171	5	4	5,000	20,000
Loose Wheels (467)	41	23	4	14,700	220,000
Broken Axles (450)	13	40	4	19,200	120,000
Defective Journals (451-454)	269	30	2	14,000	140,000

The fourth column in the table shows the number of cars derailed. Again the median value is used. Note that there are no major differences. Two, three, or four car derailments are typical. This is somewhat surprising in view of the differences indicated for the median speeds of derailment. It would be expected that higher speed derailments would result in more derailed cars, but a further review of the data indicates no significant correlation between speed and the number of derailed cars.

The fifth and sixth columns show the value of damage caused by the derailment. This includes both equipment and track damage. The fifth column shows the median value. There is no significant trend except a slight correlation with respect to speed. Typical derailments produce $5,000 to $15,000 damage. The sixth column gives an indication of the potential severity of a derailment by presenting the 90 percentile damage value. The meaning of this value is that 90 percent of the derailments would have damages below the level indicated and 10 percent would have damages above this value. There are large differences between derailment causes. Derailments resulting from excessive wheel wear show a 90 percentile value of $19,000 where $500,000 is shown for broken truck bolsters. Note that broken components like wheels and truck castings have the largest values and thus offer the potential for severe derailments.

Removal of Defective Components: The study of accident statistics alone does not present a complete picture of the number of components which fail in service. Many components are found to be defective during routine inspections and are removed before they fail. Thus the number of components which become defective under service conditions is substantially larger than the number of component failures which lead to accidents.

The AAR Truck Casting Removel Reports, for example, indicate that about 1000 truck bolsters and approximately one-fourth this many side frames are removed from service yearly because they are defective. Excessive wear is one of the major reasons for their removal. When severe wear patterns develop they occur at the interconnections between components (e.g., the bolster friction shoe pockets, side frame columns, the bolster center plate area).

The most common location for the development of severe wear on a side frame is the column. This type of wear results from interactions with the truck bolster caused by persistent truck swiveling motions or by excessive vertical oscillations resulting from a poorly damped suspension system.

Many bolsters are removed from service because of cracked or broken center plate rims. This is generally due to high speed car coupling impacts of empty cars. Another reason for bolster removal is the development of badly worn center plate rims. The major cause of this condition is uncontrolled truck swiveling motions which occur during high speed operations with light weight cars.

Several hundred thousand wheels are removed from service each year because of defects. An analysis of the reasons for their removal shows that only a small number (on the order of 3 percent) are removed because of evidence of cracking or overheating, which would make the wheel more susceptible to fracture (Ref. 6).

Bearings are usually removed from service for seal defects, loose end caps, and evidence of overheating(Ref. 5). It has been established that in many cases overheating results from overlubrication of the bearing. When bearings are removed for inspection they are examined for defects such as oversized ring or cone, fatigue pits or spalls on a cup race, brinelling, and corrosion on roller or raceway.

Design Specifications: Side frames and bolsters are cast with low to moderate carbon steels meeting the requirements of AAR Specification M201 (Ref. 7). Most castings are made with either Grades B or C steel having minimum yield strengths of 38 and 60 ksi respectively.

Bolster and side frame castings must be designed in accordance with AAR specification M202 and M203 (Ref. 7). These specifications require that nominal design stresses must be kept below 16,000 psi in Grade B castings for prescribed combinations of vertical and lateral loads. The corresponding maximum design stress for Grade C castings is 25,000 psi. The specified vertical load used in side frame design calculations is 1.5 times the nominal maximum static load on the side frame. The specified vertical design load used in the bolster calculation is the nominal static load.

Side frame and bolster designs must be subjected to static load tests in order to demonstrate certain load/deflection criteria and maximum load carrying capacity.

Side frames are also subject to fatigue test requirements under M203. This specification calls for new side frame designs to be tested in machines which apply a complex pattern of vertical, lateral, twist, and impact loads on side frame castings. The requirement calls for testing a group of four sample castings for an average of 100,000 cycles of load without the development and extension of a transverse crack to 1/2 inch.

Railroad wheels are manufactured in accordance with AAR Specification M107 for wrought steel wheels or M208 for cast steel wheels (Ref. 7). Five classes of moderate to high carbon wheel steels are defined on the basis of carbon content and hardness.

There are no design stresses specified for wheels. Instead various geometric configurations are specified. Minimum dimensions are specified at critical locations (e.g., plate thicknesses at hub and rim fillets).

Roller bearings must possess a minimum life expectancy of 500,000 miles with the full rail load acting radially for one-half the mileage. Life expectancy is defined as the life at which no more than 10 percent of the bearings require replacement due to fatigue initiated spalling or flaking on the load carrying surfaces (Ref. 8).

Axle design is specified in terms of dimensional requirements which are based on limitations in stress values (Ref. 4). The allowable stress levels reflect extensive fatigue tests.

Inspections: Inspection requirements for freight car truck components are based on the AAR Interchange Rules and the current FRA Railroad Freight Car Safety Standards (49CFR Part 215). The FRA standards require a predeparture inspection to insure that cars are not dispatched with critical defects which have the potential to cause derailments. Wheels must be checked for excessive wear, cracks, and signs of overheating. Any axle with a crack or with a gouge more than 1/8 inch deep must be withdrawn from service. Roller bearings must be checked for signs of overheating and damaged seals. Roller bearings must also be checked for defects when disassembled to permit wheel replacement. Side frames and bolsters must not be permitted to operate with cracks 1/4 inch or more in the transverse direction on a tension member.

Actions to Improve Reliability: A number of activities are underway which are expected to improve the reliability of freight car trucks. Wheels are expected to become more reliable by the development of an improved design specification. The AAR has a program to determine the feasibility of specifying maximum design stresses under various load conditions as a means of improving the integrity of wheel designs. The feasibility of using a design stress condition has been made possible by the development of modern finite element analysis techniques which allow for the calculation of stresses in wheels under complex loading conditions including mechanical loads at the wheel/rail interface and thermal loads introduced by tread braking.

The reliability of truck bolsters is expected to be improved by the development of a fatigue performance specification. The RPI/AAR Truck Research Safety and Test Project has been working on this specification for several years. The work began by making measurements of the loads acting on freight car trucks over many thousands of miles of railroad service in order to adequately define the environment in which freight car truck components operate (Ref 9). Fatigue tests were also conducted on a number of bolsters to determine their fatigue strength. Based on these results it has been possible to develop recommendations for a fatigue test specification which will insure that a given bolster design has an adequate margin against failure during its anticipated life.

A trend in the construction of conventional freight car trucks is utilization of longer spring travel. Most new cars now are equipped with

3-11/16 inch travel springs. Within the last 2 years, the use of the D-7 spring, which gives 4-1/4 inch travel, was introduced. Wheel/rail interaction forces are significantly reduced by the use of longer travel suspension springs.

The concern over truck hunting phenomena has led to the development of devices which prevent this unstable type of motion. One way of reducing the hunting tendency is to provide some damping restraint to truck swivel motions. The use of the center plate extension pad or constant-contact side bearings are ways of providing this restraint.

Over the last several years, much activity has been directed toward the development of new types of freight car trucks. The goal has been the development of trucks with features which will reduce wheel wear and wheel/rail interaction as a means of increasing safety and reliability. These trucks would be more expensive than conventional trucks, but the reduction in wheel wear and other beneficial effects would be expected to more than compensate for the added cost.

Different design philosophies are evident in these advanced truck designs. Truck hunting phenomena, for example, are minimized if the truck is rigidized to reduce its tendency for out-of-square deformations, and this feature is evident in most advanced truck designs. The principal design objective of other trucks is the reduction of wheel wear and wheel/rail forces on curves through the use of self-steering mechanisms (Ref. 10).

Long Range Activities: Work of a long range nature which is now in progress can be expected to lead to more reliable freight car truck operation. In recent years instrumentation systems have been developed for accurately measuring the wheel/rail interaction forces. Measurement programs have been conducted, and more are planned, to quantify this aspect of the service load environment. The lack of such data in the past has hampered efforts to get more reliable designs.

Other experimental work is being conducted at the Facility for Accelerated Service Testing. This facility is at the Transportation Test Center (TTC) and is used to investigate the long-term effects of railroad operations on track and equipment. This work is being conducted on a special track which is made up of a number of different types of track construction. A mixed train of 100-ton capacity cars operates over the track to generate a high rate of usage.

Another facility, the Stability Assessment Facility for Equipment, is being developed at TTC to evaluate the dynamic response characteristics of rail vehicles. The track is instrumented to measure vertical, lateral, and longitudinal forces, and displacements. The track will contain known geometric variations to induce vehicle dynamic modes.

The utilization of advanced wayside inspection equipment offers the possibility of identifying and removing defective cars before they lead to accidents. Some wayside inspection systems, like hot box detectors, have been utilized for many years. Other equipment, such as a system to detect improper dynamic operation of the suspension system, is in the early stages of development. A joint government/industry group has been organized to oversee the installation and operation of a wide variety of wayside inspection systems at TTC. These devices are located on a special track section where cars containing known defects can be used to examine the responsiveness of the detection equipment.

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FRACTURE OF STEEL PLATE MATERIALS UNDER ABUSIVE SERVICE CONDITIONS IN RAILROAD TANK CARS

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Abstract: In recognition of the potential for catastrophic accidents involving railroad tank-car operations, the Federal Railroad Administration has actively participated in a continuing industry/government research program to enhance the safety of these operations. Selected parts of this program will be discussed with emphasis on the challenges presented by abusive service conditions that are associated with accidents, the ameliorative measures that have been taken to date, and the direction of the present work aimed at furnishing improved guidelines for fracture-safe design.

Key words: Fracture control; hazardous materials; impact transition; pressurized tank car; stress-rupture.

Background--Studies related to the behavior of railroad tank cars in abusive service are conducted in recognition of the potential for hazardous derailment and costly other consequences of failures of railroad components. Facts about railroad service in the United States indicate that freight service is big business [1]. About 500 billion net ton miles of cargo are transported annually in about two million freight cars. Out of this total railroad freight, about 40 billion net ton miles of cargo are transported in almost two hundred thousand tank cars. As tank cars are commonly used to transport hazardous materials, their safe performance under normal operating conditions as well as in accidents is of great importance to the railroad industry and to the public. Recent data [2] indicate that the accident rate for all hazardous materials shipped as railroad freight (primarily in tank cars) is about twenty accidents per billion ton miles. Thus, hazardous materials, are involved in about 800 railroad accidents annually. In these accidents, the number of tank cars carrying hazardous materials is of the order of 4000, and about 200 of these cars release hazardous materials as a result of these accidents.

Accidents involving tank cars include both self-pressurized and unpressurized cars. Unpressurized cars are used to transport commodities that have very low vapor pressures, such as flammable liquids (like naptha) as well as various dry chemicals (like sulfur). Self-pressurized tank cars are used to transport liquified gases that have significant vapor pressures under normal operating conditions at atmospheric temperatures. These pressurized liquids include chlorine, anhydrous ammonia, and propane.

While accidents involving the unpressurized cars can sometimes be serious and costly, these cars can often be abused badly and even ruptured without consequences of significance. Pressurized tank cars have a much greater potential for catastrophe in accidents. The lading is frequently a liquid, which may be flammable or toxic, with a significant vapor pressure that increases with temperature and gives rise to increases in the applied tensile stresses in the shell- and head-plate materials of the tank car. The increased applied stresses can lead to a decrease in the critical flaw size required for unstable fracture, as the critical flaw size is related to the inverse square of the applied stress. For these and other reasons, fracture-safe design is a continuing concern for the pressurized tank cars.

Pressurized steel tank cars are produced to the Department of Transportation (DOT) specifications for four basic classes: 105A, 109A, 112A, and 114A, with Classes 114 and 112 having special requirements [3]. Although pressurized cars are sometimes fabricated from aluminum plate materials (some cars of DOT Classes 105 and 109), the studies at the National Bureau of Standards (NBS) have been concerned only with steel plate materials. Most commonly, these steels are produced to specification AAR M-128 (Grades A and B). Other grades of steel permitted [3] for use in this application are produced to various ASTM specifications and they include ASTM A285-70a (Grades A, B, and C), A515-70, and A516-70a (Grades 55, 60, 65, and 70), A537 (Grade A), and A302-70a (Grade B).

Pressurized tank cars are fabricated by welding and they consist of a central section of a number of shell courses. This central section is closed at each end by a 2:1 semi-eliptical head plate. The shell plates are cold formed in the as-rolled condition. The head plates are hot formed and the temperatures of forming should be controlled in a manner that produces properties equivalent to those of normalized plates. After forming, the plates are welded and the welded vessel, with its integral attachments, is stress relieved. While some anomalies in the welding practices employed on particular vessels have been observed [4], results of impact tests of weldments have been favorable [5], and in general weldments have experienced a rather favorable service record; welding practices will not be discussed in this report.

A principal focus of the NBS studies of the plate materials used in the fabrication of pressurized cars has been on a steel that is most commonly used for these cars, AAR TC-128 steel (Grades A and B). This report will deal with pressurized tank cars in general and with the relationship between the properties of TC-128 steel and the requirements of the abusive service that pressurized cars are apt to experience in service.

This steel is produced to the AAR M-128 specification [6], which is revised periodically. This steel is commonly used as the plate material for tank cars that are used in the transportation of hazardous materials such as cars of the DOT Classes 112 and 114, which are used in transportation of propane and anhydrous ammonia. In this report, some aspects of the M-128 specification will be discussed in relation to the requirements for service. In addition, general observations on the microstructure, the mechanical properties (especially impact properties), and the quality of the plate material will be summarized. This is followed by a brief discussion of the FRA/AAR cooperative programs that led to a rulemaking known as HM144 for hazardous materials, which involves significant changes in the regulations for transportation of hazardous materials. Finally, work in progress on the development of analytical models for computation of critical flaw sizes for unstable fracture in the pressure vessels used in this application will be mentioned. In keeping with the theme of this symposium, significant problems that have been solved recently and challenging problems for the coming decade will be highlighted.

Comments on Specification AAR M-128--The specification M-128.00 for high strength carbon-manganese steel plates for tank cars requires a tensile strength in the range of from 81,000 to 101,000 psi and a minimum yield point of 50,000 psi, with elongation in two inches to exceed 19.0 percent [6]. Under this specification, AAR TC-128 Grades A and B steel plate materials are produced to fine-grain practice and the plates are required to be normalized only when used for low temperature service.

The requirements for chemical composition, given as percent by weight, include maximum levels of carbon and manganese of 0.25 and 1.35, respectively. Thus, the ratio of manganese to carbon (Mn/C) for steels produced to this specification would be about five. This is favorable when compared to that of older tank-car steels that are no longer permitted for this application, ASTM A212-A and -B steels. These older steels had still higher maximum levels of carbon (0.28 and 0.31 for Grades A and B, respectively) and a lower maximum manganese level of 0.90 percent. The more favorable Mn/C ratio of the TC-128 steels would tend to promote improved impact transition temperatures.

For cars built before 1978, the maximum level of phosphorus was 0.04 percent and that of sulfur was 0.05 percent. These levels can be achieved without special steelmaking practices and they might be considered to be generously high. In July of 1978, these maximum levels were adjusted downward to the current maximum phosphorous and sulfur levels of 0.035 and 0.040, respectively. Steel that had been aluminum killed and produced to meet the requirements of Grade B of the M-128 specification (pre-1978) has been shown [7] to fail the bend test. The inclusion content of this particular steel is high in comparison with other plates tested (4, 5, 8), all of which passed the bend requirement of the M-128 specification. The bend test requires that the specimens shall stand being bent cold through 180° without cracking on the outside of the bend portion, through an inside diameter equal to twice the thickness of the plate.

While the bend test can be used as a qualitative test of ductility and steel quality, it is not a very definitive test related to the requirements for steel plates in abusive service. In this type of service, temperatures may be significantly below room temperature, at which the bend test is conducted, and loading rates may be substantially greater than those used to conduct the bend test. The bend specimen is neither notched nor pre-cracked. However, the fracture behavior of steel plates of the types in question are well known to be sensitive to strain rate, temperature, and the presence of notches and cracks. For these and other reasons, the bend test is not considered a definitive test of resistance to fracture. A definitive test must be one that is based on some measure of fracture toughness.

Fracture Toughness--For abusive service, it becomes apparent that a balance of properties is required. The requirements for strength and ductility are based on traditional design requirements. The results of tensile tests provide adequate measures of the yield strength, the tensile strength, and the ductility. For abusive service, other properties may become important. It will be shown later that the short-term creep (or stress-rupture) properties can be important when the plate materials are exposed to a fire. Further, the fracture toughness must be considered. This material property governs the size of a flaw that will propagate unstably in a plate that has been ruptured or partially ruptured in an accident.

Fracture toughness is generically defined as the resistance to propagation of a crack. Various measures of fracture toughness are available. For purposes of this report, these measures are categorized into two classes, direct measures and indirect (or alternative) measures. The direct measures are those that have a highly analytical basis in fracture mechanics. The results of these measures are normally expressed in terms of the stress-intensity factor (K), or values of J derived using the J-integral concept. The results of these tests are used directly in fracture control. Normally, these tests are expensive when compared with the alternative tests.

Results of alternative tests are often useful for fracture control, based upon either service experience or empirical correlations with fracture mechanics tests. While there are about seven different shapes of specimens used as alternative tests for steels of the types in question here, only one is used widely. It is the single-edge notched specimen, tested in three-point bending, such as the Charpy V-notch (CVN) specimen and the dynamic-tear (DT) specimen; another common test is the nil-ductility transition (NDT) test, which uses a face-notched crack-starter weldment in a specimen tested in three-point bending.

These types of tests are used to characterize the fracture toughness of tank-car steels because steels of the types used in this application usually exhibit a strong transition from ductile to brittle fracture and under impact loading this transition can occur at temperatures in the service range of temperatures. Further, these tests are generally inexpensive and effective in characterization of the transition.

With the above mentioned balance of properties in mind, the fracture resistance of TC-128 and other carbon-manganese steels used in tank cars is now further described. The microstructure is a mixture of ferrite and pearlite that is characteristic of carbon-manganese steels. The fracture resistance of these steels is highly dependent on orientation of the test specimen [5,9]. This is illustrated in Figure 1, which gives the CVN energy absorption data for six orientations of specimens. The specimens were tested over a range of temperatures selected to establish the transition from ductile to brittle behavior. Specimen orientations are designated by a two-letter code, with the first letter representing the normal to the crack plane and the second letter representing the expected direction of crack propagation. The letters L, T, and S are used to represent the longitudinal, transverse, and short-transverse directions of the plate.

The effects of orientation of specimen and notch were found to be similar for each of two steels tested, a head plate and a shell plate [9]. In this work, it was shown that the orientations of the specimen and the notch: (1) greatly affect upper-shelf values of lateral expansion (L.E.) and of energy absorption (E.A.); (2) affects the transition temperatures at which a given level of L.E. or E.A. is obtained; and (3) has little effect on the lower-shelf values of L.E. and E.A. or on the temperature limits of the transition zone or on the 50 percent shear-fracture-appearance values.

As an example of the above specimen orientation code, in TL specimens, the normal to the crack plane is the transverse direction. In TL specimens, the direction of crack propagation is the longitudinal (principal rolling) direction. Specimens of this orientation are commonly called transverse specimens. This is the orientation of lowest resistance to extension of a through-thickness flaw, and it is commonly called for in specifications.

Resistance to extension of flaws present on the plate surfaces, represented by the TS and LS orientations, is greater than that for through-thickness flaws. Internal flaws, those contained within the surfaces of the plate, are represented by orientations SL and ST. Tests of the SL and ST orientations reflect the resistance to extension of these internal flaws under stress that is normal to the rolled surfaces of the plate, such as the stress that results from the welding of attachments to the surfaces of the plate materials.

The results shown in Figure 1 indicate that resistance to extension of surface flaws (LS and TS) is greater than that for the corresponding through-thickness flaws (LT and TL) and resistance to extension of internal flaws (SL and ST) is least. This anisotropy of impact properties typifies that of many steels commonly used for pressure-vessel applications, and it demonstrates that resistance to propagation is highly dependent on orientation. This anisotropy, and the fracture resistance in general, are related to the quantity, types, and shapes of inclusions in steels. Therefore, mill practices have been developed to minimize these tendencies. For example, transverse properties in plate materials are enhanced by (1) more extensive cross-rolling of the plates and (2) inclusion shape control by chemical means. Fracture toughness is generally improved by decreasing the amount of sulfur, as this decreases the number and size of manganese-sulfide inclusions.

Thus, transition temperatures are affected by the orientation of the test specimen. This is one of many factors. Others include specimen geometry, loading rate, and the presence of a pre-crack.

The dynamic-tear (DT) specimen is much larger than the CVN specimen. It is 5/8-inch thick by 7 1/8-inches long by 1.6 inches wide. The CVN specimen is 10 mm (0.394 inches) square by 55 mm (2.165 inches) long. The thickness of plate materials used for tank cars is often 5/8- to 11/16-inches thick. As a result, the DT specimen can be prepared as a full-plate thickness specimen, but the CVN specimen represents only part of the plate thickness. Therefore, variations in properties throughout the thickness, which sometimes are observed particularly in head plates [8,9], could be misrepresented in tests of Charpy specimens. Further, it is noted that, due principally to its greater thickness, the DT specimen yields higher transition temperatures when compared with those of the CVN specimen. However, when the CVN specimen is tested in the pre-cracked condition, the transition temperatures are often similar to those of the DT test specimens. Thus, in using these alternative tests for estimation of fracture toughness of pressure-vessel steels, it is important to recognize that the CVN specimen is not a full plate thickness specimen. Further, if the CVN specimen is not pre-cracked, it is likely to give transition temperatures that are significantly lower than those obtained with either a pre-cracked CVN specimen or with the DT specimen.

The DT specimen is prepared with a sharp pressed notch. During a DT test this sharp notch facilitates initiation of a crack. Then the crack propagates to completely fracture the specimen; the bulk of the energy absorbed in this test represents the energy required to propagate the crack. With its 1.6 inch width, the full crack extension is more than three times that obtained with a CVN specimen. This permits the DT specimen to build up large shear lips during propagation of the crack along the full width of the run-out distance of the specimen. Thus, with this long propagation path, this specimen is suitable for comparing materials for relative resistance to propagation of a crack. The comparison can be made using either measures of energy absorption or lateral expansion. The latter is measured at the striking line of the specimen.

Using the DT test, energy absorption and lateral expansion of longitudinal (LT) and transverse (TL) specimens were measured for each of two plates of TC-128 steel [9]. Tests were conducted over a range of temperatures that extends well beyond the onset of the ductile upper shelf.

The DT test lateral-expansion data are shown in Figure 2 for one of these two steels. These results, and those for the other plate that is not shown, indicate an embrittlement at about 550 °C (1050 °F). The nature of this embrittlement has not been fully established. In an attempt to explain this embrittlement [10], CVN impact specimens were exposed for 30 minutes at 550 °C and then tested over the range of near-ambient temperatures at which the transition from brittle to ductile fracture is normally observed. The results of the untempered (or unexposed) specimens were not significantly different from those for the exposed or tempered specimens, and therefore it was concluded that this embrittlement is not due to temper embrittlement of the steel.

What remains are the questions of (1) the mechanism of this embrittlement, (2) the effect of time of exposure on the magnitude of the embrittlement, and (3) the significance of this embrittlement in relation to service performance.

Further, it is noted that the resistance to crack propagation in the transverse direction (the lower line in Figure 2) is substantially lower than that of longitudinal specimens. The direction of crack propagation represented by transverse specimens is the circumferential direction in the tank-car shell plates. These observations give rise to another question: "Can improved transverse properties substantially reduce the tendency for an undesirable type of circumferential failure in tank cars exposed to elevated temperatures?" This type of failure, shown in Figure 3, is not uncommon and it leads to rocketing of a section of the tank car. The rocketing behavior is serious, with apparently higher potential for personal injuries or property damage. The magnitude of the decrease in fracture resistance of tank-car steels at elevated temperatures (the 550 °C embrittlement) appears to be sufficiently large so that a significant reduction in the resistance to crack propagation is to be expected at temperatures near 550 °C. Further, because transverse properties are much poorer than longitudinal, the potential for the development of an easier path of propagation in the circumferential direction of a tank car exists. For these reasons, this work will be continued.

<u>Cooperative FRA/AAR Research</u>--The NBS studies on tank-car steels are part of the broader research programs conducted cooperatively by the Federal Railroad Administration (FRA) and the Association of American Railroads (AAR). As part of this FRA/AAR cooperative effort, tank cars have been subjected to collisions similar to those encountered in rail yards. These tests were used to understand the events that lead to failures of tank cars. In addition, statistics of actual impact events from accidents have been compiled by the FRA in an attempt to determine the most vulnerable area (or the area most worthy of protection) on a tank car.

When tank cars fail in accidents, fires sometime develop. Due to the serious consequences of these fires, the cooperative research program has included fire tests of tank cars. As a result of these studies, it was found [11] that the time to failure for a tank car exposed to an all-enveloping fire could be significantly increased by the use of an ablative coating on the plate materials of the tank car. An uncoated car failed in only 24.6 minutes, just about the time that it takes for a fire company to arrive at the scene of an accident and set up its equipment. A thermally coated car remained for 93.7 minutes before it failed; this car had vented most of its hazardous cargo (propane) before the failure occurred. As a result, the explosion that accompanied the failure of the thermally shielded car was much less violent.

As part of this work, a failure analysis of the insulated tank car, was conducted [12]. This analysis indicated that the failure mechanism begins with the formation of a stress-rupture (short-time creep) crack which extends by void coalescence until its length is sufficient for further extension by tensile overload or shear failure. The stress-rupture properties of this plate of TC-128-B steel were characterized [13]. This work points up the importance of the elevated-temperature properties of tank-car steels. The strength and the notch tensile strength at temperatures in the range of from 430 °C (800 °F) to 680 °C (1250 °F), as well as the stress-rupture properties for times within two hours of exposure, are important material properties that must be considered in the overall balance of properties that are needed for optimum performance of plate materials used for tank cars in abusive service.

Improved Safety--As a result of these NBS studies and many other studies in the cooperative FRA/AAR program, a regulation known as Docket HM144 for Transportation of Hazardous Materials was written into law. This rulemaking is designed to improve the safety of transportation of flammable compressed gases and anhydrous ammonia. It provides specifications for thermal shielding, improved couplers, and puncture protection of head plates in pressurized tank cars of DOT Classes 112 and 114. The thermal shields may be either a coating of insulating materials or insulating materials encased in an outer steel jacket. The couplers are shelf couplers of the types E or F, which are designed to prevent cars from uncoupling in accidents and thereby to prevent punctures. The head-plate puncture protection involves options, one of which is the use of head shields that are installed in front of the head plates at both ends of the tank cars. Their presence increases the relative speed required to cause fracture from two colliding cars. Thus, the regulation involves three steps for improved safety: thermal shielding, improved couplers, and head shields. In addition, relief valve capacity has been altered for improved performance in accidents. Existing cars will be retrofitted by the end of 1980 and new cars are built to these new requirements.

Future Work--The optimum balance of properties for plate materials used in tank-car service must be based upon the requirements of normal service and abusive service. This fact has focused attention on various approaches to fracture-safe design. One analysis [14] has indicated that the fracture toughness of plate materials used for pressurized tank cars is adequate for the normal service requirements. This means that the flaws inherent in the plate materials and those that are introduced during fabrication are small in relation to the service stresses and the fracture toughness of the materials. Under abusive service, large flaws are punctured into the vessel, stresses can become relatively much higher, and the propagation of these large flaws can be expected. The questions are, "What levels of flaw size can be tolerated with the present grades of steels that are used in this application and are there alternative steels that would furnish a better balance of properties for improved fracture resistance at ambient and elevated temperatures?"

The present specifications [3] for steel plate materials used for pressurized tank cars has no fracture toughness requirement, except when the steel is used for low temperature service--such as for the transportation of carbon dioxide, which requires steel with 15 ft·lb energy absorption at -50 °F (-46 °C). Thus, the fracture toughness of the plate materials used in most pressurized tank cars is assumed to be adequate for fracture-safe design, provided that the steel meets the chemical and other requirements. The specifications do include a thickness requirement that is based upon the burst pressure. This is a design requirement that assumes the plastic collapse mode of failure. Under this failure mode, an increase in the strength of the material leads to increases in the burst pressure and in the critical size of the flaw required for unstable fracture. The flaw sizes for the plastic collapse mode of failure are greater than those for the elastic or elastic-plastic failure modes, and this makes the transition temperature of the material an important consideration.

To assure that failure will be by the plastic collapse mode, the ductile-to-brittle transition must occur at temperatures below the service temperatures at which failures are anticipated. When it is not that low, the analytical problem is to relate the fracture toughness to the critical flaw size for the elastic or elastic-plastic failure mode under which failure is expected. When this is done, adequate resistance to crack extension can be assured by judicious selection of materials on the basis of their fracture toughness properties.

The transition temperatures of materials used in the current pressurized tank cars are not sufficiently low to always assure upper-shelf plastic collapse behavior. Thus, failure is to be expected under the other failure modes, and analytical models are being developed for understanding the levels of protection that can be expected from the materials now in use and from candidate materials for the future.

Hence, the paradox is that both improved strength and improved toughness may be sought for improved service performance in pressurized tank cars. These properties are normally inversely related, unless cost is of no concern. Therefore, the challenge of the coming years is to furnish this improved balance of properties, including improved stress-rupture properties, at a marginal increase in cost of the steel plate materials. It has been stated [14] that no significant transition-temperature improvement can be achieved by changing to other ferrite-pearlite types of steels. In view of the balance of properties required for this application, this author would agree with that statement. Perhaps other types of steels hold promise for the future. Candidate materials would have to include micro-alloyed steels, commonly referred to as high-strength low-alloy (HSLA) steels.

For the ambient-temperature aspect of the problem, there are two approaches to fracture-safe design. One is the historic transition temperature approach which provides safety against unstable fracture by specifying that the lowest anticipated service temperature always be above some ductile-brittle transition temperature (DBTT). The DBTT can be defined by tests such as the drop-weight NDT, the Charpy V-notch, and the dynamic tear. Since the transition temperature approach does not directly specify acceptable stress levels or flaw sizes, it is highly dependent on correlations with service performance.

The second approach to fracture-safe design is based on fracture mechanics and it was initially developed to handle materials that do not exhibit a ductile-to-brittle transition behavior (high strength steels, aluminum, titanium). This approach provides the relationship between the applied stress on a flawed structure and the stress intensity (K) at the leading edge of the flaw. This approach was initially used in applications in which plane-strain conditions are developed near the front of a crack. In tougher materials, this condition does not commonly prevail and methods have been developed to deal with plane-stress conditions near the crack front. In addition, there are many versions of the fracture mechanics approach, and a number of fracture toughness parameters are now used, such as the dynamic initiation fracture toughness, crack arrest fracture toughness, and elastic-plastic fracture toughness.

Attempts have been made to correlate data from the two fracture-safe design approaches (DBTT and fracture mechanics), and to mix the two approaches into a hybrid approach such as the ASME reference fracture toughness curve (K_{IR}) approach.

Future work on pressurized railroad tank cars will examine past and current attempts to apply these two approaches to fracture-safe design. A preliminary effort has already been published [15] by the AAR. It incorporates the K_{IR} curve concept and several parameters from the DBTT approach. This is a beginning. Other studies are in progress under sponsorship of the AAR and the FRA. An approach being developed at the

NBS incorporates direct and indirect measures of fracture toughness. The direct measures are used in the establishment of the toughness requirements for the application and the indirect measures are used through correlation with results of the direct measures, in the establishment of materials qualification standards.

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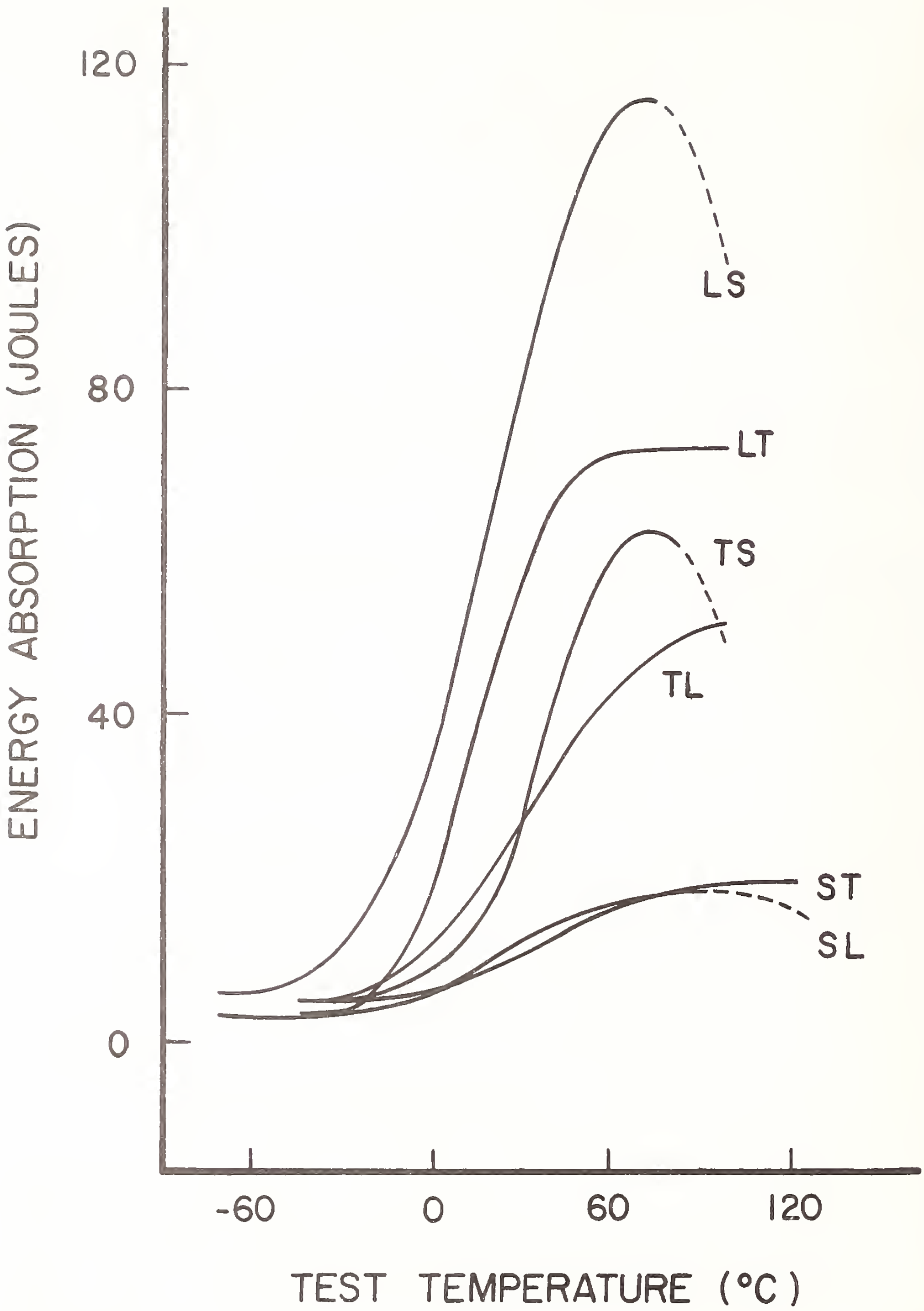

Figure 1. CVN impact test results for six orientations of AAR TC128-B steel specimens.

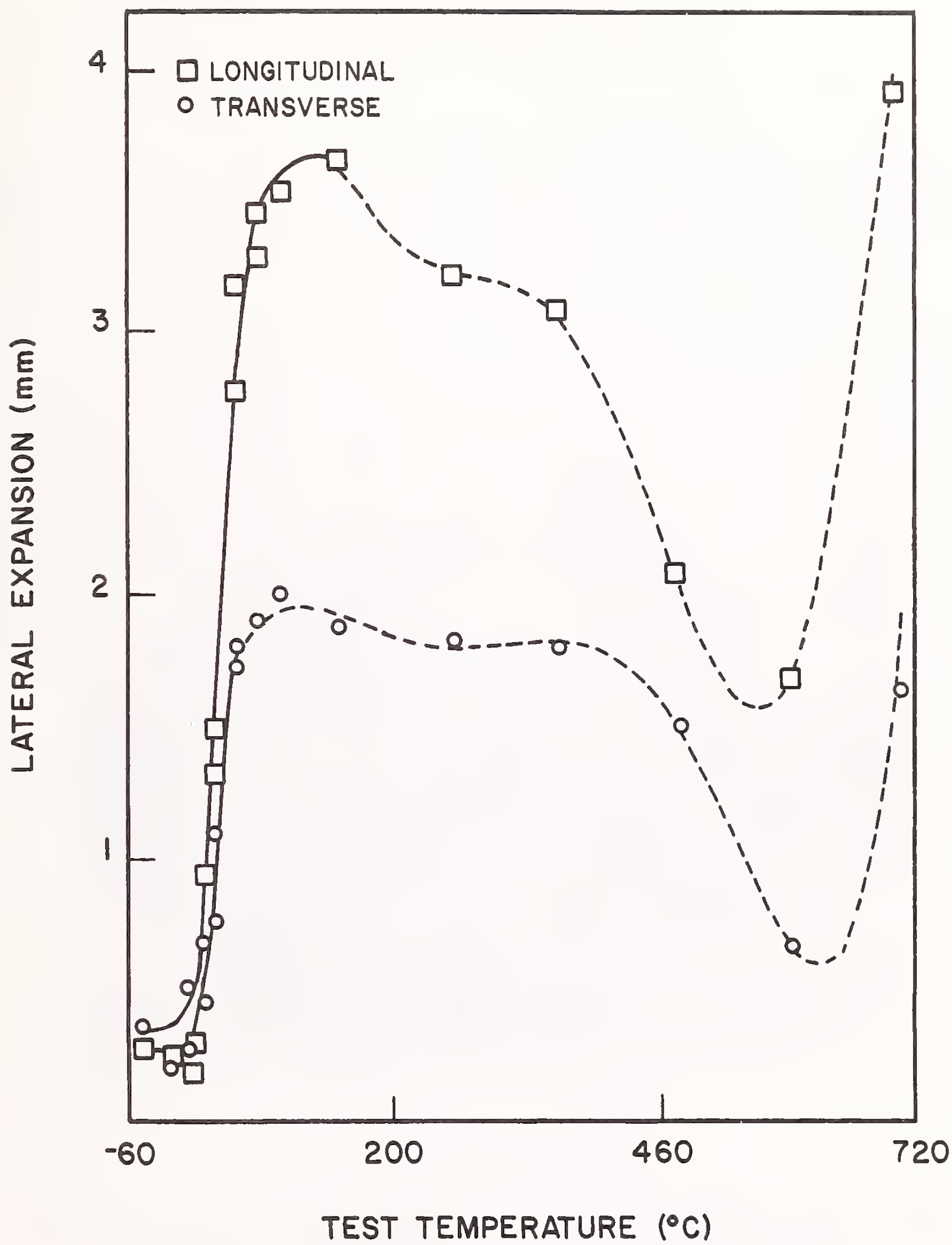

Figure 2. Lateral expansion in dynamic tear (DT) tests conducted over a broad range of temperatures.

Figure 3. A tank car failed at elavated temperatures showing a circumferential crack propagation.

FRACTURE ANALYSIS OF CAST STEEL COMPONENTS IN RAIL VEHICLES

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Abstract: Couplers in rail vehicles made of AAR C-grade cast steel have been tested in the laboratory in simulated, full-scale fatigue tests. Fracture analyses were carried out on broken components to characterize cause and mode of failure for comparison with field failures. The majority of the failures initiated at flaws on the surface, mainly casting defects, characterized to fall in the range of K_f=1.5. In the laboratory, cracks initiating at these flaws propagated in fatigue to the point where catastrophic failure occurred in a brittle mode. This fracture sequence is similar to that found in field failures. A metallurgical analysis which included microscopic examination as well as mechanical testing was carried out. Most of the failed components were composed of normalized steel with pearlitic structures having poor dynamic toughness and transition impact properties which are believed to be the major factors leading to failure.

Key words: Cast steels; fatigue crack growth rates; fracture analysis; mechanical testing; microstructure; rail vehicles; SEM fractography.

Introduction: The railroad industry is the largest consumer of cast steels for rail vehicle parts such as couplers, yokes, knuckles, bolsters and side-frames. These parts are made from four grades of cast steel, differing from one another by heat treatment and slightly by chemical composition, resulting in a variety of microstructures as well as strength and toughness levels [1].

The couplers, yokes and knuckles are made primarily of C-grade quenched and tempered steel and C-grade normalized and tempered steel. In 1971-1972, a joint project of AAR and a few railroads was carried out in which couplers, yokes and knuckles that had failed in service were examined [2],[3]. Within a period of one year, and with only a few railroads participating, there were over 4,000 failed parts reported. Since that time, a shift was recommended towards the C(Q&T) steel which is believed to show a better fatigue life endurance. A full-scale laboratory fatigue test, simulating real-life loading history, was conducted to study cause and mode of failure for comparison with field failures [4]. The subject of this paper is the metallurgical fracture analysis of these failed components.

Experimental Procedure: Within the framework of the laboratory tests, four couplers, two yokes and twenty knuckles were tested to failure. Load, cycles to failure, and a summary of test conditions were noted and a preliminary visual inspection of the failed parts and fracture surfaces was made [5]. All failed parts were then carefully examined to reveal fracture origins and defects to be considered as possible crack initiation sites, and were photographed for documentation. The failed parts were then sectioned to provide samples for various tests. Fractography was performed using an ISI-60 scanning electron microscope. Chemical analyses were conducted at random representative locations to check for specification compliance. Microstructures were examined to reveal micro-defects and heat treatment history. Transition impact strength for both grades was measured using both the Charpy V-notch and the dynamic tear techniques for 5/8" thick specimens [6], and also using an SRI double pendulum machine. Fatigue crack growth rates were measured in accordance with ASTM standards [7] using a closed loop hydraulic testing machine system (MTS-810). A 1T compact tension specimen failed to provide a valid K_{Ic} test for either grade at room temperature.

Results: A summary of test parameters for couplers and yokes is presented in Table 1. The same testing parameters also applied to the testing of the twenty knuckles that failed.

	COUPLERS				YOKES	
No.	A-01	A-02	A-03	*A-04	B-05	B-06
AAR Cast Steel Grade	C(N&T)	C(Q&T)	C(N&T)	E	C(N&T)	C(Q&T)
Load (kip)	± 200	± 200	± 200	$\pm 200/\pm 300$	± 200	± 300
Cycles (10^3)	226.5	111	66.5	404/30	288.5	19
Estimated K_f	1.3-1.5	1.3-1.5	1.5	1.3-1.5	1.5	1.5
Fatigue Characteristics	2 origins at corner and surface, 80% of section.	3 origins at corner surfaces, 1x0.5 inches size.	2 origins at corner surfaces, 1.5 inches and 0.5 inch, long.	2 origins at corner surfaces, 0.5x0.5 inches size.	Multiple initiation joined to form a huge crack-5 inches long (80% of section).	Multiple initiation joined to form shallow crack, 1.5 inches wide.
Fracture Appearance	Brittle.	Rough texture with large shear lips.	Brittle, chevrons pointing to origin.	Rough texture with evident shear lips.	Brittle.	Rough texture.

*Engine coupler, loaded at ± 200 kip for 404,000 cycles, followed by 30,000 cycles at ± 300 kip to failure.

Table 1: Couplers and yokes, test parameters and fracture visual inspection.

An attempt was made to identify the locations of fracture origins, and it was observed that most of the fractures started at high stress concentration areas, as is schematically shown in figure 1 and demonstrated in figure 2.

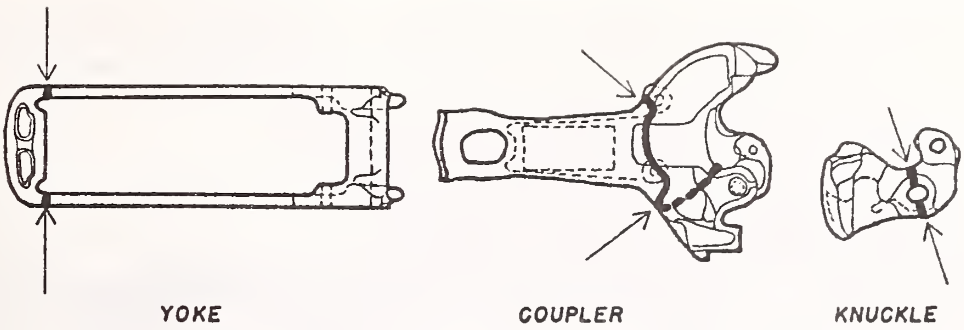


Figure 1: Location of fracture in couplers (A-03), yokes (B-05) and knuckles (D-18).

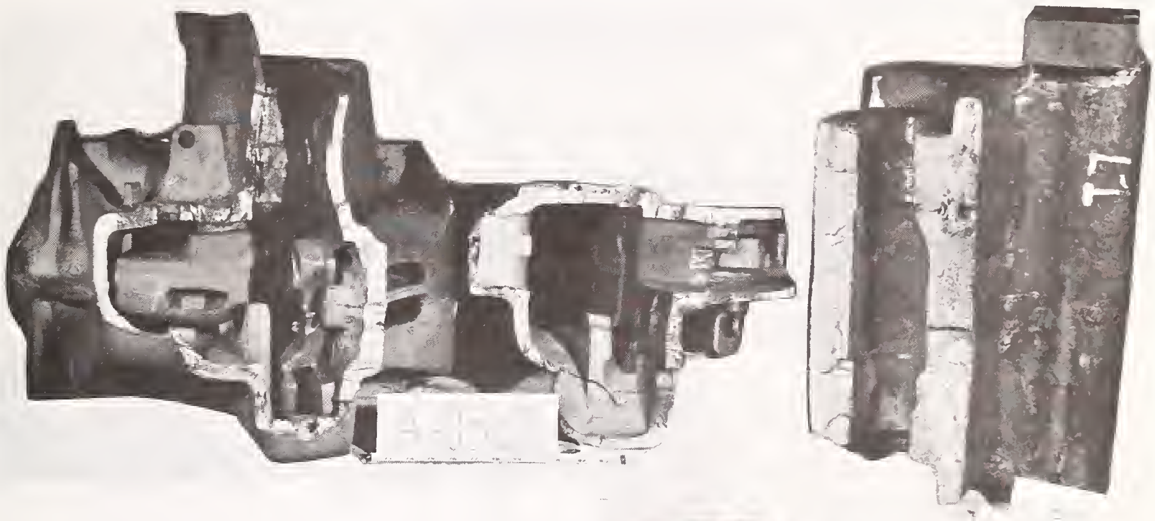


Figure 2: An overall view of representative failed components, in Reference to Figure 1.

As indicated, most of the failed components showed some amount of fatigue crack growth. Huge fatigue cracks, as large as 80% of the cross section, can be seen in figure 3, whereas shallow, multiple-origin cracks are shown in figure 4.

Figure 3: Large fatigue cracks in failed coupler (A-01) and yoke (B-05).

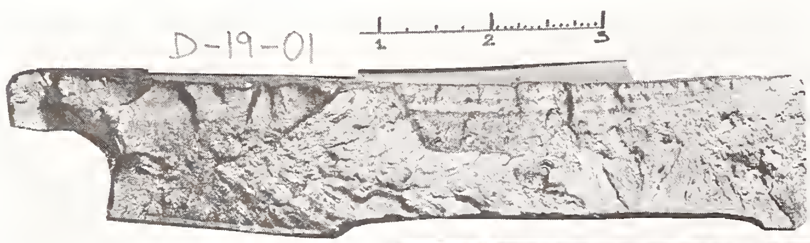


Figure 4: Multiple origin shallow fatigue cracks in failed yoke (B-06) and knuckle (D-19).

In 90% of the failed components, the cast surface roughness, estimated to have K_f values of about 1.5, was responsible for crack initiation sites. Two knuckle failures were associated with huge casting voids, as shown in figure 5. One of these failures (C-18) exhibited a large fatigue crack with chevron marks pointing very clearly to the fracture origin.

Chemical analyses at random locations show that all the components were within the limits of the AAR specifications, as shown in Table 2.

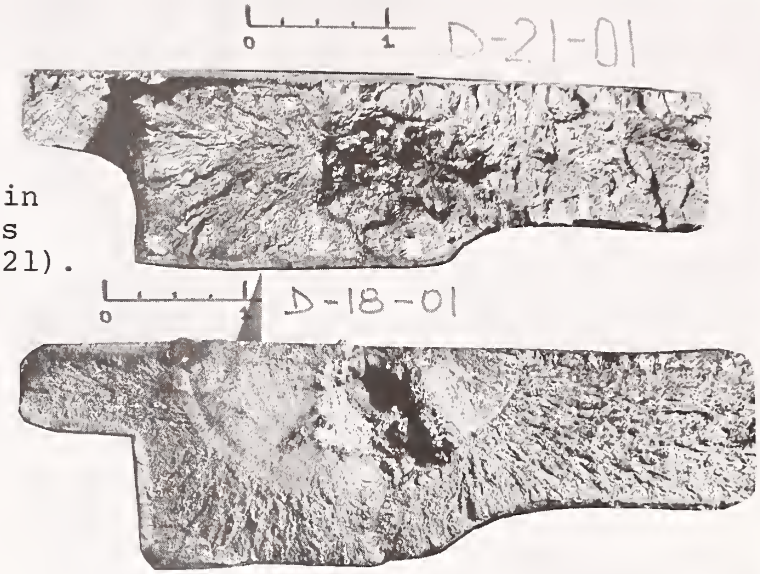


Figure 5: Large casting voids in failed knuckles (D-18) and (D-21).

	COUPLERS	YOKES	KNUCKLES	AAR-SPEC.
Iron	balance	balance	balance	balance
Carbon	0.24 -0.27	0.27 -0.31	0.18 -0.31	0.30 max.
Manganese	1.09 -1.48	1.08 -1.41	1.26 -1.60	1.85 max.
Phosphorus	0.016-0.017	0.021-0.023	0.012-0.027	0.04 max.
Sulfur	0.030-0.031	0.033-0.038	0.020-0.036	0.04 max.
Silicon	0.37 -0.40	0.40 -0.49	0.35 -0.46	0.55 max.

Table 2: Chemical composition of failed components.

Metallographic examination to reveal the microstructure was carried out on all components. Figure 6 shows representative microstructures for both C grades.

The C(N&T) grade, having primarily a pearlitic structure, shows slightly different patterns of microstructure from part to part, depending on cooling rates (TTT diagrams), the manufacturer, and more so, the position in the heat treatment load. This variability repeats itself in the CVN data (discussed later). The C(Q&T) grade, having a martensitic structure, did not show any anomalies. Typical porosity and inclusions, as seen in figure 6b, were found in most parts and are

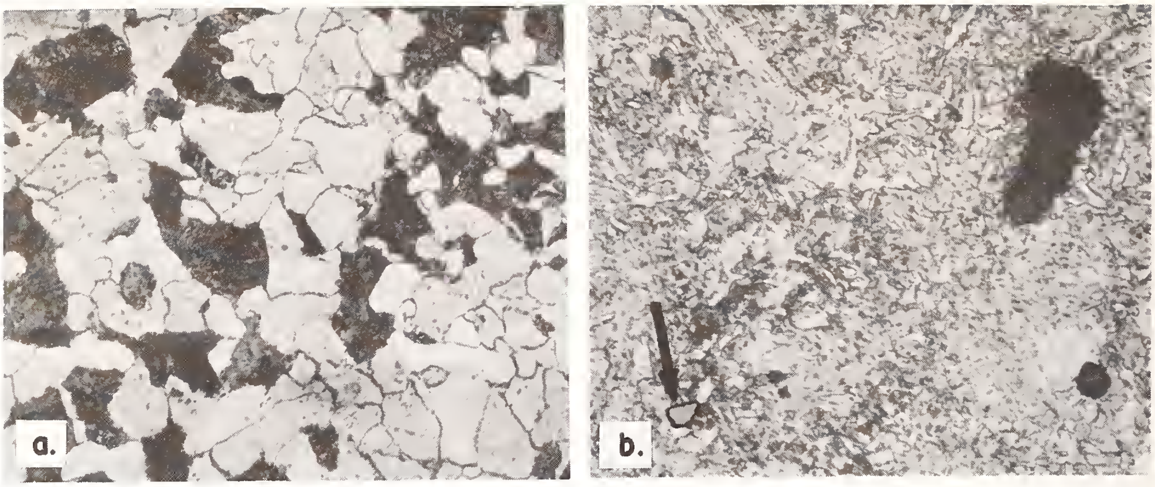


Figure 6: Typical microstructure of: (a) C(N&T) grade, B-05 yoke, and (b) C(Q&T) grade, B-06 yoke, (X500).

regarded as normal. A few parts, as mentioned before, had large, randomly spread cavities, but these cavities did not seem to influence test results. Scanning electron fractography supports the results of the visual inspection (Table 1), and, as seen in figure 7, shows a brittle-cleavage mechanism for the C(N&T) grade and a ductile-shear, microvoid-coalescence mechanism of fracture for the C(Q&T) grade. The particles and inclusions seen in figure 7a were generally rich in phosphorus, nickel and manganese, as determined by the SEM energy dispersive X-ray technique.

A small shrinkage cavity within the fracture path is shown in figure 8. These cavities ranged in size from 10 mils to over 0.5 inch and show a three dimensional dendritic solidifying pattern (figure 9).

Four knuckles, all C(N&T) grade, and one C(Q&T) yoke provided the samples for the Charpy V-notch impact tests to determine the transition behavior (figure 10). It was mentioned earlier that variations in C(N&T) grade microstructures were observed, probably due to differing cooling rates. The wide band of energies for lower and upper shelves for this grade is probably attributable to these variations. However, the range of transition temperatures seems to stay fairly constant, in the range of 50-75^{o}F.

The C(Q&T) grade is very much superior in terms of upper shelf, as well as transition temperature which is in the range of -100 to -50^{o}F. The percentage shear as a function of temperature is shown in figure 11 for the C(Q&T) grade.

One coupler, C(N&T) grade, and one yoke, C(Q&T) grade, provided samples for the dynamic tear tests. The dynamic tear test data, shown in figure

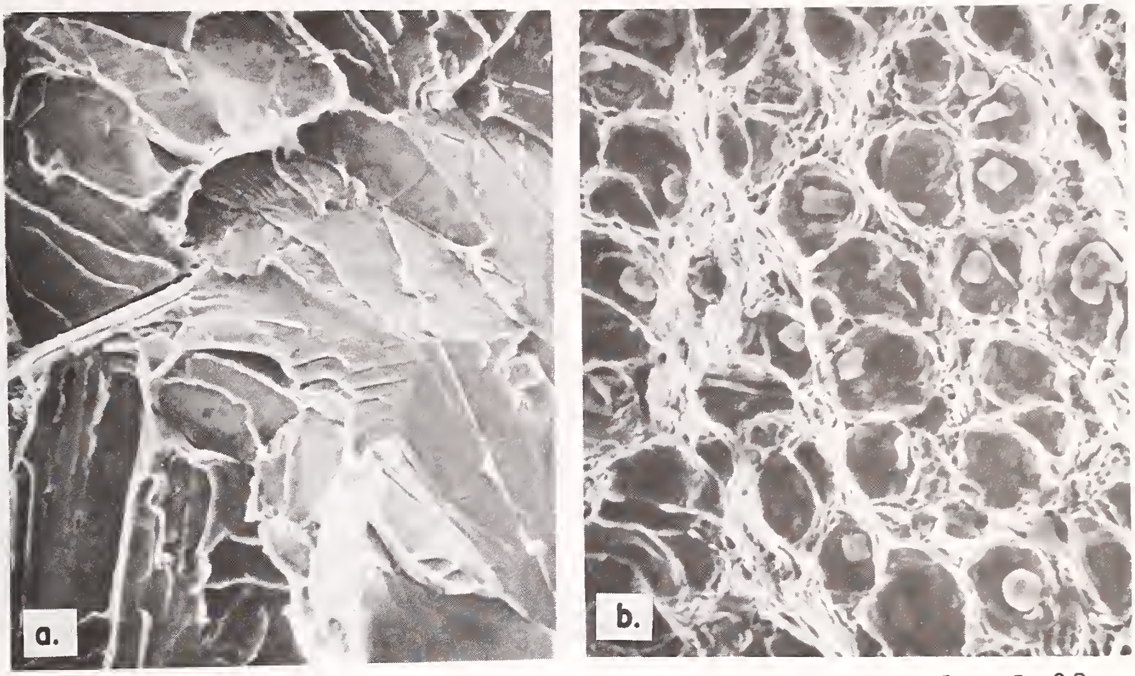
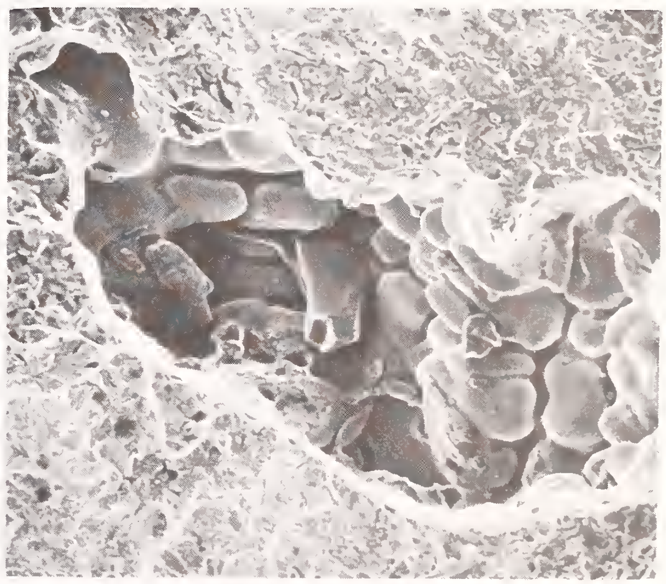


Figure 7: Typical fractography of: (a) C(N&T) grade, A-03 coupler, and (b) C(Q&T) grade, B-06 yoke, (X500).

Figure 8: Shrinkage cavity on a fracture path, C(Q&T) grade, B-06 yoke (X100).



12, support the CVN data and result in values of N.D.T. = +50^{0}F for the C(N&T) grade and N.D.T. = -100^{0}F for the C(Q&T) grade. Figure 13 shows the percentage shear for the DT C(Q&T) grade fractures.

Figure 9: A three dimensional dendrites pattern, in a shrinkage cavity, C(N&T) grade, B-05 yoke (X150).

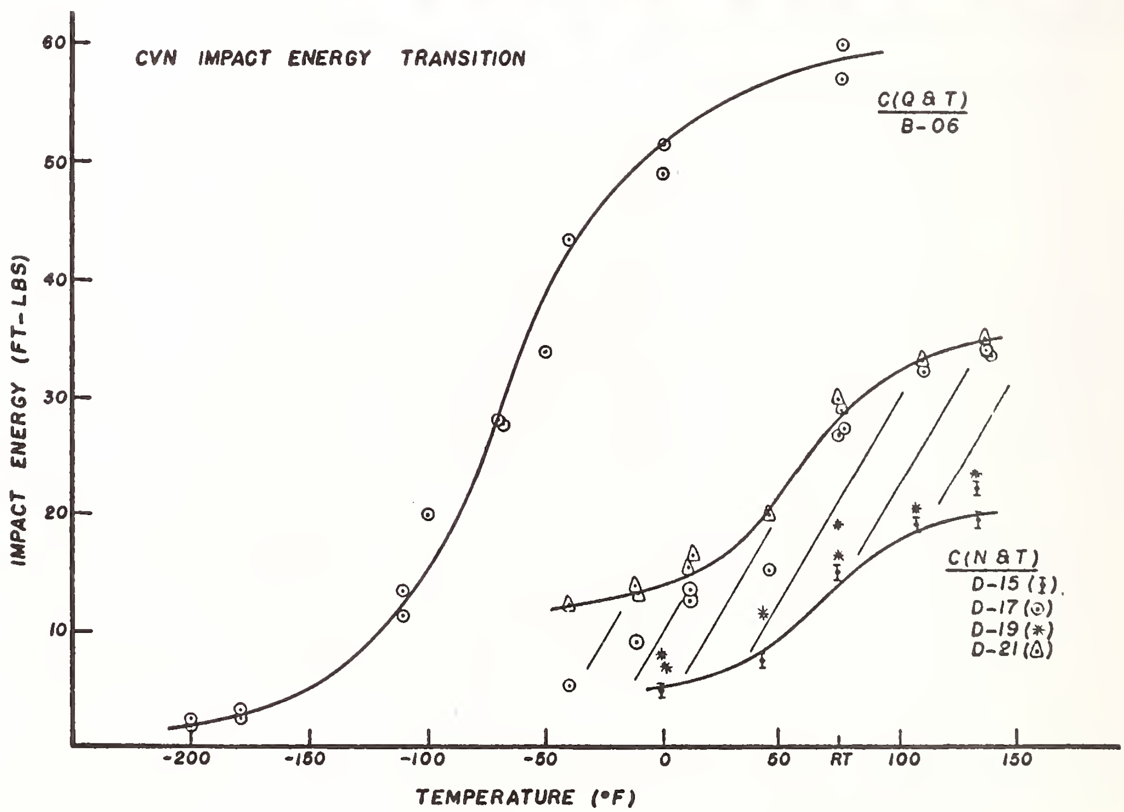


Figure 10: CVN transition impact energy for both grades.

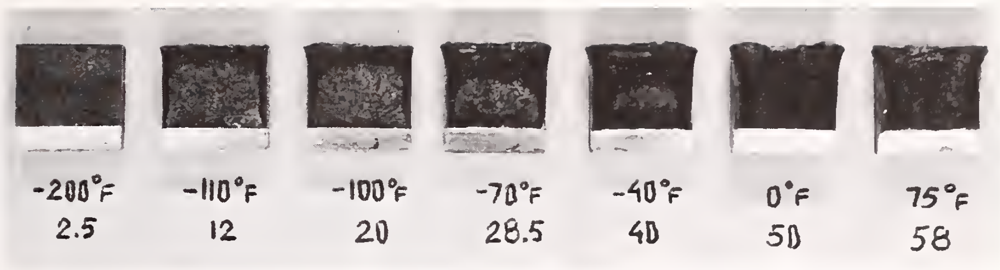
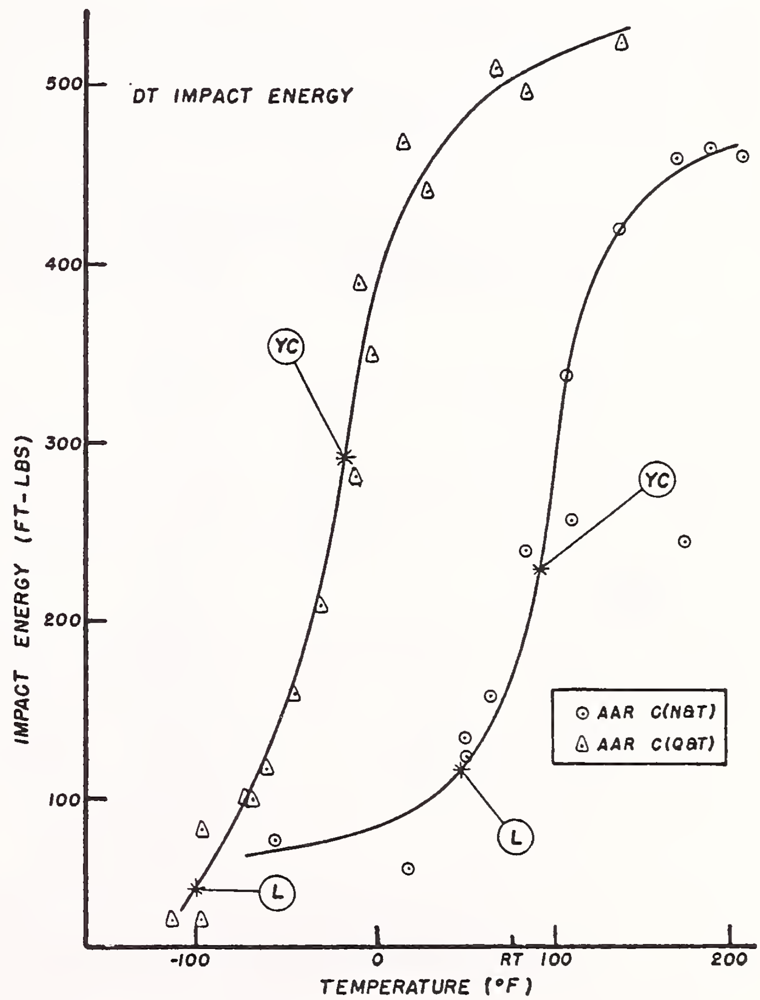


Figure 11: Percentage shear of C(Q&T), B-06 yoke in the CVN test. Temperature and energy (ft-lbs.) are listed.

Figure 12: DT transition impact energy for both grades.



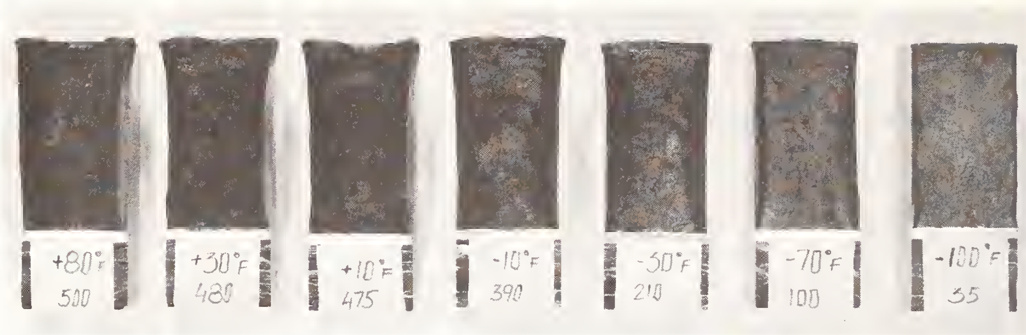


Figure 13: Percentage shear of C(Q&T), B-06 yoke in the DT test. Temperature and energy (ft-lbs.) are listed.

Compact tension specimens were tested to determine fatigue crack growth rates. Data from these tests are shown in figure 14. As expected, no significant differences arise between the two grades. In figure 15, which shows CT fatigue specimens, cavities and porosity can be seen in

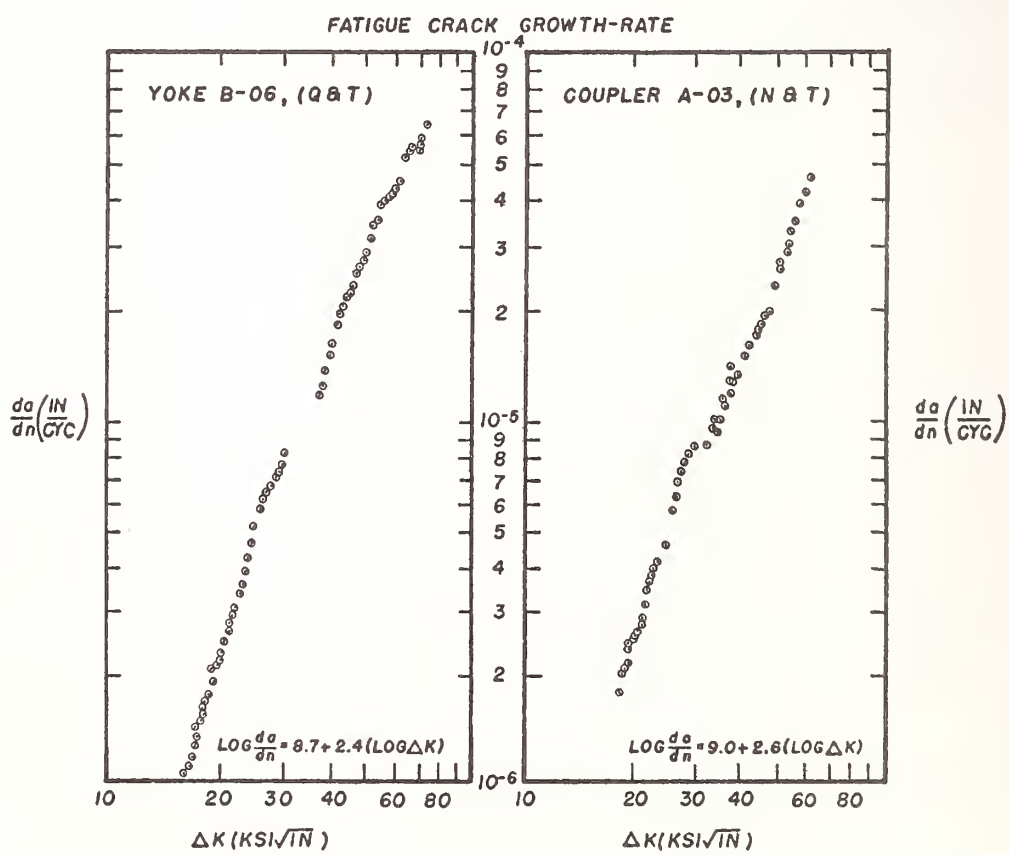


Figure 14: Fatigue crack growth rates for both grades resulting in almost identical rate equations.

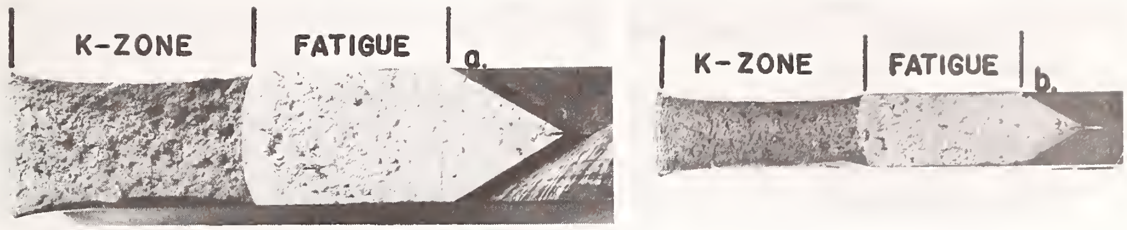


Figure 15: CT specimens fracture surfaces of: (a) C(N&T) grade (A-03) coupler, and (b) C(Q&T) grade (B-06) yoke.

the fatigue path, and ductile tear associated with large shear lips at the K-zone explains why we could not get valid linear elastic fracture toughness data, K_{Ic}.

Summary and Conclusions: The extensive work done in the metallurgical evaluation did not appear to reveal any unusual findings, but rather the very explicit and well known behavior of the AAR C-grades of cast steel. It should be emphasized that the statistics of the C(Q&T) grade within this program are poor and it is really hard to establish a conclusive summary based on two components. However, as mentioned earlier, the results seem to agree very well with what is already known and can be summarized as follows:

1. All test materials appear to be of conventional and accepted quality for production castings.
2. Cast surface roughness with estimated K_f values of 1.5 seem to be unavoidable and are regarded as normal. This roughness is probably responsible for most crack initiation sites.
3. Voids and cavities in the material's bulk do not seem to influence component performance.
4. Most of the failed parts exhibited a certain amount of fatigue crack growth, although extremely scattered in terms of number of cycles to failure. The ASTM fatigue crack growth test reveals identical rates for both grades (see figure 14). It is therefore really hard to understand the wide variety of cycles to failure. A few factors that may have some effect are suggested: (a) it is impossible to estimate the initial K values and the shape of the initiation site defects and therefore differences will result in the number of cycles for the defect to start growing as a fatigue crack, (b) the differences in microstructure of the different components, as well as the related scattering of the CVN transition curves for the C(N&T) grade present probably the most solid explanation for these variations, (c) loading rates calculated from test frequencies and peak loads show a range of 8 to 40 KIP/min. Being in the intermediate range of loading rates [8], a factor of five might turn out to be significant, and (d) there are uncertainties in regard to load

history, primarily with possible overshooting on starting or stopping as well as retardation effects due to long breaks within a test.

5. Applying Pellini's approach for fracture transition of steels [9], and using the DT data presented earlier, the following values are developed: for C(N&T) grade, L=50^{o}F and YC=90^{o}F, and for the C(Q&T) grade, L=-100^{o}F and YC=-20^{o}F. These values are in good agreement with what is known and applicable for these grades of cast steels [9,10].
6. Laboratory experience and service experience appear very similar. Fatigue crack sizes at failure appear normal for yokes, scattered for knuckles, and somewhat larger for couplers, probably due to the absence of occasional impact loading during the tests.
7. Fracture appearance of the C(N&T) grade is typical of brittle fracture as expected and as found in service failures [2,3]. The appearance of the C(Q&T) grade is also typical for ductile fracture, as expected at room temperature. There is not enough service analysis documentation to support the above observation.
8. It is recommended that extensive fracture mechanics characterization of these two grades of steel, as well as the other two grades that are in use - the B normalized and the E quenched and tempered, be performed for a better understanding of their performance.

Acknowledgement: Gratitude is expressed to Dr. S. Mostovoy of IIT, Chicago, Illinois for time and support and very beneficial discussions, to B. E. Largent for the photographic work, to M. C. Fec for graphics, and to B. Vaughan for typing the manuscript.

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SESSION II

RAIL VEHICLES AND STRUCTURES

Chairman: C. G. Interrante, National Bureau of Standards

REQUIREMENTS FOR ON-BOARD FAILURE
DETECTION SYSTEMS FOR RAIL-VEHICLES

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Abstract: On-board failure detection systems for rail vehicles are proposed as a means of reducing train accidents particularly in the high speed range. In this paper, accidents are broken down in broad categories of cause and speed range. Special emphasis is given to equipment caused accidents as a percentage of total accidents for all speeds and then for 3 speed ranges--10 mph and below, 11 mph-30 mph, 31 mph and above. When the accidents occurring at speeds of 31 mph and above are isolated, equipment caused accidents represent a larger portion of the accidents than when the analysis includes the lower speeds. Isolating component failure accidents, a similar trend is noted with certain components causing a very small percentage of accidents at lower speeds, but causing an increased percentage of the accidents at higher speeds. These components are discussed in terms of past research with on-board sensors as well as possible candidates for future research and development. Various approaches are discussed to examine methods to use the sensors in a system for possible safety benefits. The advantages and disadvantages of each type of system are discussed.

Key words: Contact derailment sensor; g-sensing derailment detector; local derailment; nitinol sensor; on-board failure detection system; overheated bearings; thermal switch sensor; train line.

INTRODUCTION

Before evaluating on-board failure detection systems for rail vehicles, the problem must be initially defined in terms of the accidents which could possibly be prevented. This paper, therefore, will first discuss railroad accidents as reported to the FRA for the last 10 years, and then examine various on-board failure detection systems for rail vehicles for their advantages and weaknesses.

ACCIDENT DATA ANALYSIS

The total number of accidents reported to the FRA during the period of 1968 to 1978 is shown in Figure 1. These accidents represent all those involving a rail vehicle which incurred damages at or above a monetary reporting threshold.* Although yearly fluctuations are present, an overall rise in accident numbers can be seen between 1968 and 1974. While in 1975, it appears that the number of accidents dropped, it must be noted that a change in reporting requirements was made in that year and affected the statistics accordingly. The increasing trend resumes between 1975 and 1978 and merits investigation as to the cause.

In 1978, of the total number of reported accidents, 11,277 (77.7%) were derailments; 1,477 (13.1%) were collisions; and 1,037 (9.2%) were other types of accidents. While the following analysis of speed and damage factors will include all of these accident types, it should be kept in mind that the predominant type of accident is the derailment category.

Classification of 1978 accidents according to speed (See Table 1) reveals that approximately two-thirds (67.8%) of all accidents occur at speeds of 10 mi/hr (16 km/hr) or less. This speed range, however, represents only 30.5% of the dollar damage. Conversely, the speed range 30 mph and above accounts for only 10.1% of the accidents, but 32.3% of the dollar damage. The relationship between damage incurred and speed is the result one would expect and is further shown in Figure 2 which shows damage per accident as a function of speed.

Train accidents can also be classified according to their causes. For the year 1978, accidents broke down into the following categories and percentages:

Track caused	42.5%
Equipment caused	19.3%
Operating practices	25.2%
Other	13.0%

Table 2 further delineates this information into speed ranges versus cause. In analyzing this data, it is noted

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*For the years 1968-1974, the reporting threshold was $750; in 1975-1976, it was increased to $1,750; and for the years 1977-1978, it rose again to $2,300.

REPORTABLE ACCIDENTS

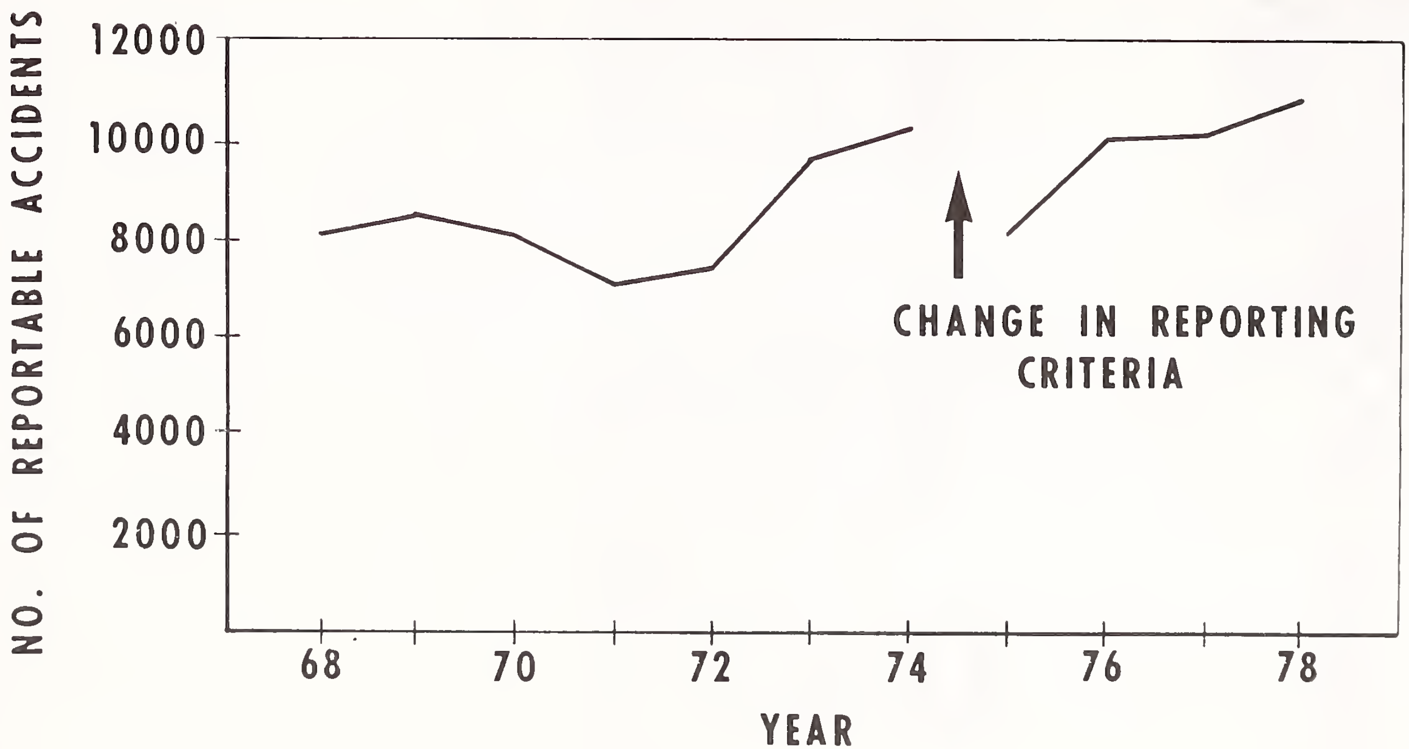

Fig 1

ACCIDENTS-DAMAGE BY SPEED RANGE

SPEED (MPH)	% OF ACCIDENTS	% OF DAMAGE
0-10	67.8%	30.5%
11-30	20.3%	36.1%
31 & ABOVE	10.1%	32.3%

Table # 1

ACCIDENT SPEED - $ DAMAGE

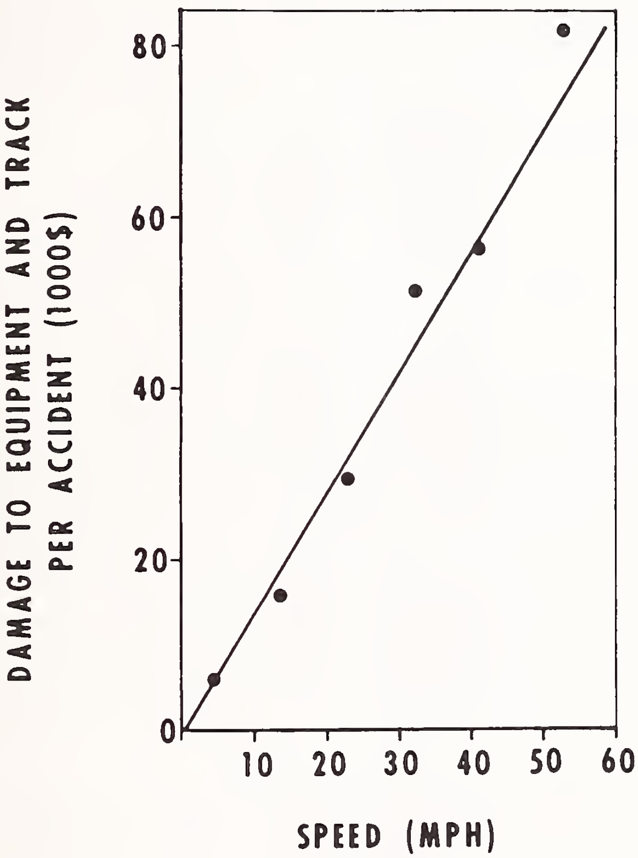

Fig # 2

TRAIN ACCIDENTS/CAUSE 1978

SPEED RANGE (MPH)	TOTAL	TRACK CAUSED	EQUIPMENT CAUSED	OPERATING PRACTICES	OTHER
0-10	7655	48.2%	12.4%	30.6%	8.8%

Table # 2

that while track causes represent 48.2% of the low speed accidents, they represent only 20.5% of the high speed accidents. Inversely, while equipment causes result in only 12.4% of the low speed accidents, in the high speed accident range, equipment causes result in 48.0% of the accidents. As shown earlier, the higher speed accidents result in the most amount of damage. It can be asserted, therefore, that equipment caused accidents represented nearly one-half of the most serious train accidents in 1978.

Causes relating to component failures are further delineated for more specific examination using standard FRA cause codes such as:

- Frogs, switches & track appliances
- Axle bearings
- Rails and joint bars
- Wheels
- Couplers and draft system
- Brakes
- Side bearings

Based on train accident data for the years 1977 and 1978, the leading causes of accidents due to equipment failures were identified. Tables 3, 4 and 5 rank the equipment failures which caused the most accidents for each of the speed ranges. It is interesting to note that the primary cause of accidents in one speed range may not show up at all as a major cause at another speed range. For example, axle bearing failure is the leading cause of train accidents in the high speed range of 30 mi/hr and above. However, this classification of cause does not appear as a major cause in the low speed range of 10 mi/hr and below.

It is apparent that the leading causes of accidents at the most serious, higher speed ranges are vehicle related and these are the prime candidates for failure detection systems.

DETECTION SYSTEMS

One possible method of providing for prevention of these equipment failures is through the use of on-board failure detection systems. On-board detection systems could, by early detection of impending equipment malfunction, prevent a large number of the serious accidents which occur on railroads.

Most past research concerning use of on-board sensor systems has involved detection of overheated bearings (hot boxes)

COMPONENT FAILURES
(SPEED RANGE 0-10 MPH)
1977-1978

1. FROGS, SWITCHES & TRACK APPLIANCES	2067	(33.4%)
2. RAILS AND JOINT BARS	2032	(32.8%)
3. WHEELS	472	(7.6%)
4. COUPLERS AND DRAFT SYSTEM	329	(5.3%)
5. BRAKES	295	(4.8%)
6. SIDE BEARINGS	269	(4.4%)
ALL OTHERS	725	(11.7%)
TOTAL	6189	

Table # 3

COMPONENT FAILURE
(SPEED RANGE 11-30 MPH)
1977-1978

1. RAIL AND JOINT BARS		553	(28.6%)
2. AXLE BEARINGS		231	(12.0%)
PLAIN JOURNALS	174		
ROLLER	57		
3. SIDE BEARINGS		222	(11.5%)
4. WHEELS		173	(9.0%)
5. COUPLER AND DRAFT SYSTEM		163	(8.4%)
6. BRAKES		151	(7.8%)
ALL OTHERS		438	(22.7%)
TOTAL		1931	

Table # 4

COMPONENT FAILURE
(SPEED RANGE 31 MPH & ABOVE)
1977-1978

1. AXLE BEARINGS		241 (19.2%)
PLAIN JOURNAL	173	
ROLLER	64	
2. WHEELS		199 (15.8%)
3. RAILS AND JOINT BARS		164 (13.1%)
4. COUPLERS AND DRAFT SYSTEMS		162 (12.9%)
ALL OTHERS		491 (39.0%)
TOTAL		1257

Table # 5

and the occurrence of local derailments.* On-board sensors can be developed much further, however, for other train applications, such as monitoring of:

- overheated or cracked wheels
- excessive rock and roll
- hazardous material leakage
- potential damage to special cargo
- unreleased hand brakes.

In general, the complete on-board detection system is made up of three segments: (1) the on-board sensor itself; (2) a method for transmitting the signal emitted by the sensor and (3) a correcting response for avoiding an impending accident.

On-Board Detection Sensors

To date, research has been conducted on on-board sensors which are designed to detect hot boxes on freight cars. Figure 3 illustrates one particular type of a hot box sensor which utilizes a wire made of nitinol, a unique metal with the property of changing shape when a certain temperature is reached. In this case, the nitinol ring will spread when the metal reaches 250° F. As the ring spreads, a firing pin is released which in turn activates a thermal battery and a signal is emitted.

A second type of overheated bearing sensor involves the use of a thermal switch (See Figure 4). When the switch reaches a given temperature, the system is activated and sends a radio signal to the engineer. The unique aspect of this sensor is that it is in the form of a bolt and replaces one of the actual bolts which secures the bearing end cap.

In terms of detecting local derailments and taking corrective action to prevent a major derailment, the following two sensors have been developed. One such device is a g-sensing derailment detector (See Figure 5). This truck mounted device senses "g" levels in a vertical direction and when the seismic mass is sufficiently displaced upon

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*A local derailment is defined as a derailment of a single wheelset.

OVERHEATED BEARING SENSOR
(NITINOL TYPE)

SEQUENCE OF OPERATION

1. HOT BEARING DEVELOPS
2. NITINOL SENSES 250°F
3. FIRING PIN IS RELEASED
4. STAB PRIMER ACTUATES THERMAL BATTERY
5. BATTERY SUPPLIES 6.2 VOLTS TO BRAKE ACTUATOR

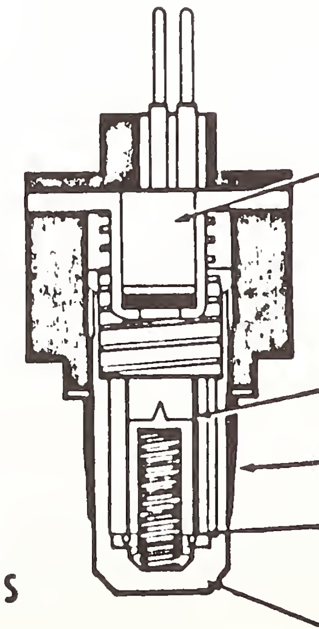

Fig # 3

OVERHEATED BEARING SENSOR
(THERMAL SWITCH TYPE WITH TRANSMITTER)

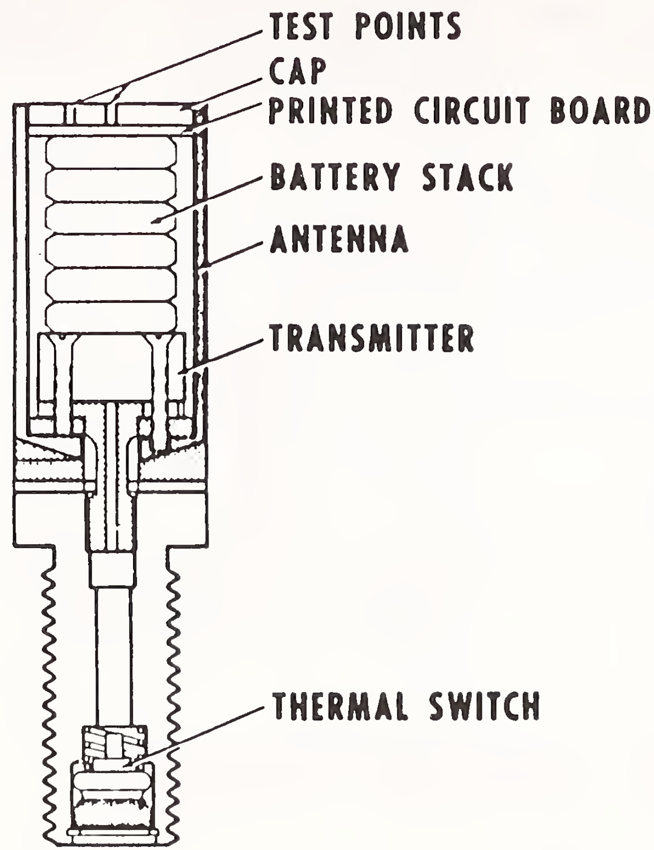

Fig # 4

G - SENSING DERAILMENT DETECTOR

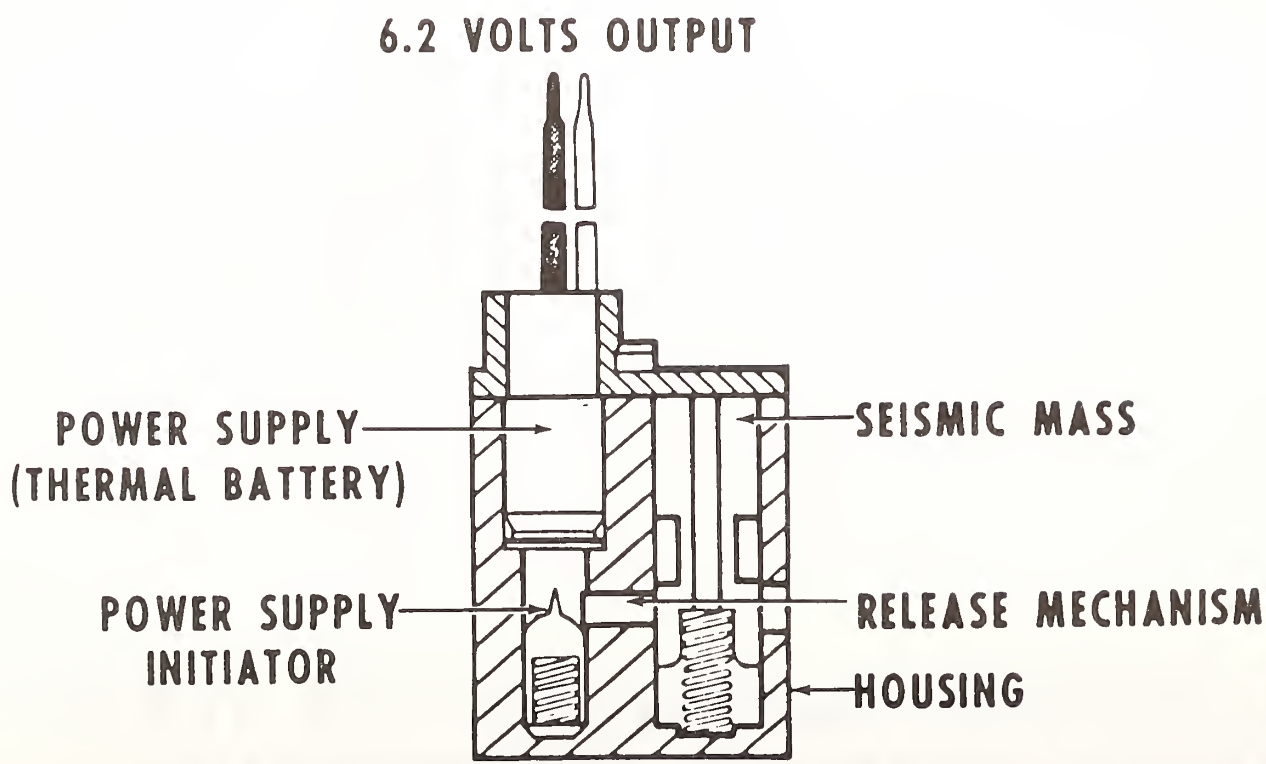

Fig # 5

derailing, a release mechanism allows the signal power supply to be initiated. During the testing of this device, it was found that normal "g" levels encountered over rough track in the winter could be higher than that caused by an actual low speed derailment in the summer. This would mean that the sensors would be required to be adjusted frequently to accomodate environmental changes or it would result in false alarms.

In an effort to avoid the disadvantage associated with the seismic sensor, a second type of local derailment sensor has been investigated. This sensor, shown in Figure 6, is based on contact with the rail upon derailment. With this sensor, a disc on the bottom of the sensor attached to the wheel opposite the derailed wheel will make contact with the rail. The disc is attached to a plunger type of mechanism which severs a shear pin and activates a pulse battery which emits a signal.

Signal Transmission

There are basically two ways in which the sensor's signal may be transmitted to the crew; by radio or by a train line. There exist drawbacks to both transmission methods, however, as discussed below.

First of all, there is the problem that the locomotive heading the train consist must have the proper receiver to match the emitter on the particular freight car. In view of interchange requirements, it is not practical to key a freight car to a specific locomotive. Moreover, all freight cars and locomotives would be required to operate on the same radio frequency. This may cause problems if the signal is not sufficiently directional. When trains are in close proximity to one another, for example, as in a yard operation, the wrong locomotive may receive the signal.

Contemplation of a train line transmission system must start with the understanding that train lines, at present, do not exist on freight cars. An extensive modification program to install the on-board system or train line alone, therefore, would be required to implement such a plan. Moreover, should the line on one freight car malfunction, the entire system would go down. This alone may dictate that should the transmission of the signal to the crew be desired, radio transmission would be the more practical approach.

CONTACT DERAILMENT SENSOR

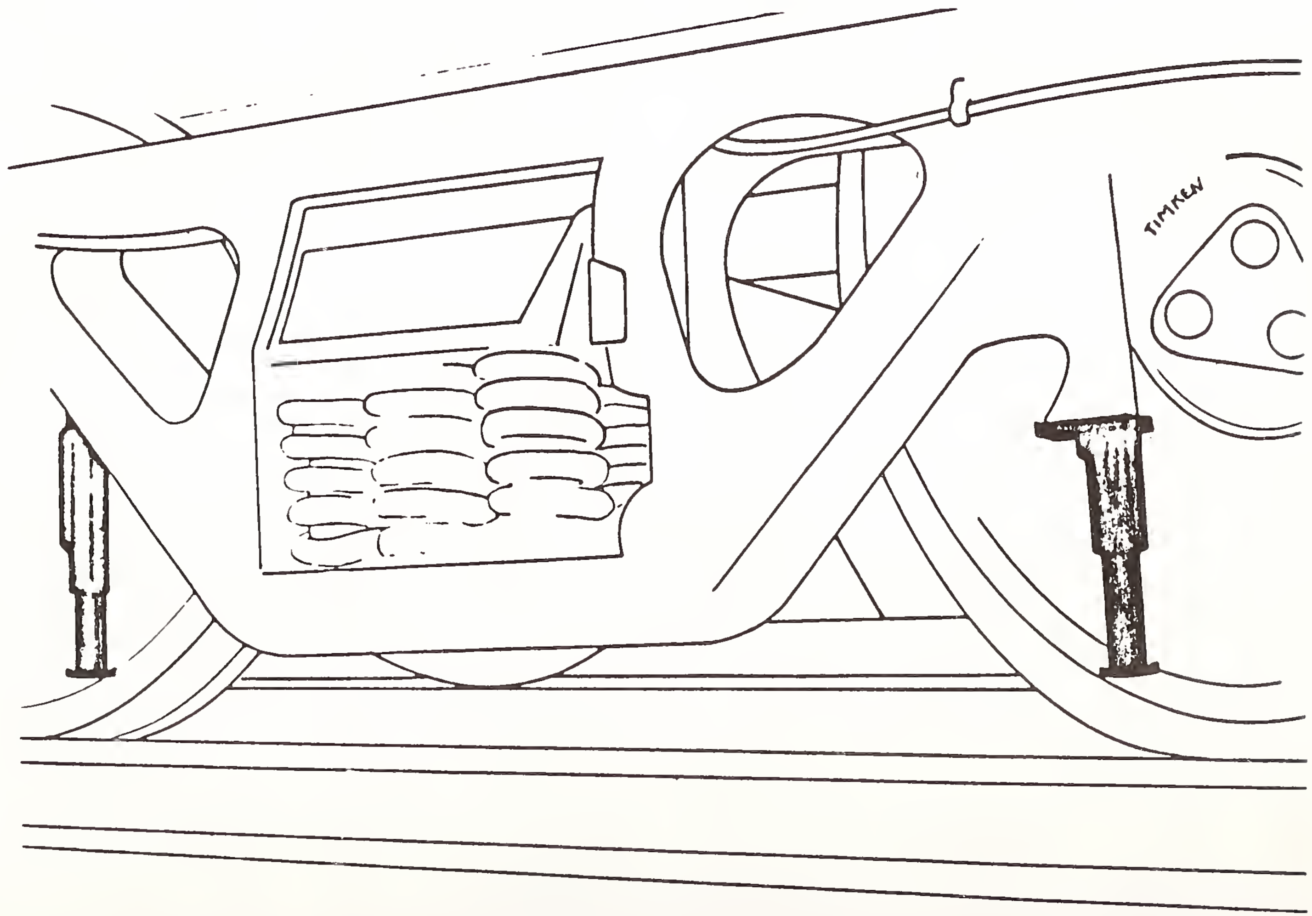

Fig # 6

System Corrective Response

Once a problem arises and the sensor is activated, there are two options as to what corrective response action can take place. One type of system response has already been discussed in that the signal is transmitted to the train crew in the locomotive and the crew will then take appropriate measures to correct the problem. A second method of response entails automatic activation of the brakes, to be accomplished with a device such as that shown in Figure 7. This device, when activated by an electric signal, vents the brakeline at a predetermined rate. The overall automatic system is shown in Figure 8.

At present there is no clear preferred approach as to corrective response action, as both present their own advantages and disadvantages. The automatic system requires no decision and brake action is immediate. There are arguments, however, that there are situations in which immediate brake action is inappropriate and may actually result in a more severe derailment. There also exists opposition to any modification to the brake pipe as required in the automatic system. On the other hand, with an automatic system, a specially equipped locomotive is not required, thereby facilitating portions of one fleet to function safely when used on other railroads. As in the case of radio transmission, the automatic system's benefits are proportional to the number of cars equipped.

In support of transmission of the signal to the crew, there exists the capability to transmit noncritical information which does not require immediate action. For example, information such as detection of rail flaw could be transmitted to the engineer who could then relay that data to the dispatcher. An automatic system, being limited to a predetermined brake application, could not serve this purpose.

CONCLUSION

As indicated above, each type of system has its own advantages. Since each system, if properly developed, could accomplish the safety function, further investigations must necessarily consider economics. At this time, adequate information is not available on cost, potential benefits or reliability of on-board failure detection systems. Further research and evaluation will be required to provide this information.

BRAKE CONTROL

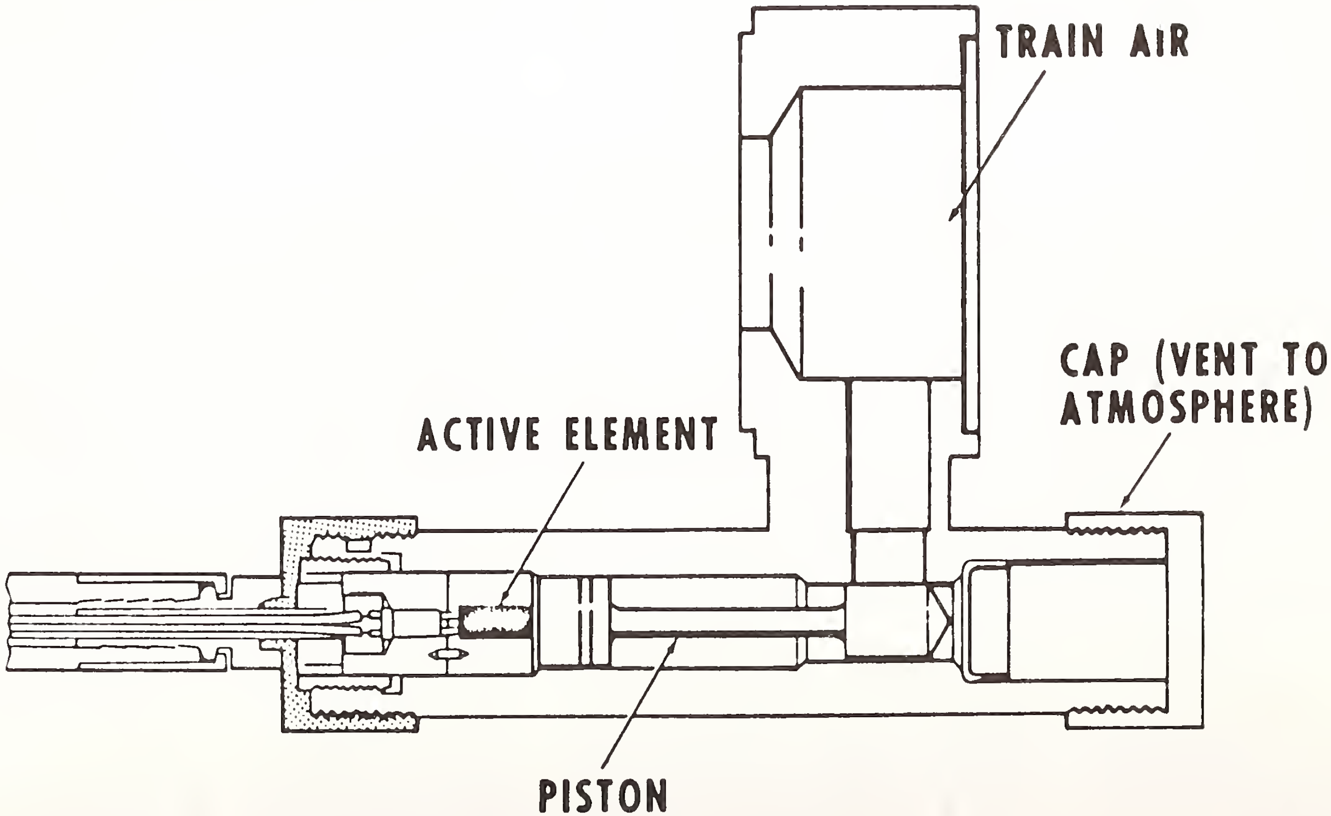

Fig # 7

AUTOMATIC SYSTEM

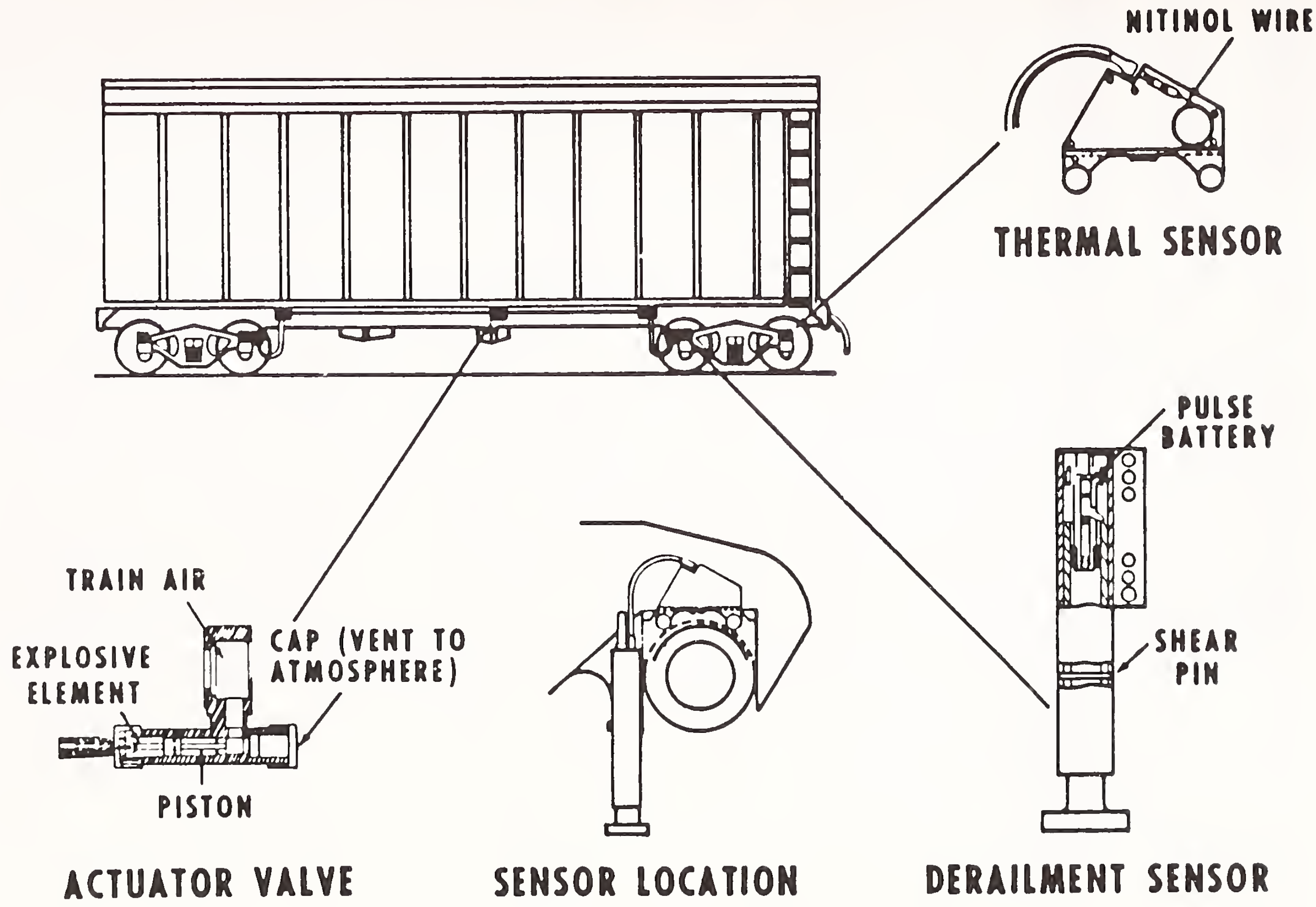

Fig # 8

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DETECTION OF RAIL DEFECTS AND PREVENTION OF RAIL FRACTURE

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Abstract: Rail transportation safety is maintained in part by periodically conducting a continuous search of track to find and correct rail defects. The state of the art of this detection technology is reviewed to illustrate, in general terms, the capabilities available to the railroads. A joint government/industry effort is now in progress to determine whether the periodic inspection strategy can be improved by means of an adaptive control algorithm that permits variation of the inspection interval based on actual histories of detected defects. The results of some preliminary studies are presented to illustrate how simulation techniques can be used to assess the effectiveness of proposed changes of inspection strategy.

Key words: Crack detection; inspection interval; rail flaw detection.

BACKGROUND

About five hundred derailments occur annually in the United States as a result of rail fracture. Although many of these derailments have only economic consequences, public and/or train crew safety can be affected by occasional accidents involving high-speed operations and/or hazardous material transport. The government and the railroad industry thus have parallel interests in the goal of reducing derailments caused by rail fracture.

Since the complex behavior of track makes it impossible to predict the times and locations of fractures, control is established primarily by means of periodic inspection with detector cars specifically designed to find rail defects while they are still growing slowly in a subcritical state. The railroads and rail inspection services, as well as the government, have actively engaged in research to improve subcritical flaw detection capability. The flaw detection capabilities available to the railroads are outlined in the following section. It will be seen that that technology is imperfect, viz: detection cannot be guaranteed for large defects approaching fracture, even though there is some chance of detecting small ones which are still in the early stages of growth; also, the ability to accurately classify defects by size is limited.

In the railroad environment, these imperfections in flaw detection can lead to an overcommitment of scarce maintenance resources in order to repair or remove small defects. Since rail defects generally appear to grow very slowly, it would seem advisable to concentrate maintenance on the larger defects approaching fracture, if an effective strategy could be found to achieve this goal.

The current Federal regulations require a minimum of one detector-car inspection annually for certain classes of track. However, many railroads inspect certain sections of track more often, and all railroads back up the detector-car search with much more frequent visual inspections. This existing approach to control rail defects will be compared to an adaptive control scheme which is currently the subject of a joint government/industry study. The study objective is to determine whether it is possible to rebalance resource commitments by varying the inspection interval based on actual counts of detected defects.

The effectiveness of adjusting inspection frequency based on defect count can be assessed by means of a model which simulates the dynamics of a population of defects that have been generated by track usage, grow under further usage, and are subjected to periodic detection. Results from some preliminary studies with a simplified population dynamics model will be presented to show that the adaptive strategy is a promising approach to control of rail defects. This paper concludes with some remarks about the possible effects of defect spatial distribution on the effectiveness of the adaptive strategy and a summary of plans for future research.

DETECTION VEHICLE TECHNOLOGY

The fleet currently inspecting the nations's railroads consists of 72 rail flaw detection vehicles. Forty-six of these are owned and operated by individual railroads and the remaining 26 by inspection services. The inspection services provide service for railroads who do not own their own equipment, and supplement the inspection capabilities for railroads who have inspection vehicles. Only one railroad currently has the capability to satisfy its inspection needs without help from an outside service.

During 1979, the fleet logged over 300,000 miles of track inspection. This inspection mileage was almost equally divided between the railroad fleet and the inspection service fleet. The total mileage inspected consists of both single (once per year) and multiple inspections. The single, annual inspection complies with Federal Track Safety regulations. Individual railroads use multiple inspections to insure high safety and maintenance standards on lines carrying high tonnage.

The inspection methods used on the vehicles include ultrasonic, magnetic or a combination. Twenty-five of the combination systems are owned and operated by one of the inspection services and nine are owned by five of the railroads.

The overall performance of the systems (basically speed and flaw detectability) varies quite significantly. This variation can be attributed primarily to the large number of different designs and configurations used and the range of signal interpretation skill possessed by different equipment operators. The mode of operation used in the United States is to hand verify all flaw indications sensed by the detection vehicle (Figure 1). There is heavy dependence on human judgement in both operations. The operator must interpret the test data to determine if the system flaw indication warrants further investigation and whether the results of the hand test, using ultrasonic techniques, indicate a rail flaw requiring remedial action.

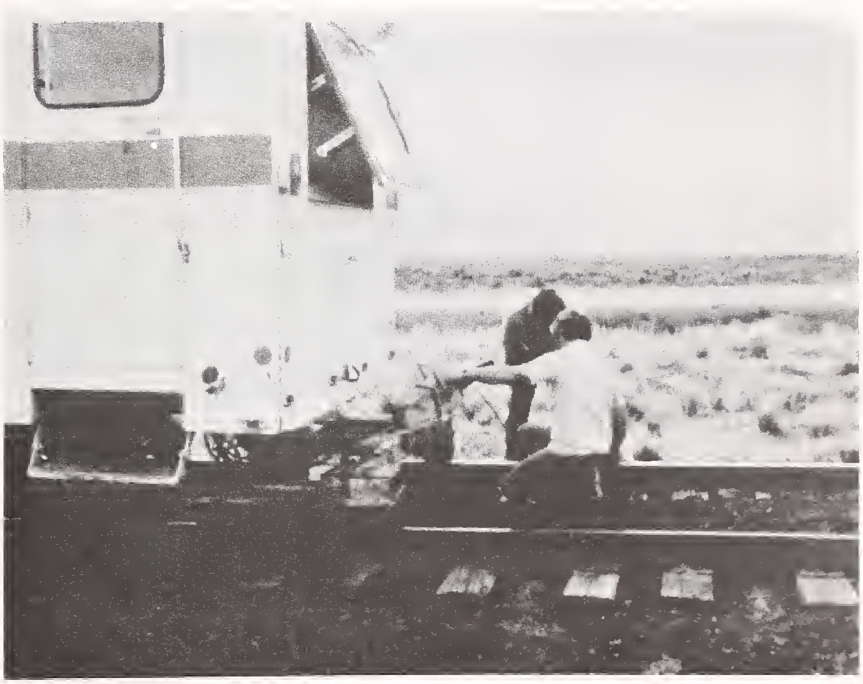


FIGURE 1. HAND MAPPING FOR DEFECT VERIFICATION

Approximately 12 basic designs appear in the national fleet. Various combinations of sensor arrays, data processing and display schemes are used. Sensor array designs vary from three sensors per rail on either ultrasonic or magnetic systems to as many as 15 on combination systems.

Ultrasonic transducers are coupled to the railhead through either sled or wheel carriages, using a variety of fluid couplants.

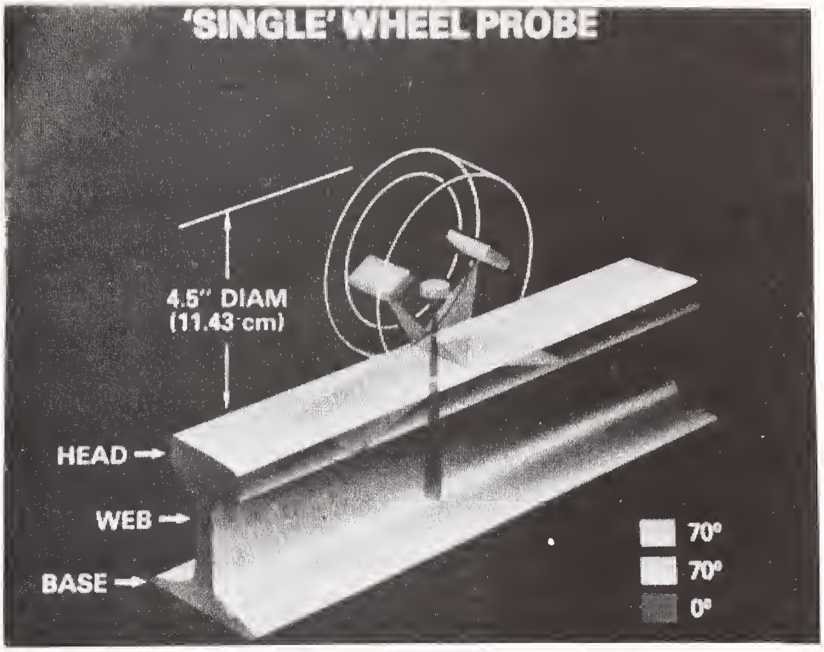
Figure 2a represents the array with the fewest number of transducers used in the industry. Figure 2b shows a tandem wheel configuration with additional transducers oriented to enhance the detection of certain types of defects and to provide coincidental (multiple channel) detection of other defects to improve the signal-to-noise ratio and thus increase the probability of detection. Even with the tandem multiple transducer sensors, defects can be located and/or oriented so as to not reflect or block probing signals, and can thus escape detection. Another major contributing factor which limits the detectability of ultrasonic systems is the problem of coupling acoustic signals into and from the rail. Grease from oilers on curved track, or rail-head surface anomalies can prevent acoustic signals from entering the rail. The combination of the technology limitations and typical operating conditions make today's inspection operations imperfect.

The processed inspection data is presented to the operator in one of two ways: a pictorial B-scan presentation which is found in 22 vehicles of the fleet; or variations of gate/threshold processing coupled to a proper chart. Figure 3 compares the two presentation methods.

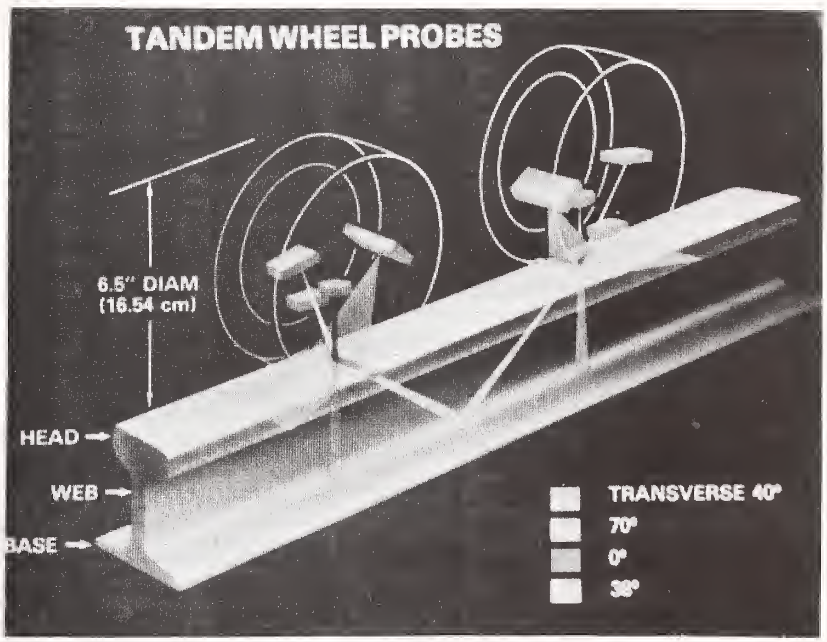
The systems using the B-scan presentation have an advantage over the gate/threshold systems in that more descriptive data is presented to the operator, allowing quicker and better characterization of the flaw. However, the B-scan systems currently in the field have no data screening capability. Consequently, the large volume of data presented to the operator can limit the maximum inspection speed. Maximum inspection speeds of up to 15 mph have been attained, but the fleet average inspection speed ranges from 2 to 6 mph. Inspection speed is heavily dependent on the condition of the rail under inspection and the number of stops required to hand verify flaw indications.

Both hy-rail and rail-bound vehicles (Figure 4) are used in the rail inspection industry to house the inspection equipment. The trend in the railroad fleet has been toward the use of more hy-rail vehicles. This type of vehicle offers more mobility and, as a result, less interference with traffic when operating in an inspection mode or when in transit between inspection sites.

There are 43 hy-rail vehicles and 3 rail-bound cars currently in the railroad fleet. Rail-bound cars are used almost exclusively in the inspection service fleet (25 out of 26). The larger vehicles provide space for the crew to live on-board and to house the heavy magnetic induction equipment used by the service.



(A) SINGLE ULTRASONIC WHEEL PROBE



(B) TANDEM-WHEEL ULTRASONIC PROBES

FIGURE 2. TYPICAL FLAW DETECTION WHEEL SENSOR ARRAYS

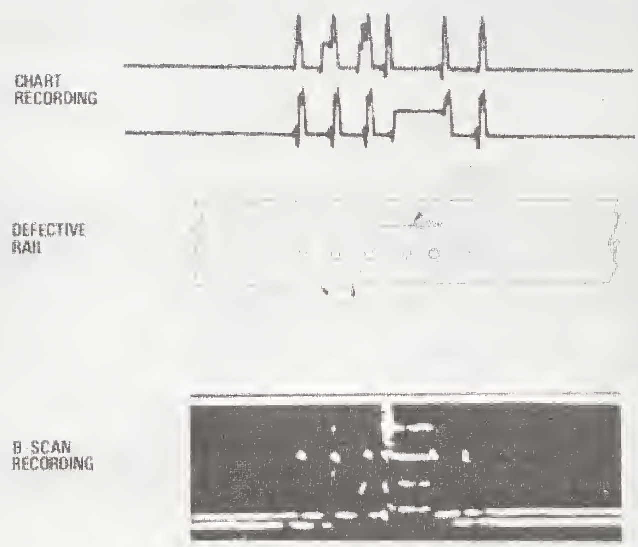
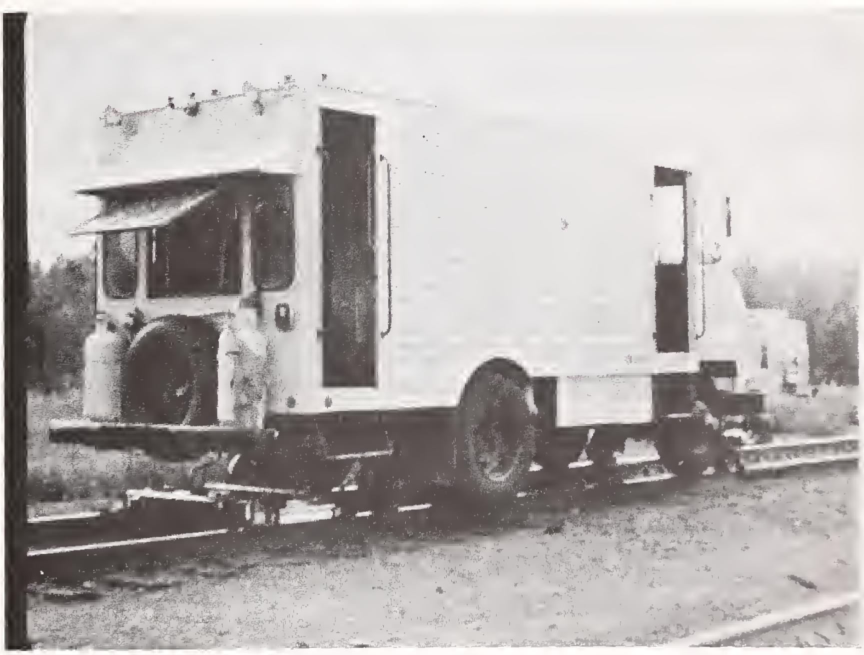


FIGURE 3. COMPARISON OF CHART RECORDING AND B-SCAN PRESENTATIONS

The trend within the railroad fleet over the past 10 years has been to improve system capabilities by increasing the number of sensors. This is being accomplished by adding transducers to the ultrasonic system or using ultrasonic and magnetic systems in combination on the same vehicle. The ultrasonic and magnetic inspection methods nicely complement each other to provide a more thorough inspection than either system operating independently. Magnetic systems have good detectability for defects located in the upper portion of the rail head, particularly near the surface where ultrasonic systems are deficient. Ultrasonic systems are effective in the lower portions of the head, and in the web region where magnetic systems have almost zero detection probability. The major service fleet has used a combination system for the past twenty years; nine combination systems are currently operational in the railroad fleet.

It is apparent that a wide diversity of detail design and configuration exists in the national fleet, although the major inspection service maintains a uniform design throughout their 25-car fleet, and the owner-railroads are working toward the same objective within their own fleets. Since the introduction of ultrasonic inspection systems

(A) RAIL BOUND DETECTOR CAR OWNED AND OPERATED BY SPERRY RAIL SERVICE



(B) HY-RAIL DETECTOR CAR OWNED AND OPERATED BY BURLINGTON NORTHERN RAILROAD

FIGURE 4. TYPICAL DETECTION VEHICLES

in the U. S. in the early 1960's, there has been no significant change in the basic design used by the industry. This particular ultrasonic application resulted partly as a spin-off from German technology. The magnetic rail inspection systems are basically the same design and configuration used in the 1930's.

Some work is currently in progress in the industry to adapt high-speed data processing to rail inspection. This work has been primarily directed toward aiding equipment operators by screening data and presenting the inspection results in a form that can be quickly and reliably interpreted. The results of this work are planned to be applied in the field within the next year.

Current government developmental work on automating inspection equipment has aimed at increasing inspection speed and reliability by relieving the equipment operator of the need to manually adjust controls or to make operational decisions. Results of this work include the development of automatic controls to adjust amplifier gain, adjust signal gates and maintain sensor alignment. The automatic amplifier gain control compensates for variation in acoustic signals caused by changes in the metallurgical properties of rail steel and sensor-to-rail coupling conditions. The gate control device automatically re-adjusts signal gates, when the heights of rails under inspection change, without having to stop the vehicle. The automatic sensor alignment device controls the lateral position of the sensor relative to the rail head to insure that the sensor is aligned directly over the acoustic center of the rail. This alignment device compensates for sensor mis-alignment caused by worn rails or changes in rail head size. These automatic controls have been tested and are currently in use in the field.

The detectability and repeatability of rail inspection operations depend primarily on two basic factors: equipment performance and operational conditions. The performance variability within the fleet is probably proportional to the number of different designs and configurations. Also, changing environmental conditions cause performance variation, even in a single equipment set. Finally, inspection results are directly dependent on a human operator. Studies of the performance of operators have revealed a definite variation over a given time period. Common practice in the rail inspection industry is to rotate operators within the crew to prevent fatigue and boredom.

Although basic equipment performance can be improved through R & D and operator skills can be enhanced by training, the environment on revenue track is virtually uncontrollable. Rail surface conditions are probably the most significant. Shelling rail and engine burns, for example, can mask underlying defects from acoustic or electromagnetic inspection systems. Dirty rail surfaces can prevent acoustic signals from entering the rail, rendering ultrasonic systems completely ineffective. Weather conditions, particularly extreme cold, have a definite effect on the performance of all types of inspection operations. Track geometry deviations have a significant effect on sensor positioning and thus influence the effective inspection speed. Thus, it is likely that mobile rail inspection will remain an imperfect art in the future.

CONTROL OF RAIL DEFECTS

It is apparent from the foregoing review that net populations of undetected defects must be assumed to remain in track after periodic mobile inspections. Railroads control these populations by means of frequent visual inspections designed to spot large-sized defects and broken rails as they occur. This second line of defense remains effective as long as it is not overwhelmed by large quantities of these critical defects.

Minimum inspection requirements embodied in the current Track Safety Standards take an open-loop approach to control of the net population, as shown schematically in Figure 5. The simplicity of this approach is administratively advantageous. The scheme is technically disadvantageous because it requires that inspection resources be committed uniformly, regardless of variations in the numbers of defects occurring at different track locations, unless a railroad possesses the resources to conduct more than the minimum required inspections.

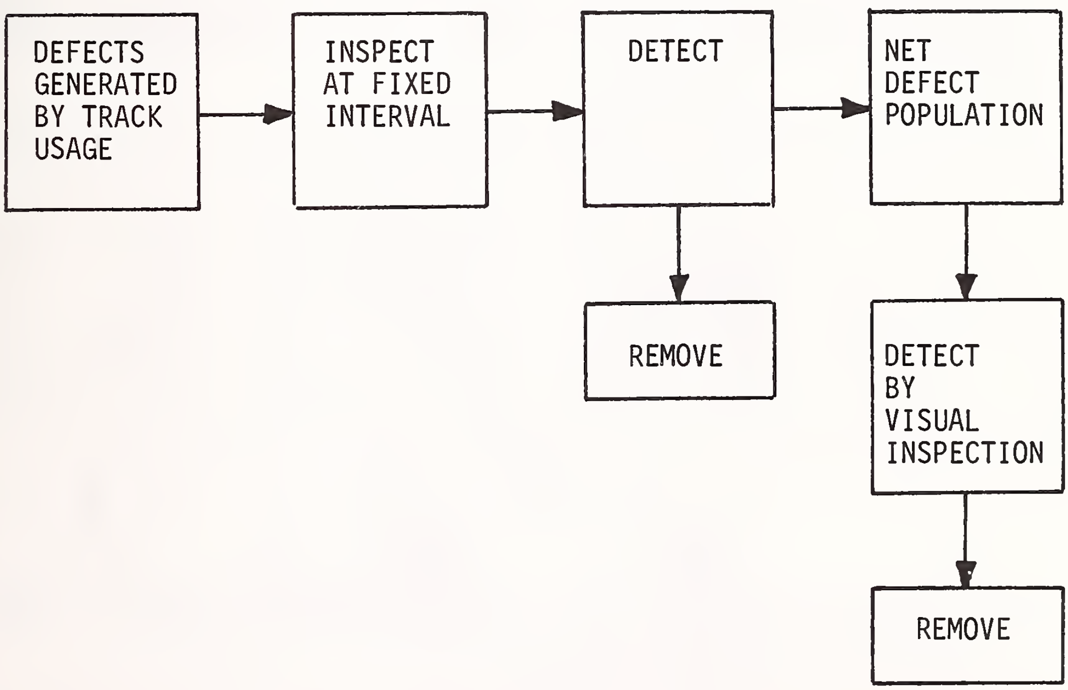


FIGURE 5. CURRENT APPROACH TO CONTROL OF RAIL DEFECTS

The Federal Railroad Administration and the American Railroad Engineering Association are investigating a modified scheme for defect control. The results of detector-car inspections are to be fed back as an adjustment to the inspection interval as shown schematically in Figure 6. Here it is assumed that a certain generic defect type is "critical", in the sense that derailment incidence is a function of the population level for the given type. In addition to removal or other remedial action, the detection process is now followed by classification to identify the count of the critical-type defects that have been detected. The count for critical-type defects is then used to adjust the interval to the next inspection.

The proposed approach is administratively complex, but it offers a potential for flexible allocation of resources away from "well behaved track" toward track that is beginning to generate increasing populations of defects. It is hoped that this closed-loop approach will lead to reduction of the net defect population without any significant increase in total inspection resources. However, it remains to be seen whether the inspection interval adjustment scheme is practical and whether it will allow a railroad to react to changing track usage.

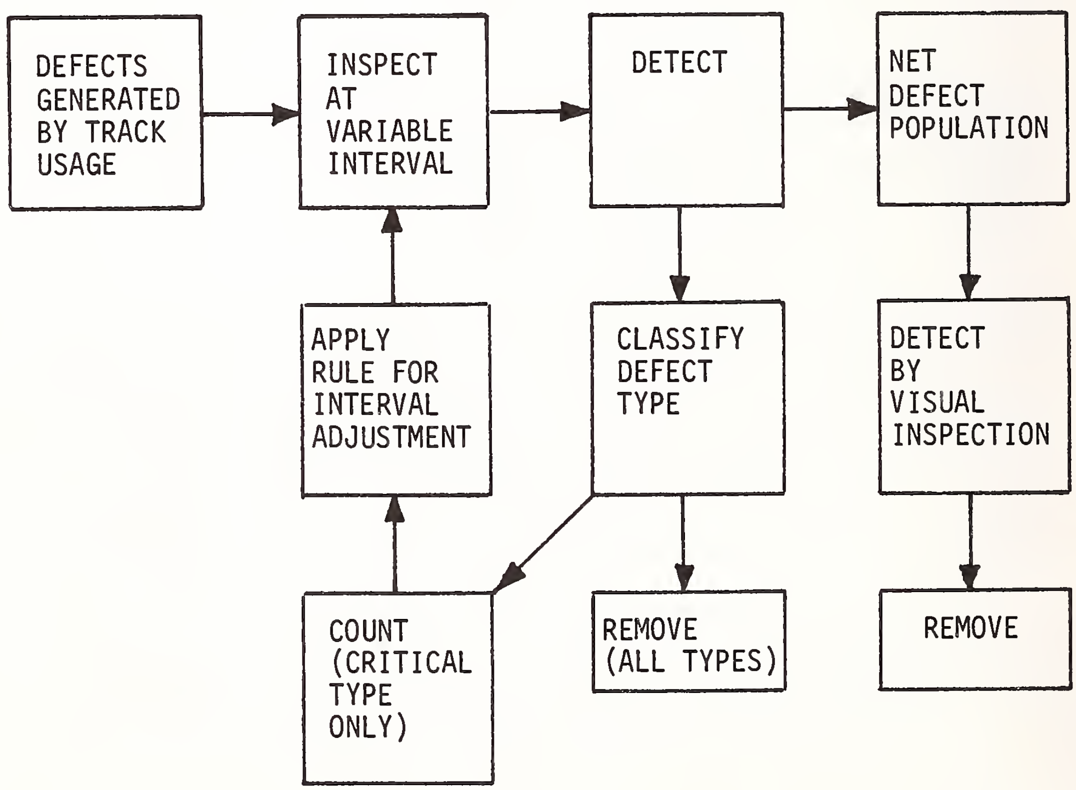


FIGURE 6. PROPOSED APPROACH TO CONTROL OF RAIL DEFECTS

The latter question is the focal point of a comprehensive study, just getting underway, in which the defect occurrence statistics of two Class 1 railroads will be used to formulate a simulation model for testing the technical effectiveness of the proposed control scheme. The simulation studies will investigate flexibility available to the railroads. In the following sections, results will be presented from some preliminary studies in which the close-loop concept was investigated to first order by neglecting effects associated with the distribution of defects along the track.

DEFECT POPULATION DYNAMICS

The reaction ability of an inspection interval adjustment rule can be assessed by means of a population dynamics model which simulates a nominally uniform section of track. Thus, the only independent variable in the simulation is track usage, represented by traffic tonnage in million gross tons (MGT).

Figure 7 summarizes the conceptual elements of a generic simulation model applicable to a single defect type. The continuous range of defect size is divided into a finite number of subranges ("bins") to permit detailed tracking of the population. These bins extend from an "initiation size", S_O, based on metallurgical laboratory data, to a "critical size", S_{CR}, based on fracture mechanics and/or analyses of service defects. The range from S_O to S_{CR} spans the macrocrack growth regime. In particular, S_{CR} is set such that rail breakage can be expected within one or a few train passages after a defect reaches this size.

An empirical initiation model feeds defects into the first bin as a function of accumulating tonnage. A macrocrack growth model then distributes these defects across the other bins as a function of tonnage accumulated since initiation. As the defects are moved to larger sizes, a "minimum detectable size", S_D, is encountered. For sizes exceeding S_D, there exists a finite probability for detecting the defects by means of the detector car. The model accounts for periodic inspection by removing appropriate fractions of the defects in each detectable bin. The sum of these counts D_C, is then considered to be available for use in adjusting subsequent inspection intervals. Those defects which escape detection to reach size S_{CR} are accumulated into a subpopulation representing the burden on the backup visual inspection effort.

APPLICATION TO DETAIL FRACTURE

Use of the population dynamics model will be illustrated with data which approximate very roughly the behavior of detail fractures. Figure 8 illustrates a detail fracture that grew in a rail in the Facility for Accelerated Service Testing (FAST) at the DOT Transportation Test Center. The detail fracture is considered to be a critical

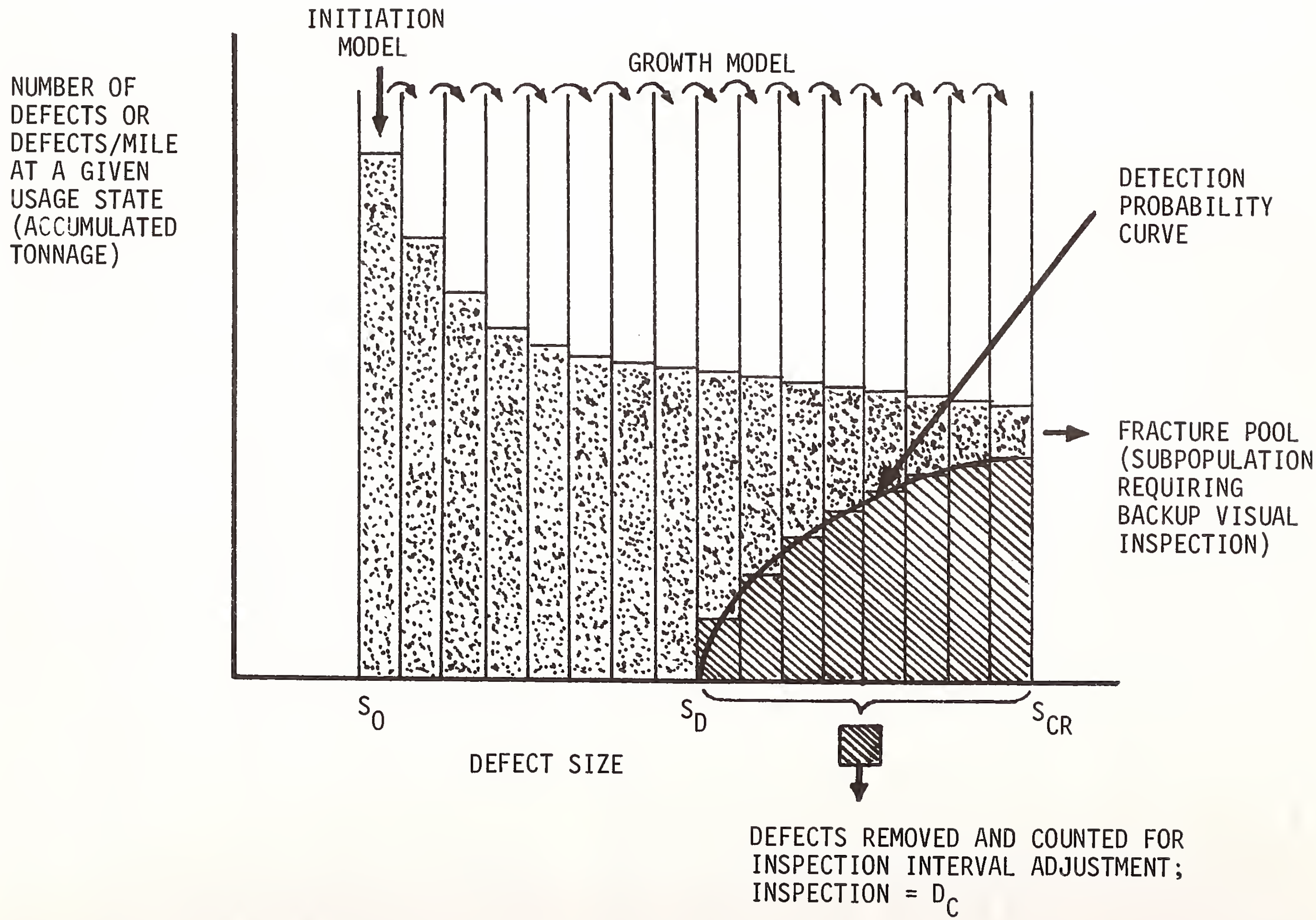


FIGURE 7. DEFECT POPULATION DYNAMICS MODEL

FIGURE 8. EXAMPLE OF DETAIL FRACTURE AT TTC FACILITY FOR ACCELERATED SERVICE TESTING

(Note: Detail fractures in revenue track generally exhibit a smooth fracture face, rather than the "clamshell" pattern shown here.)

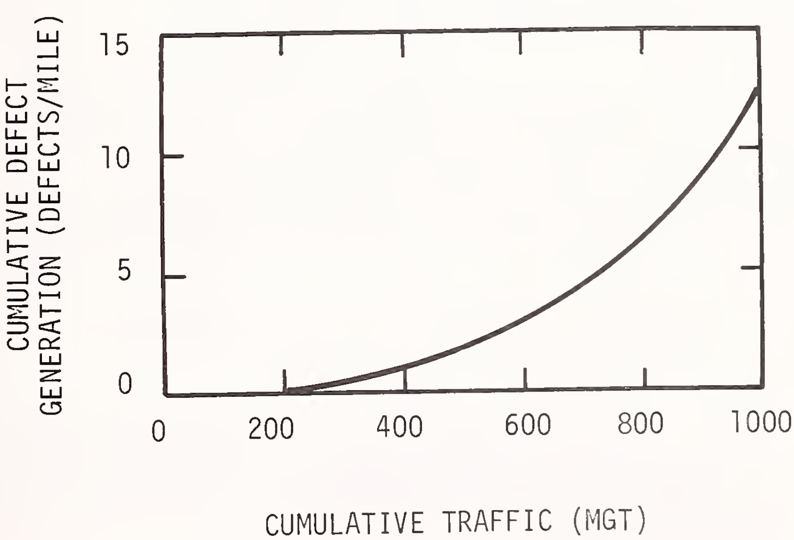


FIGURE 9. TYPICAL DEFECT GENERATION BEHAVIOR

defect type because its location inside the rail head makes it more difficult to detect than defects which grow in a surface cracking mode.

Figures 9 through 11 summarize the data that were used in the simulations. Figure 9 depicts a generic curve for the cumulative number of defects per mile, D generated by aging track:

$$D = CT^3 \quad (C = 1.25 \times 10^{-8} \text{ Defect/Mile - MGT}^3) \quad (1)$$

where T is the accumulated tonnage in MGT.

Equation 1 is an empirical fit to aggregated data for many different defect types, and is thus not strictly applicable to detail fractures. In this case, however, the best available data have been adopted for illustrative purposes only. The initiation model derived from Equation 1 is the count of defects, ΔD, which appear during the interval, ΔT between inspections:

$$\Delta D \cong \frac{dD}{dT} \Delta T = 3CT^2 \quad \Delta T \quad (2)$$

The defect size, S, measured as a percentage of the rail head area (HA) is assumed to increase exponentially with the tonnage accumulated after initiation. The regime of macrocrack growth is assumed to span the following range:

Initiation size,	S_o = 1% HA
Minimum detectable size,	S_D = 5% HA
Critical size,	S_{CR} = 90% HA

(The value of S_{CR} is probably somewhat high. Figure 10 depicts that portion of the defect growth curve between S_D and S_{CR}.

Defect detection is simulated by a generalized detection propability curve of the form

$$P(S) = 1 - \exp\left\{\left[\frac{S - S_D}{S_{CR} - S_D}\right]^M \ell n \ (1 - P_{CR})\right\} \quad (3)$$

where $P_{CR} = P(S_{CR})$ and M is a dimensionless parameter used to change the shape of the detection curve and the average detection probability. A measure of overall detection propability is obtained by averaging P(S) across the size range:

$$P_{AV} = (S_{CR} - S_D)^{-1} \int_{S_D}^{S_{CR}} P(S)dS \quad (4)$$

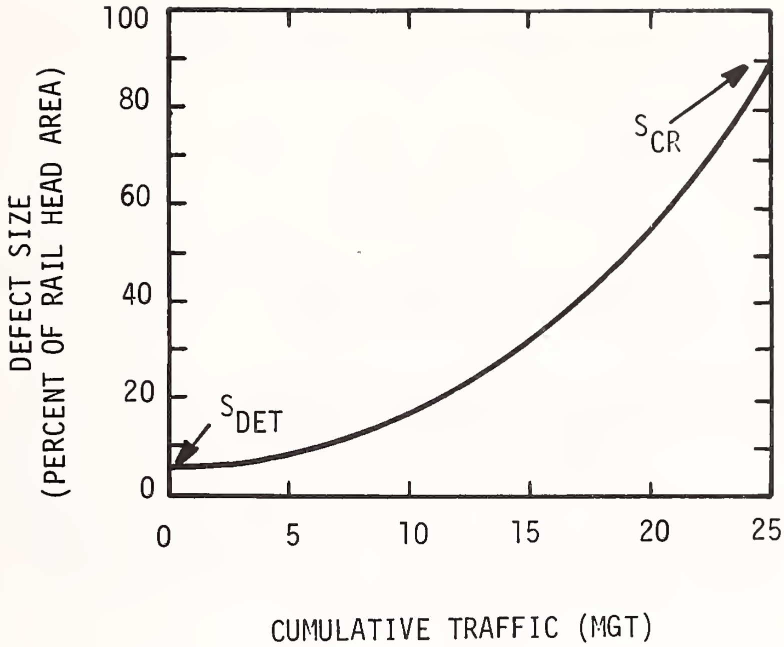


FIGURE 10. ASSUMED DEFECT GROWTH BEHAVIOR

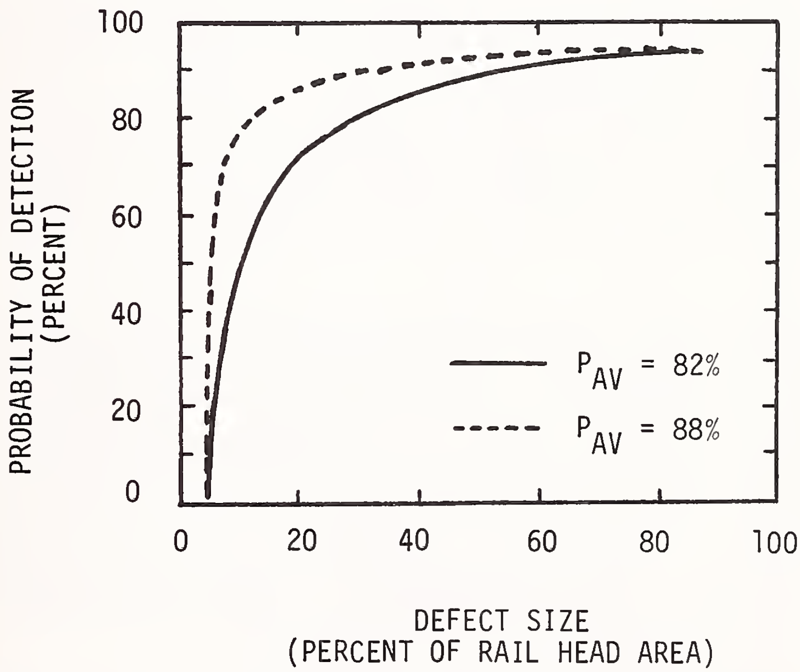


FIGURE 11. ASSUMED PROBABILITY OF DEFECT DETECTION

Figure 11 depicts two assumptions for P(S) corresponding to the following parameter values:

$P_{CR} = 95\%$ $M = 0.5$ $P_{AV} = 82\%$

$P_{CR} = 95\%$ $M = 0.25$ $P_{AV} = 88\%$

Comparative analyses with these cases will be used to illustrate the effect of improving detection probability.

SIMULATION RESULTS

The foregoing data were used to make a simulated test of the effectiveness of the following scheme for inspection interval adjustment. The initial inspection interval is set at ΔT = 20 MGT, which lies within the range of intervals currently used on typical revenue track. Whenever the detected defect count, D_C, (see Figure 7) exceeds a trigger value, D_A, the succeeding inspection interval is reduced to $\Delta T'$ in accordance with:

$$\Delta T' = D_C \ \Delta T / D_A \qquad (5)$$

Thereafter, the inspection interval is fixed at $\Delta T'$ until D_C again exceeds D_A.

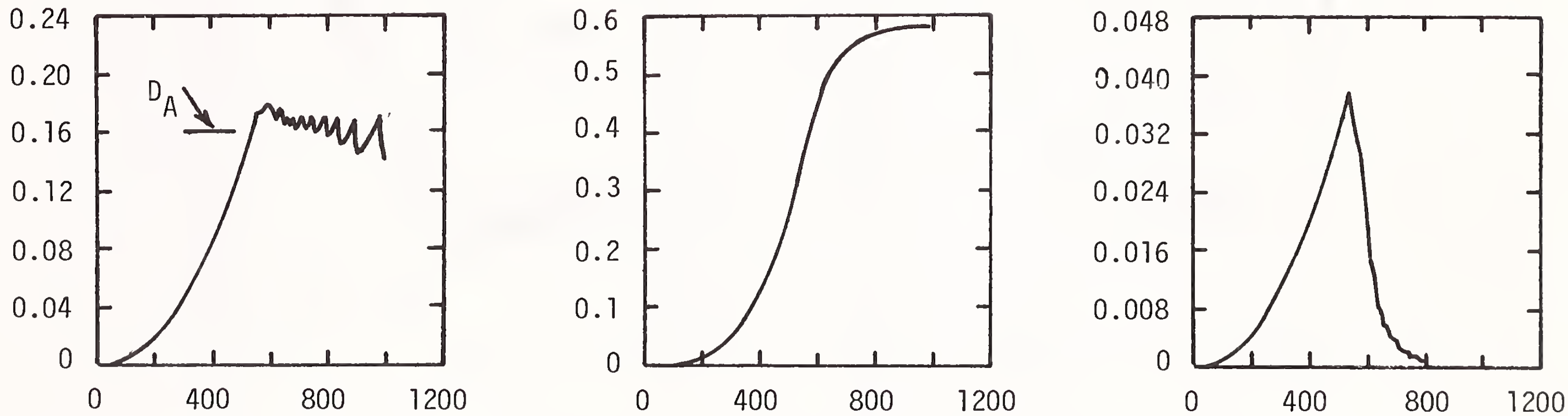
The simulation begins at zero accumulated tonnage, and is continued until the inspection interval decreases to 1 MGT. Figure 12 depicts the results for the baseline case (inspection with P_{AV} = 82% and trigger D_A = 0.16 defect per mile). Figure 12a shows that interval reduction begins at about 460 MGT. Thereafter, the detected defect count oscillates as the effects of inspection interval reduction and defect population pressure alternately dominate. The simulation reaches 1,000 MGT of accumulated traffic when the inspection interval reaches 1 MGT.

The remainder of Figure 12 depicts two possible measures of the burden placed upon backup visual inspection resources. Figure 12b shows the accumulated number of defects per mile which manage to escape detection and thus enter the pool of imminent breaks and fractures. This curve would have continued to rise at an accelerating rate, had the inspection interval not been adjusted. However, the adjustment rule eventually brings the breakout pool under control at a level of 0.58 defect/mile. Figure 12c illustrates the number of subcritical defects which will enter the breakout pool during each inspection interval. This parameter rises to about 0.038 defect/mile, but thereafter decreases rapidly as the inspection interval is reduced.

The baseline results will now be compared with two variations: a 7-percent improvement in average detection probability (P_{AV} = 88% instead of 82%); or a 25-percent reduction of the trigger (D_A = 0.12 instead of 0.16 defect/mile). Figure 13 depicts the effects of these changes on

ORDINATE UNITS: DEFECTS PER MILE

ABSCISSA UNITS: MILLION GROSS TONS OF TRAFFIC



(A) DEFECTS DETECTED COUNTED AND REMOVED PER INSPECTION

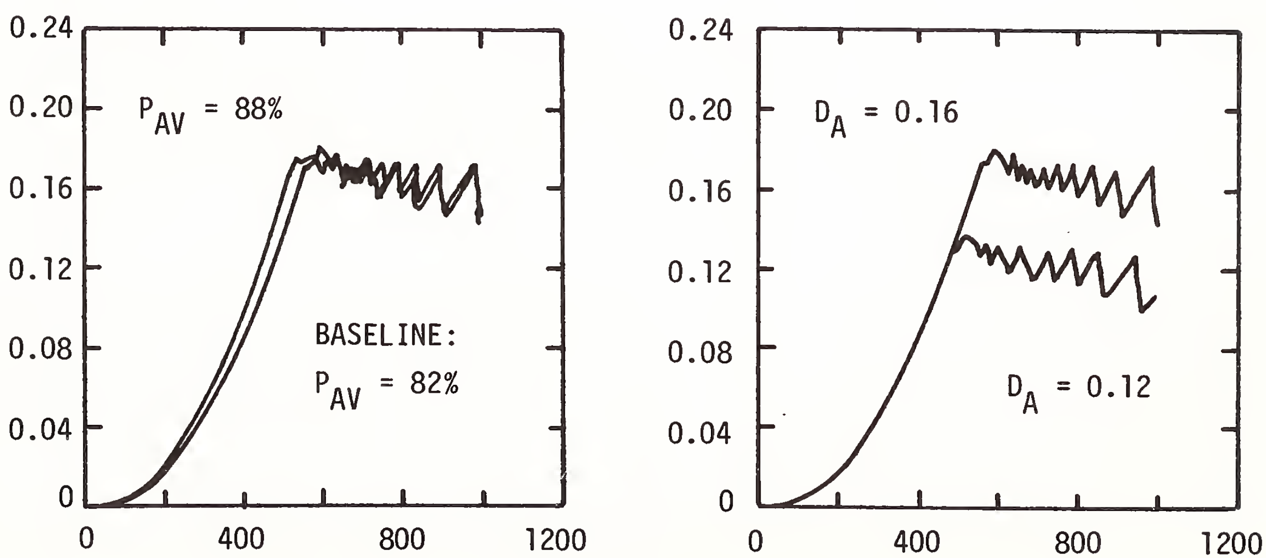
(B) CUMULATIVE UNDETECTED DEFECTS ATTAINING IMMINENT BREAKOUT SIZE

(C) NUMBER OF SUBCRITICAL DEFECTS WHICH WILL REACH IMMINENT BREAKOUT SIZE BEFORE NEXT INSPECTION

FIGURE 12. BASELINE SIMULATION OF DEFECT POPULATION DYNAMICS

ORDINATE UNITS: DEFECTS PER MILE

ABSCISSA UNITS: MILLION GROSS TONS OF TRAFFIC



(A) EFFECT OF 7 PERCENT IMPROVEMENT IN DETECTION PROBABILITY (D_A FIXED AT 0.16)

(B) EFFECT OF 25 PERCENT REDUCTION IN TRIGGER LEVEL (P_{AV} FIXED AT 82 PERCENT)

FIGURE 13. RESPONSE OF DETECTED DEFECT COUNT TO IMPROVED DETECTION OR LOWER ACTION THRESHOLD

inspection management. The inspection interval reductions begin earlier than the baseline in both cases. Improved detection with baseline trigger appears to maintain about the same control on defect count as the baseline. On the other hand, baseline detection with reduced trigger controls the defect count to a much lower level.

One might conclude from the foregoing results that trigger reduction is the more effective of the two improvements. However, the other simulation outputs reveal that 7-percent improvement in P_{AV} is actually more effective than 25 percent reduction of D_A. Figure 14 shows that the visual backup inspection burden is alleviated much more by the detection improvement than by the trigger reduction.

DISTRIBUTION OF DEFECTS ALONG TRACK

It is unrealistic to expect that defect occurrences will be distributed uniformly, in a statistical sense, over arbitrarily long sections of revenue track. Also, it is unlikely that the segmentation of track for defect accounting purposes can be made to correspond precisely to the boundaries between track sections having different inherent defect rates. Under these conditions, it is prudent to investigate the effectiveness of defect frequency dependent inspection when defects are nonuniformly distributed along the track. Work on spatial modeling has been started, but no complete simulations have yet been conducted.

The first step in the investigation was taken by analyzing several years of field data from a two hundred mile section of revenue track. The analysis has indicated that a Poisson distribution might describe the count of defects detected per 10-mile segment. The family of Poisson distributions obeys a unique linear relationship,

$$\beta_1 = \beta_2 - 3 \qquad (6)$$

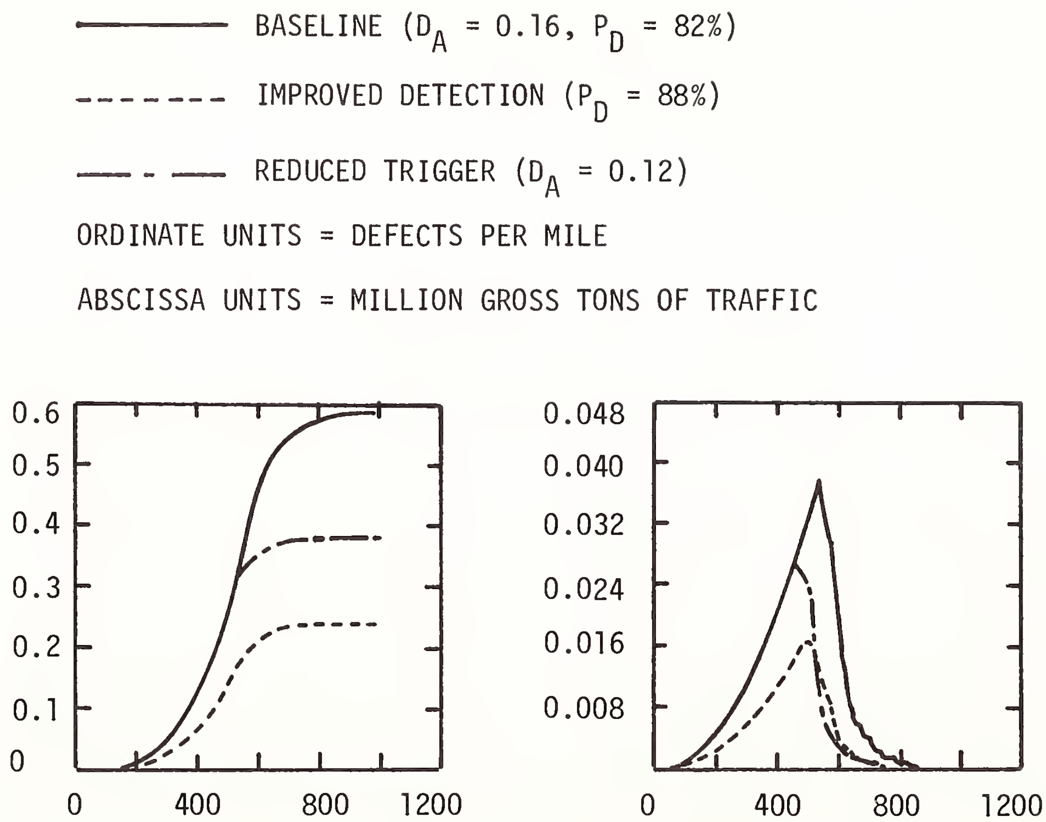
between the shape descriptors of its probability density:

$$\beta_1 = |\mu_3/\sigma^3| \quad \text{(skewness)} \qquad (7)$$

$$\beta_2 = \mu_4/\sigma^4 \quad \text{(kurtosis)} \qquad (8)$$

where σ^2, μ_3 and μ_4 are respectively the second, third and fourth central moments of the distribution.

Figure 15 compares the Poisson model relationship with estimates of β_1, β_2 computed from the field data, and from a simulation run with a random number generator known to produce Poisson data. Comparison of the simulation and field data indicates that the field data scatter can be attributed to a finite sample-size effect. Therefore, the Poisson distribution appears to provide an appropriate spatial model.



(A) CUMULATIVE UNDETECTED DEFECTS ATTAINING IMMINENT BREAKOUT SIZE

(B) NUMBER OF SUBCRITICAL DEFECTS WHICH WILL REACH IMMINENT BREAKOUT SIZE BEFORE NEXT INSPECTION

FIGURE 14. RESPONSE OF FRACTURE POOL TO IMPROVED DETECTION OR LOWER ACTION THRESHOLD

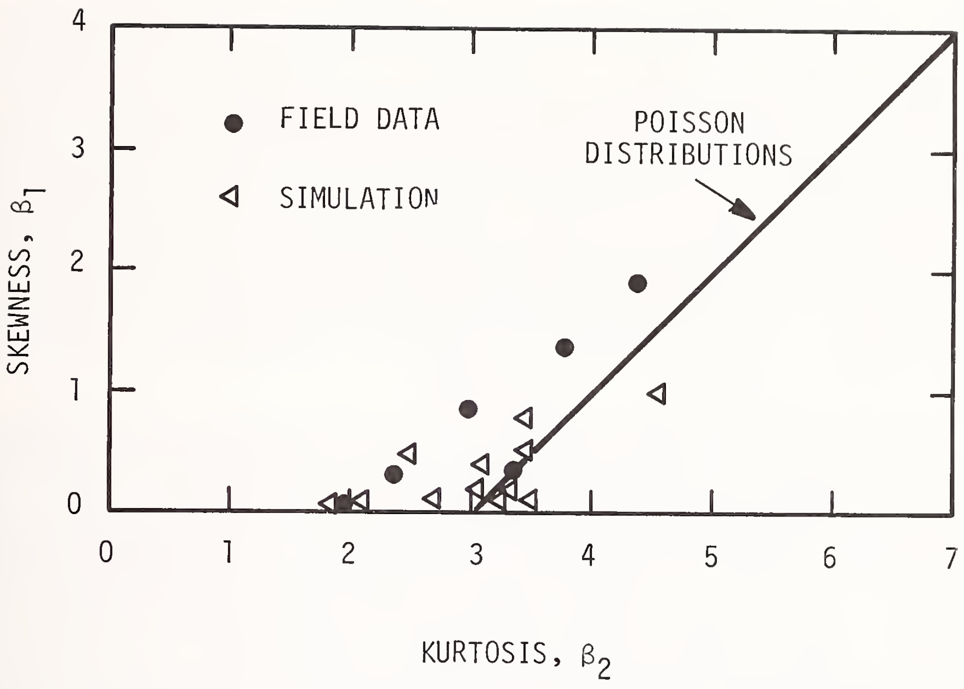


FIGURE 15. SHAPE DESCRIPTORS FOR POISSON DISTRIBUTION AND DATA HISTOGRAMS

An additional and independent rationale for adopting this model is that Poisson data is distributed with no bias in location, i.e. the appearance of a defect in one rail is equally likely as its appearance in any other rail. This type of characteristic appeals to the statistician as a working hypothesis, in the absence of any definite evidence to the contrary. The Poisson model will be used in the future to conduct more comprehensive analyses of field data, as well as to test the sensitivity of closed-loop control to spatial nonuniformity.

CONCLUDING REMARKS

Preliminary results have been presented from studies based on a simplified model of rail defect population dynamics. The simulation included a closed-loop scheme for controlling the defect population by varying the inspection interval in accordance with the findings of past inspections. The results indicate that the closed-loop concept is a promising approach to a better allocation of inspection and maintenance resources. The opportunity for better allocation arises because the inspection interval might be lengthened during the early track life stage, when a backup visual inspection program can cope with the defect population. In other words, it may be possible to transfer mobile inspection resources to the later stages of track life, when the defect population begins to rise rapidly. The preliminary study has also shown that the simulation answers can be sensitive to underlying assumptions in the model. A specific illustration of sensitivity to detection probability has been presented. The model undoubtedly possesses similar sensitivities to the rate of defect generation, the rate of defect growth, and the characteristics of defect distribution along the track. Because of these sensitivities, it is obvious that better input data will be needed before such simulations can be used to support decisions about the management of rail defects in revenue track. The wide diversity of detector car equipment and operating conditions requires that detection probability bounds be established for the purposes of the simulation studies. Also, detailed information is needed about the occurrence histories, spatial distributions and growth rates of specific types of rail defects.

The Federal Railroad Administration and the American Railway Engineering Association are currently engaged in a cooperative effort to improve track performance standards. A major component of this effort is devoted to the collection of refined data on rail defect behavior. The task of better characterizing defect detection probabilities is still in the planning state. However, work on the other aspects is underway. Defect occurrence and spatial distribution statistics are to be developed from detailed studies of the defect histories on two Class 1 railroads. Defect growth rates are to be measured in a series of tests in which flawed rails removed from revenue track will be subjected to accelerated service testing at the DOT Transportation Test Center. The results of these investigations will ultimately be combined in realistic simulations to assess options for improving track safety.

AUTOMATED NDE FOR DETECTION OF BRAKING ABNORMALITIES OF TRAINS

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Abstract

Wayside instrumentation was designed, fabricated, and tested that demonstrates the feasibility of detecting braking system performance on moving rail cars. This instrumentation is being converted into a Braking Inspection System (BIS) prototype. The objective of the BIS is to provide a digitized output for braking alarm signals, wheel weight, wheel temperature and braking reaction force.

Currently, "braking" or "no-braking" conditions can be detected and presented in a digitized output. "Inadequate" or "excessive" braking is dependent on the actual braking force magnitude. Thus, efforts are underway to discriminate the discrete braking force from a measurement signal that includes a number of random forces not associated with the braking action.

SESSION III

HIGHWAY AND ROAD BRIDGES

Chairman: J. Krugler, Federal Highway
Administration

FAILURE ANALYSIS OF HIGHWAY BRIDGES

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Abstract: An evaluation of cracking that developed in two bridge structures is considered in the paper. Fatigue cracking was first observed at the Yellow Mill Pond Bridge on the Connecticut Turnpike (I-95) in 1970. Fatigue crack growth resulted in complete fracture of a tension flange in one of the girders. Smaller cracks were discovered in several other beams. Fatigue cracking started at the end weld of cover plates welded on the rolled section which forms the longitudinal girders of the bridge structure.

The results from the bridge observations are compared with laboratory fatigue test data on small size and full size cover-plated beams. The laboratory tests and analytical predictions of crack growth agree well with the observations made on the Yellow Mill Pond Bridge.

A large crack was discovered in November 1973 in a fascia girder of the suspended span of the Quinnipiac River Bridge near New Haven, Connecticut. Crack propagation was found to occur in different stages. A detailed examination of the fracture is given in the paper. Fatigue cracks were found to originate at lack of fusion areas in horizontal stiffener splices. After the crack penetrated the web thickness it resulted in brittle fracture of the web. Crack instability developed when the stress intensity at the crack tip reached the material fracture toughness. The fracture toughness was estimated from J-integral measurements and from Charpy V-Notch test data.

Key words: Analysis; bridges, crack propagation; failure; fatigue; fracture; fracture surface; fracture toughness.

1. Introduction: The web of the main girder of the Quinnipiac River Bridge near New Haven, Connecticut, cracked after nine years of service. Brittle fracture was arrested when one crack tip was near the girder neutral axis and the second crack tip in the bottom flange. The tension flange was about 40% destroyed. The investigation showed that brittle fracture initiated from a fatigue crack that had entered the girder web.

Fatigue crack growth was also observed in several cover-plated beams of the Yellow Mill Pond Bridge in Bridgeport, Connecticut. The fatigue cracks were discovered after the bridge was in service twelve years. A fatigue crack had propagated through the 32 mm (1.26 in.) thick tension flange. Crack instability did not develop until the crack had propagated several inches up the web.

The crack behavior of both highway bridges is evaluated and analyzed in this paper.

2. <u>Quinnipiac River Bridge</u>: In October 1973 a large crack was discovered in a fascia girder of the suspended span of the Quinnipiac River Bridge near New Haven, Connecticut [11]. Figure 1 is a photograph showing the bridge profile. The crack was discovered approximately 10.4 m (34 ft.) from the left or west end of the suspended span. The suspended span is 50.3 m (165 ft.) long. The structure is noncomposite and the girders are 2.8 m (9.23 ft.) deep at the crack location.

The crack initiated in the girder web and propagated to the middepth of the girder and as shown in Fig. 2 had penetrated the bottom flange surface when discovered. The bridge had experienced about nine years of service at the time the crack was discovered.

Examination of the fracture surfaces indicated that the fracture had initiated at the web stiffener intersection. The fracture surfaces indicated that a butt weld in the longitudinal stiffener had been made at this location but had never been completely fused. Close examination revealed that only a surface pass had been made and the reinforcement removed by grinding. The surfaces of this portion of the fracture were severely corroded from their exposure to the environment.

Replicas were made at the web-longitudinal stiffener intersection, so that the fracture surface could be examined by transmission electron microscopy. Visual examination at low magnification indicated that fatigue crack growth was very probable in the web. The fracture surface in the web on each side of the longitudinal stiffener indicated that a cleavage or "brittle fracture" had occurred after the crack penetrated the web thickness. Cleavage fracture extended down the web and penetrated about 25 mm (1 in.) into the flange before it had arrested.

Examination at the web-stiffener intersection confirmed that fatigue crack growth had occurred. Fatigue crack growth striations were observed adjacent to the longitudinal web stiffener break. Figure 3 shows a photograph at high magnification (49125 X) or a replica of the fracture surface. The fatigue crack striations are apparent. Estimates of the rate of crack propagation were made on the basis of the striation spacing. These indicated that crack growth rates between 1.8 and 5.1×10^{-5} mm/cycle (7×10^{-7} and 2×10^{-6} in/cycle) were occurring in the region examined.

2.1 Material Characterization: At the critical location the web and the flange are fabricated from A36 steel. Tensile coupons provided a yield strength of 254 MPa (36.8 ksi), an ultimate strength of 420 MPa (60.9 ksi) with a 43% reduction in area and 32% elongation in a 203 mm (8 in.) gage length. Standard ASTM Type A Charpy V-Notch (CVN) [6] specimens were fabricated from both the web and flange material, and compact tension specimens were made from the web.

The Charpy V-Notch tests showed that both flange and web satisfied the toughness requirements for Group 2 of the 1974 interim AASHTO Specification [2]. The average CVN impact value for the web was 27 J (20 ft-lb.) at 4° C (40° F). The flange provided an average value of 47 J (35 ft-lb.) at 4° C (40° F).

Several compact tension specimens were fabricated from material from the web. Due to the limited material removed from the web, only eight compact tension specimens 51 mm x 64 mm x 11 mm (2 in. x 2.5 in. x 0.41 in.) were made. The crack was oriented in the same direction as in the bridge structure.

The specimens were tested at five different temperatures, one specimen at -40° C (-40° F), two at -29° C (-20° F), two at -23° C (-10° F), two at -12° C (+10° F) and one at -0° C (32° F). The loading time for the fracture test was about 0.1 second.

The compact tension specimen tests could not be evaluated using the ASTM E-399 specification [8], because even at -40° C (-40° F) the available material was too thin relative to the fracture toughness. For convenience the fracture toughness was therefore estimated using the J-integral procedure proposed by Rice [9].

The empirical relationship suggested by Rolfe and Barsom [2] was used to estimate the dynamic fracture toughness K_{Ic} and the toughness for one second loading time were estimated by considering a temperature shift [2]. The estimated K-values are shown in Fig. 4 and compared with the 0.1 second compact tension tests.

2.2 Stresses at the Crack Location: An equivalent stress range from the live load can be approximated from Miner's Rule using the gross vehicle weight distribution given in Ref. 4 assuming a value of $\alpha = 0.7$ for the stress range reduction factor. This results in $S_{rMiner} \simeq 13.4$ MPa (1.95 ksi) in the flange as compared to the design stress range $S_{rDesign} = 30.0$ MPa (4.35 ksi). The stress at the longitudinal stiffener is about 60% of the maximum flange stress. Hence at the longitudinal stiffener $S_{rMiner} \simeq 8.1$ MPa (1.17 ksi). The maximum live load stress was estimated to be about 13.8 MPa (2 ksi). The average daily truck traffic crossing the bridge was estimated to result in about 1,600,000 random stress cycles per year, corresponding to the

$S_{rMiner} \simeq 8.1$ MPa (1.17 ksi). This corresponds to an average daily truck traffic (ADTT) of about 4300.

The dead load stress, σ_{DL}, the stress due to the weight of the structure was 34.1 MPa (4.95 ksi) at the level of the longitudinal stiffener on the web.

The fatigue crack started in the horizontal stiffener and grew into the web. Since the two continuous fillet welds connected the longitudinal stiffener to the web, the crack tip entered the web in a zone of high residual tensile stress.

Residual stresses are present in the bottom flange due to the welding of the web to the flange. It was assumed that the tensile residual stresses are distributed over a semicircular area at the connection between the web and the flange with a radius of half the thickness of the web, plus the weld leg size. Over this area the stresses were also assumed semicircular and their magnitude equal to half the yield point. The dead load stress in the bottom flange is 56.8 MPa (8.25 ksi) and the assumed maximum live load stress 23.0 MPa (3.3 ksi).

2.3 Analysis of Crack Growth: From the examination of the fracture surface it was apparent that crack growth had occurred in the Quinnipiac River Bridge in a number of stages and modes. These stages are illustrated schematically in Fig. 5.

Under normal traffic, and if about 6 mm (0.25 in.) of the 9.5 mm (0.38 in.) thick longitudinal stiffener were unfused, fatigue cracking would require between 2 and 20 million cycles of random traffic, depending on the proximity to a free surface, to propagate through the longitudinal stiffener thickness. If the crack had only been fused about 3.8 mm (0.15 in.) on one plate surface so that an edge crack resulted, only about one million cycles of random traffic would be needed to crack the longitudinal stiffener. Stage II of fatigue crack growth would primarily develop after the stiffener was cracked in two. Electron microscope studies of the fracture surface confirmed the presence of fatigue crack growth striations during state II.

After the weld which connected the two longitudinal stiffeners together was completely broken, the crack front moved into the web. The crack propagated under normal random loading during this stage through the web.

An estimate of the time required to propagate through the web thickness was made assuming the crack penetrated the web as a flat circular crack with center of radius at the tip of the longitudinal stiffener. This model was developed as a result of laboratory fatigue tests which demonstrated a comparative mechanism of fatigue crack growth. The stress intensity factor K for a flat-circular crack with the crack front approaching a free surface is

$$K = \frac{2}{\pi} \sigma (\pi a)^{1/2} \left(\frac{2b}{\pi a} \tan \frac{\pi a}{2b}\right)^{1/2}$$

Where σ = applied stress
a = crack length
b = plate width (width of stiffener plus web thickness)

For an initial crack size of the width of the stiffener 3.5×10^6 cycles are needed to penetrate 95% of the web thickness. The striation spacing measurements are in good agreement with these calculations.

Stage III was the brittle fracture of the web during a time of low temperature. A comparison of fracture surfaces from the compact tension tests and the web crack indicated that the web had fractured when the temperature was between -12° C (10° F) and -23° C (-10° F). The estimated maximum K-value during Stage III is indicated in Fig. 4 and compared with the material fracture resistance. Brittle fracture initiated in a zone of high residual tensile stresses. Once the crack became unstable, it propagated through the zone of lower stresses in the web and was eventually arrested in the flange.

Further fatigue crack growth (Stage IV) developed thereafter and continued until the crack was discovered and repaired.

3. Yellow Mill Pond Bridge: Fatigue cracking was first observed at the Yellow Mill Pond Bridge on the Connecticut Turnpike (Interstate Rt. 95) in 1970 [1]. Fatigue crack growth had resulted in complete fracture of a tension flange. Inspection of two beams adjacent to the fractured beam indicated fatigue cracks had propagated halfway through their tension flange. Also, small cracks were visually observed at several other beams.

The Yellow Mill Pond Bridge is located on the Connecticut Turnpike (Interstate Rt. 95) in the City of Bridgeport, Connecticut. Construction started in 1956 and was completed in 1957. It was opened to traffic in January 1958. This bridge complex consists of 28 simple-span cover-plated steel beam bridges crossing the Yellow Mill Pond channel (14 in each direction of traffic). Each bridge carries three lanes of traffic. The position of the beams and diaphragms in span 10 are shown in Fig. 6. The external facia beam (M88) of the eastbound bridge is skewed as four lanes of through traffic are being reduced to three lanes.

The beams in span 10 are W36X230, W36X280, or W36X300 sections and were rolled from A242 steel. All beams, except the interior facia beam of the eastbound roadway (M86), are fitted with two cover plates (primary and secondary) on the tension flange and a single cover plate on the compression flange.

In October-November, 1970, during cleaning and repainting of the Yellow Mill Pond Bridge, one of the cover-plated steel beams on the eastbound bridge on span 11 was found to have a large crack [4]. The crack had developed at the west end of the primary cover plate on Beam 4 and is shown in Fig. 7. It had grown from the toe of the cover plate transverse fillet weld into the tension flange and up 400 mm (16 in.) into the web.

A visual inspection showed that Beams 3 and 5 in span 11 of the eastbound roadway which were adjacent to the casualty girder had cracks along the cover plate ends. These cracks were subsequently verified by ultrasonic testing and a depth of penetration equal to 16 mm (0.625 in.) was measured. An indication of possible fatigue cracking was also observed at other details on span 10 and on span 11.

Subsequently, other inspections were made at the Yellow Mill Pond Bridge and fatigue cracks were found after removing the paint, dirt and oxides which had accumulated at the weld toe. Inspections were made in December 1970, November 1973, June 1976, November 1976, September 1977, and November 1979 [1]. The results of these inspections are summarized in Fig. 6.

3.1 Material Characterization: The girders of the Yellow Mill Pond Bridge were made from ASTM A242 steel with a yield strength of 395 MPa (57 ksi). Charpy V-Notch tests [6] were conducted on the web and flange material. The material removed from the web shows a transition temperature of 20 J (15 ft-lbs.) at -23° C (9° F). The flange provided 20 J (15 ft-lbs.) at 15° C (59° F). The difference in the transition temperature is due to the different thickness of the web and flange. There was more manganese in the flange material, which may have increased the transition temperature. The CVN test results were used to estimate the dynamic fracture toughness. The fracture toughness at an intermediate loading rate and the static loading rate were estimated from the temperature shift. The estimated fracture toughness is shown in Fig. 8 for the web and flange material.

Three point bend specimen [8] were fabricated from material removed from the web. Only four specimens could be made due to the limited material available. The specimens were tested at temperatures of -40° C (-40° F), -29° C (-20° F), and at -18° C (0° F). They were analyzed using the J-integral method proposed by Rice [9]. The specimens were tested at a loading time of 1.5 seconds which is comparable with the loading time for the bridge structures. The results of the three point bend tests are compared with the fracture resistance estimated from the Charpy V-Notch tests shown in Fig. 8.

3.2 Stresses at the Crack Location: The stresses in the bottom flange near the cover plate and weld are caused by the live load, dead load and residual stresses from rolling the beam and welding the cover plate on the beam. The residual stress distribution for rolled W36X230 profiles with welded cover plates was investigated in Ref. 10. The investigation was made using an analytical model and measurements. Stresses as high as $\sigma_y/2$ are present after welding the cover plate on the beam. The dead load stress in the bottom flange at the crack location is 27.6 MPa (4 ksi). Several investigations were made to determine the stress distribution due to the traffic load.

The empirical relationship suggested by Rolfe and Barsom [2] was used to estimate the dynamic fracture toughness K_{Ic} and the toughness for one second loading time were estimated by considering a temperature shift [2]. The estimated K-values are shown in Fig. 8 and compared with the 1.5 second three point bend tests [8].

Stress History Studies: The eastbound and westbound bridges of span 10 in the Yellow Mill Pond viaduct were selected for two major stress history studies. Both studies were conducted by the State of Connecticut and the Federal Highway Administration. The first study was conducted in July 1971 [4] and the second from April 1973 to April 1974 [5].

It was found that slightly higher stress range events occurred over the extended time period in 1973, then measured during the 1971 study.

A limited strain history record was obtained for the eastbound bridge of span 10 in June 1976.

A large stress event was recorded during the stress history sample of the west end of the eastbound bridge. Inspection of the oscillograph trace indicated that this stress event was caused by a multiple truck presence on the bridge. The stress range recorded at the end of the cover plate was 72.4 MPa (10.5 ksi) for Beam 7.

The histogram for the live load stress range is shown in Fig. 9. Miner and root-mean-square (RMS) stress values, based on the frequency and magnitude of strain events recorded in the July 1971 study and the 1976 sample were calculated. The following equations were used to calculate these values:

$$S_{rRms} = \left(\alpha_i \; S_{ri}^{\;2}\right)^{1/2} \qquad\qquad S_{rMiner} = \left(\alpha_i \; S_{ri}^{\;3}\right)^{1/3}$$

where S_{ri} is the i^{th} stress range in the spectrum and α_i is the fraction of stress ranges of that magnitude.

Based on average daily truck traffic (ADTT) measurements the total number of trucks crossing the bridge was estimated to be about 21×10^6 between the opening of the bridge and the detection of the first crack in 1970. The critical detail was estimated to experience between 21×10^6 and 37.8×10^6 stress cycles.

The live load stress histograms obtained at Yellow Mill Pond are similar to the stress histograms that have been obtained at other cover-plated beam bridges in Michigan, Virginia, Maryland, Tennessee, and Pennsylvania. The highest measured stress range varied from one histogram to another, but the vast majority of these studies indicate an effective (Miner or Rms) stress range between 6.9 MPa (1.0 ksi) and 13.8 MPa (2.0 ksi).

3.3 Fracture Beahavior: The fracture surface was heavily corroded and partially covered with paint when discovered. Because of the corrosion, no detailed fractographic investigation could be made.

The crack started at the transverse fillet weld connecting the cover-plate to the tension flange and grew completely through the tension flange. This fatigue behavior could be expected based on the results from the laboratory studies [3,10]. The observed fatigue behavior is compared with the predicted behavior in Fig. 10.

The stress intensity factor for a large semielliptical crack was determined as a function of the location on the crack front. For a crack size of 25.4 mm (1 in.) the estimated maximum stress intensity factor is 123 MPa $\sqrt{m}$. A comparison of the maximum stress intensity factor with the material fracture toughness given in Fig. 11 suggests that the static fracture resistance is applicable as no crack instability developed in the flange.

The fatigue crack completely penetrated the bottom flange and the crack front moved up the web. At this stage the fatigue life of the cover-plated girder was completely exhausted. The load carried by the casualty girder was redistributed to adjacent girders.

Numerous other details have experienced fatigue crack growth with most cracks varied between 3 mm (0.125 in.) and half the flange thickness.

4. Summary and Conclusion: The cracking that developed in two bridge structures was evaluated using fracture mechanics models for the crack development. Fatigue crack growth was the primary cause of crack development in both bridge structures. The Quinnipiac River Bridge was found to develop a fatigue crack from a butt weld of poor quality in a longitudinal stiffener. Fracture was predicted to occur when the fatigue crack had nearly penetrated the girder web.

The fatigue cracks that have developed at the end of partial length cover plates on the Yellow Mill Pond girders resulted from large numbers of stress cycles ($\sim$40 million) daily the past twenty years. The fatigue resistance of this detail was eventually found to be much less than anticipated at the time of the design. Later experimental and mathematical models of the crack growth demonstrated that the fatigue resistance was compatible with the field observations. No rapid fractures were found to occur as the maximum stress intensity at the crack tip was always less than the fracture resistance of the rolled section.

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Fig. 1 Quinnipiac River Bridge

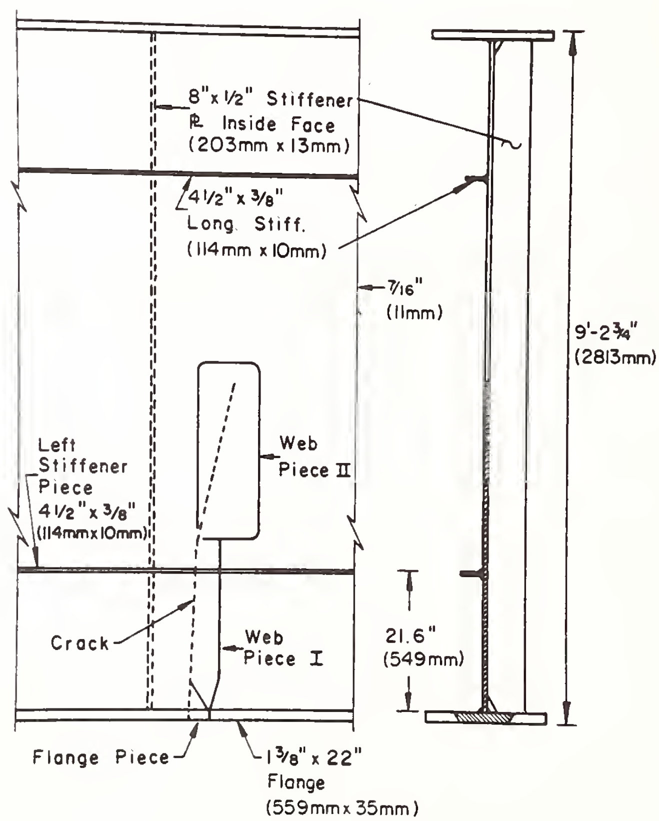

Fig.2 Schematic of the Girder at the Crack Location

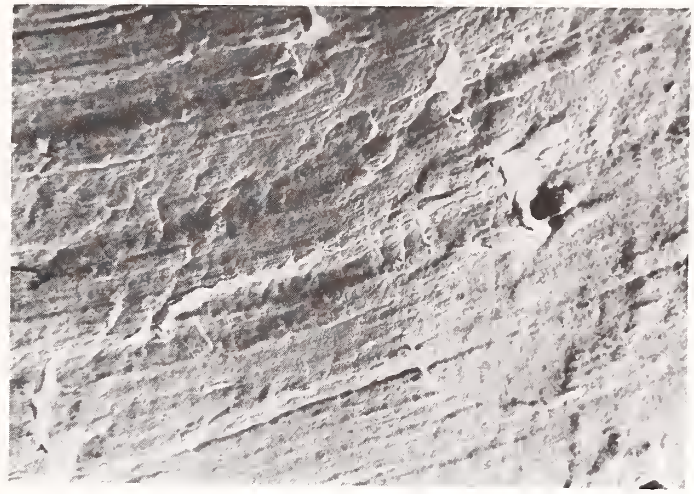

Fig. 3 Fracture Surface in Web Adjacent to Longitudinal Stiffener Showing Striation Markings (49125X)

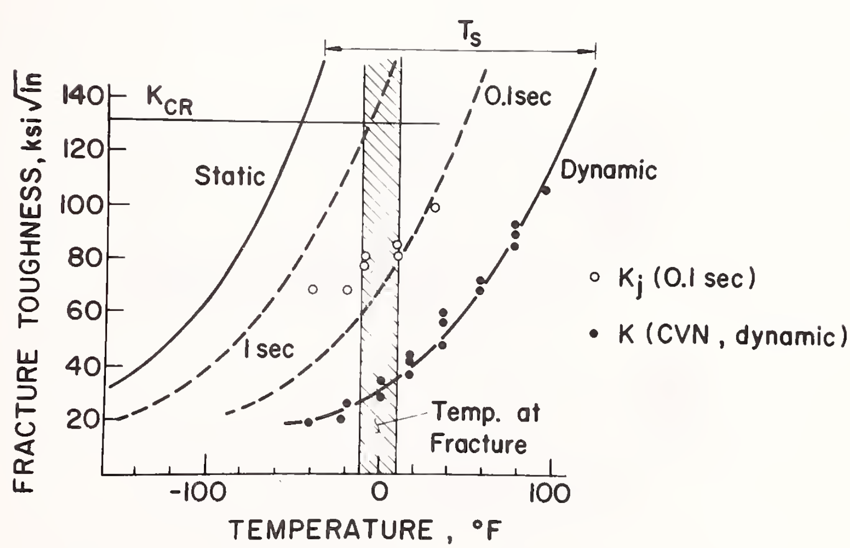

Fig. 4 Fracture Toughness and Stress Intensity Factor at Fracture

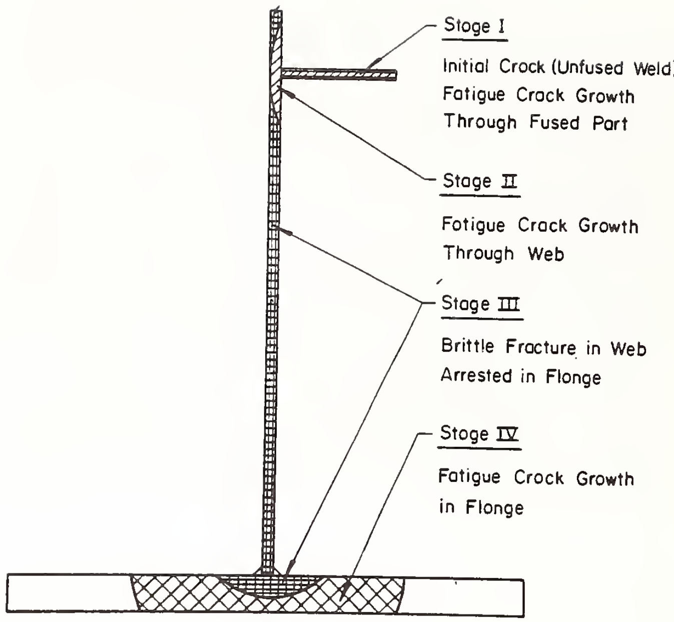

Fig. 5 Schematic of Crack Growth Stages

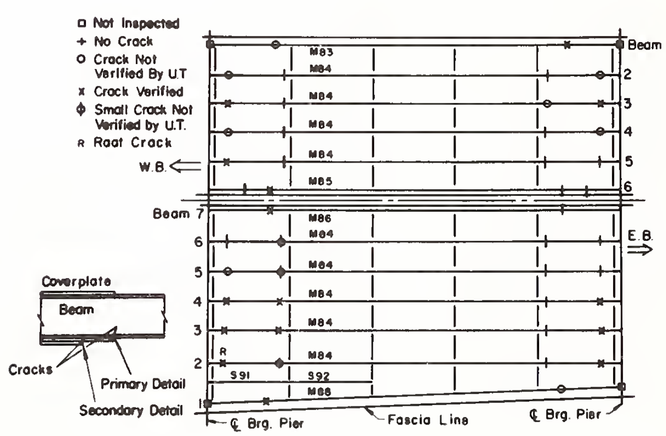

Fig. 6 Plan Showing Inspected Details in Span 10

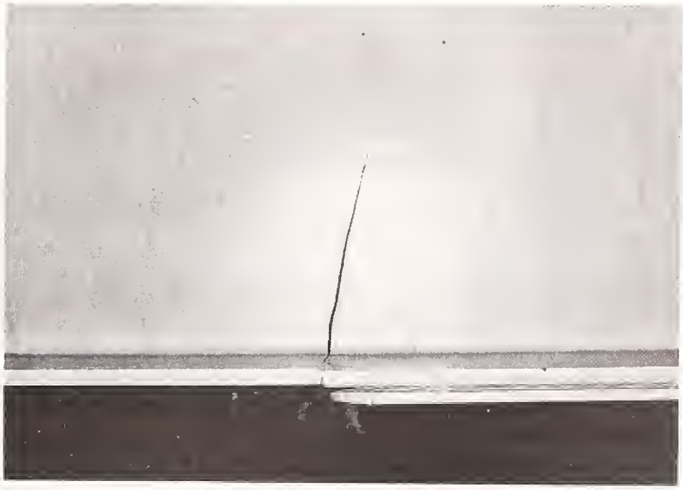

Fig. 7 Fractured Girder Discovered in 1970

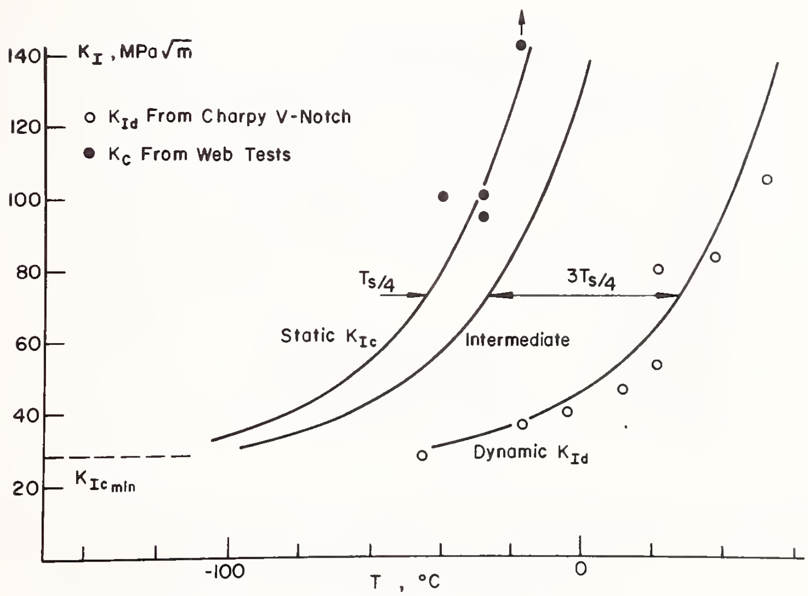

Fig. 8 Fracture Toughness, Flange Material

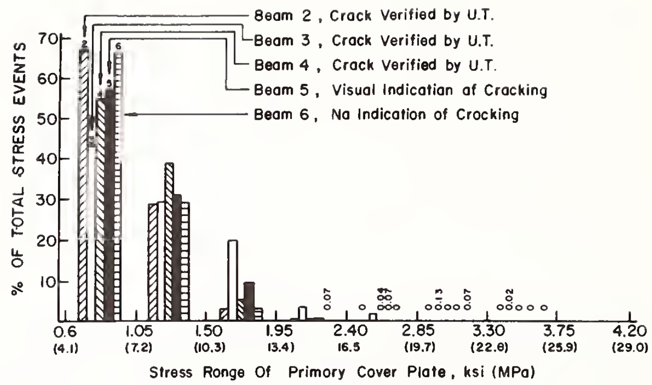

Fig. 9 Stress Range Histogram; West Ends of Eastbound Span 10, 1971

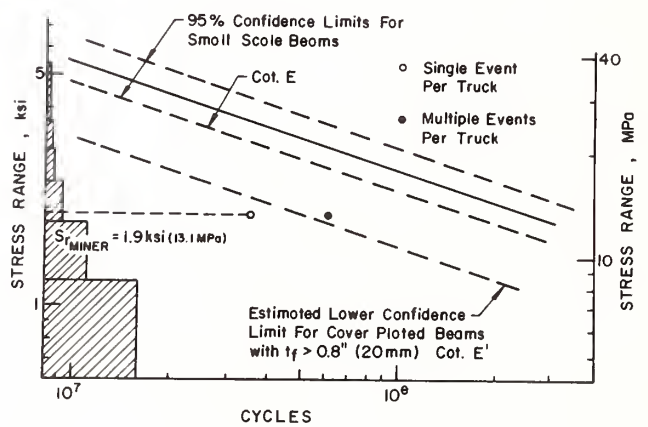

Fig. 10 Comparison of S_{rMiner} with Calculated Cycles and Fatigue Resistance of Beam Tests

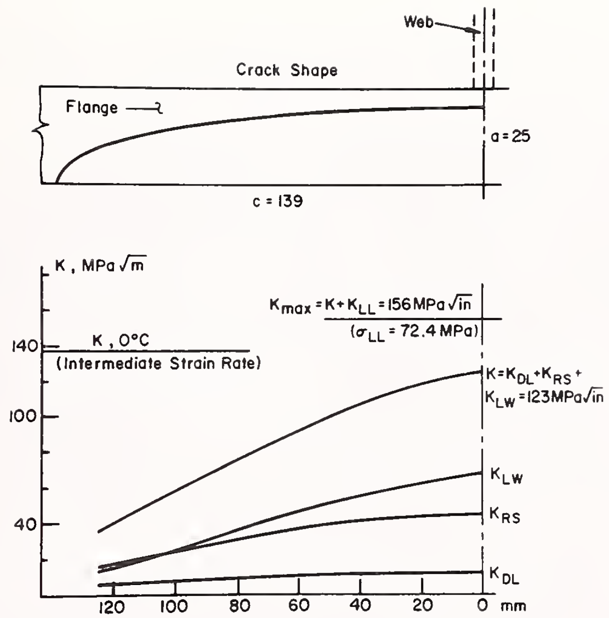

Fig. 11 Stress Intensity Factor for a = 25 mm, c = 139 mm

FAILURE ANALYSIS OF DAN RYAN RAPID TRANSIT STRUCTURE

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Abstract: On January 4, 1978, major cracks were discovered in three adjacent rigid steel frames supporting an elevated curved section of the Dan Ryan Rapid Transit.

Visual inspection of the cracks and subsequent microscopic examination of the fracture surfaces established that the fractures started at the welded junction of the bottom flange of girders that pierced the side plates of the frame. Fatigue crack growth was found in the junctions. Quality of the welding was poor, partly as a result of the geometry in the joint. However, the notch from the rough flame-cut slot was determined to be sufficient to initiate the fatigue cracking.

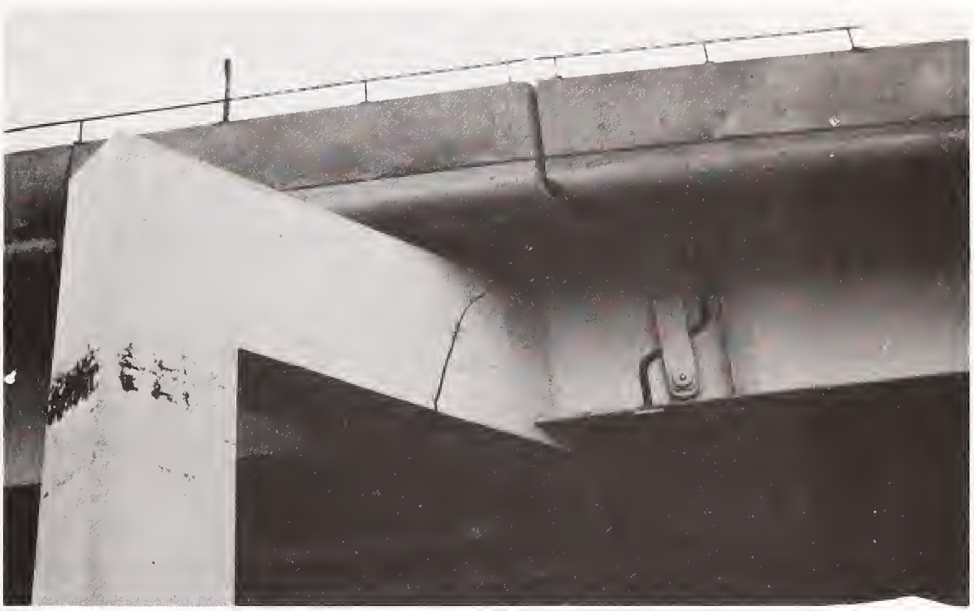
Physical tests were made to determine the appropriate stress intensity factor for the steel. This testing verified that the critical combination of a severe defect, high stress concentration, and cold temperature was sufficient to cause the fractures.

Key words: Failure; brittle fracture; fatigue; rapid transit; steel frames; welding.

INTRODUCTION

On January 4, 1978, a passenger in a train passing underneath the Dan Ryan Rapid Transit near 18th and Clark Streets in Chicago, Illinois observed a crack in one of the rigid steel frames supporting the elevated structure. These frames are commonly referred to as bents. Immediate action by Chicago Transit Authority (CTA) inspectors verified the existence of the crack in Bent No. 24, and also revealed cracking in two adjacent bents, Nos. 25 and 26. Photographs of the crack in Bent No. 24 are shown in Fig. 1.

Construction of shoring was started in the evening of January 4, 1978. The ground, which was solidly frozen at the time, was leveled with a layer of crushed stone. Mudsills, consisting of heavy timbers, were placed on the stone base on both sides of each cracked bent. Falsework towers were erected on the mudsills. As a precautionary measure, adjacent bents (Nos. 21, 22, and 23) of similar construction, were also shored.



a) Side of bent

Fig. 1 - Views of crack in Bent No. 24

b) Other side, showing
origin of fracture

Fig. 1 - (Continued)

On January 14, 1978, a series of test trains with progressively increasing loads and speeds were moved over the shored structure. The results of the test run were found to be satisfactory, and the system was returned to service on January 15, 1978.

A Technical Committee was formed to investigate the fractures and to develop recommendations for repair and retrofitting of the structure. The Technical Committee, under the direction of Commissioner Marshall Sulloway, included representatives of the Chicago Department of Public Works, Chicago Transit Authority, Regional Transit Authority, and DeLeuw, Cather and Company. Representatives on the Committee were assisted by several consultants, including Dr. John W. Fisher at Lehigh University. A report was published in January, 1979.(1)

BACKGROUND

The Lake-Dan Ryan Rapid Transit Line provides direct service between the center of Chicago and the south and west sectors of the city. On the Dan Ryan route, between 95th Street and 24th Street, the tracks are located in the median of the Dan Ryan Expressway and the future Franklin Street Connector. At 24th Street, the tracks ascend from the median onto a viaduct which extends to a junction with the old north-south elevated structure at 17th and State Streets. Bent Nos. 24, 25 and 26, where the failures occurred, are located west of Clark Street, directly south of 18th Street. At this location, the alignment of the tracks and structure is on a 400-ft radius.

The superstructure consists of four continuous and suspended plate girders with a cast-in-place concrete ballasted trough. Bents supporting the superstructure are constructed with two column legs and a horizontal member spanning between them. The bents have a box-shaped cross section. Stubs of the plate girders project 3 to 4 ft from the sides of the boxes and are connected with the intervening girders by a bolted or link connection. The top flanges of the stub girders pass over the top flange of the boxes, while the bottom flanges are welded to and pierce the boxes through slots cut into the lower part of the side plates.

Plans and specifications were prepared in accordance with criteria established by the American Railway Engineering Association "Specifications for Steel Railway Bridges", the American Welding Society "Specifications for Welded Highway and Railway Bridges", and the CTA "Design Policy for Elevated Railways". The live load used for design of the structure was the CTA Series 4000 Car for stress analysis and the Series 2000 Car for evaluation of allowable deflections. These cars have four axles, spaced 6 ft, 27 ft, and 6 ft apart, with 8 ft between the axles of successive cars. The axle loads are 24.5 kips and 17 kips for the Series 4000 and 2000 cars, respectively. However, trains composed of Series 2000 cars had been used exclusively on the Dan Ryan

Line since its opening. The design also considered the following additional loadings:

Dead Load:	Weight of track and ties (400 lbs/lin ft of track) plus weight of steel, concrete deck and track.
Impact:	As specified for electric locomotives.
Long. Force:	15% of live load without impact.
Wind and lateral forces:	450 lbs/ft minimum.
Wind on train:	300 lbs/ft on one track applied 8 ft above top of rail.
Nosing:	7000 lbs at top of rail in either lateral direction at any point of span.

Structural steel conforming to the "Tentative Specification for Structural Steel for Welding, ASTM Designation: A36" was used in the structure, with an allowable tensile unit stress of 20,000 psi.

Construction began in April, 1968 and was finished in June, 1969. The Dan Ryan Line was opened to traffic in September, 1969. On an average weekday, 467 trains pass over this viaduct. The trains vary in length from eight cars during the rush hours to as few as two cars at night.

EXAMINATION OF FRACTURES AND TESTING OF STEEL

The initial field examination of the fractures indicated that they originated at the welded junction of the girder flange tips to the box side plates. Chevron markings were observed on the fracture surfaces pointing towards the welded junctions. The fracture surfaces on Bent Nos. 25 and 26 appeared to be lightly rusted, while the fracture surface on Bent No. 24 was more heavily rusted.

During the course of the field examination, the top pin and one of two link plates connecting a girder to the stub extending from the west side of Bent No. 24 were found to be missing. Despite a search, these pieces were not found.

Measurements were made of the location and width of the cracks in each of the bents. The cracks penetrated the bottom flange plate and both side plates of all three bents, arresting at the edge of, or with only very little penetration into, the top flange plate.

In Bent No. 24, the crack extended through and also into the web of the girder. The trajectory of the crack across the bottom plate was inclined approximately 20 degrees to a normal to the longitudinal axis of the bent, and the crack developed a short spur on the west side just above the bottom plate, suggesting the presence of a torsional force component in the bent when the fracture propagated.

A test program was carried out on pieces of 3/4-in. thick steel plate taken from the sides of Bent Nos..24, 25 and 26, and a 1 1/2-in. thick plate taken from the bottom of Bent No. 25. The pieces taken from the sides of the bents included the origin of the fracture. These pieces are shown in Fig. 2.

Samples were cut from the pieces and used for the following tests:

1. Visual and microscopic examination of the origins of the fractures

2. Tensile tests

3. Chemical analysis

4. Charpy V-Notch tests

5. Compact tension tests

6. Metallographic examination

Examination

A sketch depicting the major features of the origin of fracture in Bent No. 24 is presented in Fig. 3. The area which was not welded is shown as the flame-cut surface. The vertical height of the flame-cut surface was about 3 1/4 in. A heavy multiple-pass weld was used to join the outside surface of the box to the girder flange. On the inside of the boxes, the weld was minimal. Paint was noted in gas holes and other unfused portions of the fracture surface.

A smooth fine-grained surface was observed along the outside of the weld on the outside of the box. Subsequent microscopic examination verified that this surface was the result of fatigue crack propagation through the weld. The major portion of the fracture surface, primarily above and below the area of the welded junction, is coarse grained and contains chevron markings, typical of brittle fracture.

A detailed microscopic examination was carried out on the fracture surfaces of Bent No. 26, and a limited examination on Bent Nos. 24 and 25, in order to establish the mechanism of crack growth and fracture. The piece removed from Bent No. 26 was more extensively examined because it

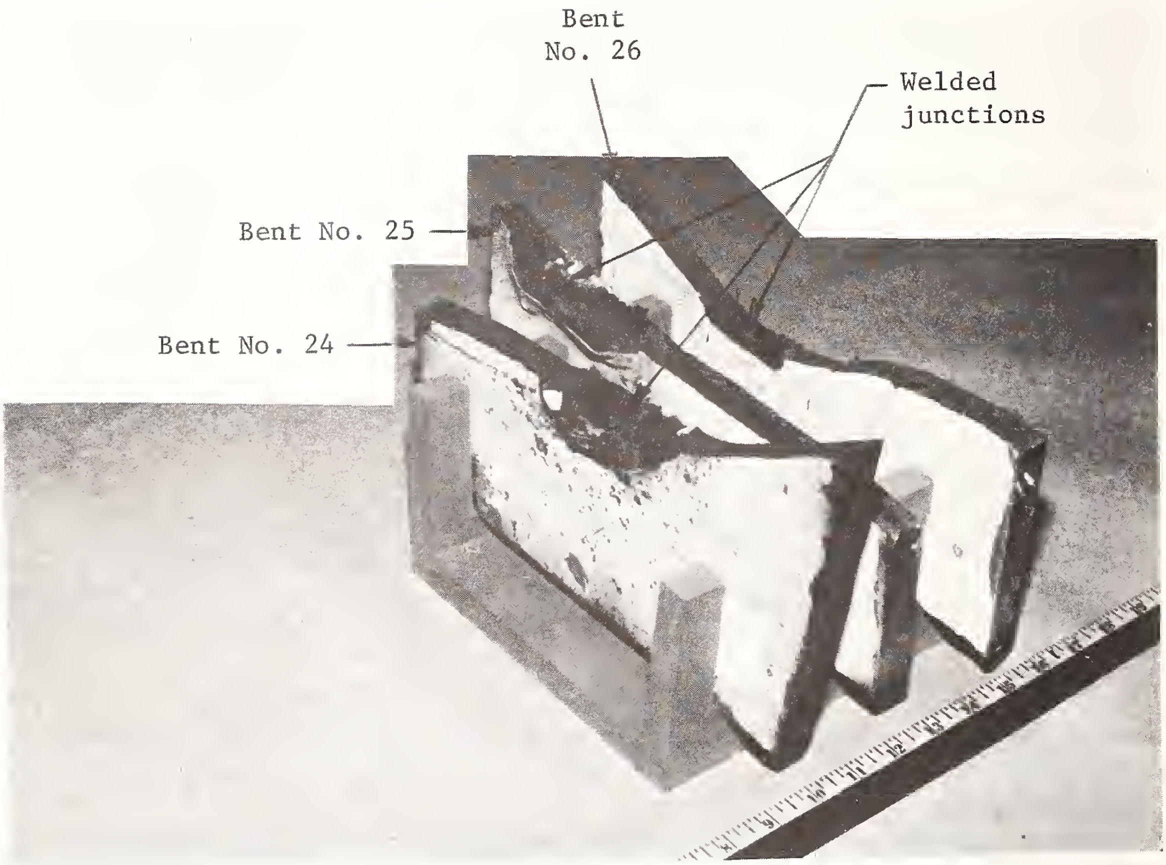


Fig. 2 - View of pieces cut from side plate
of Bent Nos. 24, 25 and 26

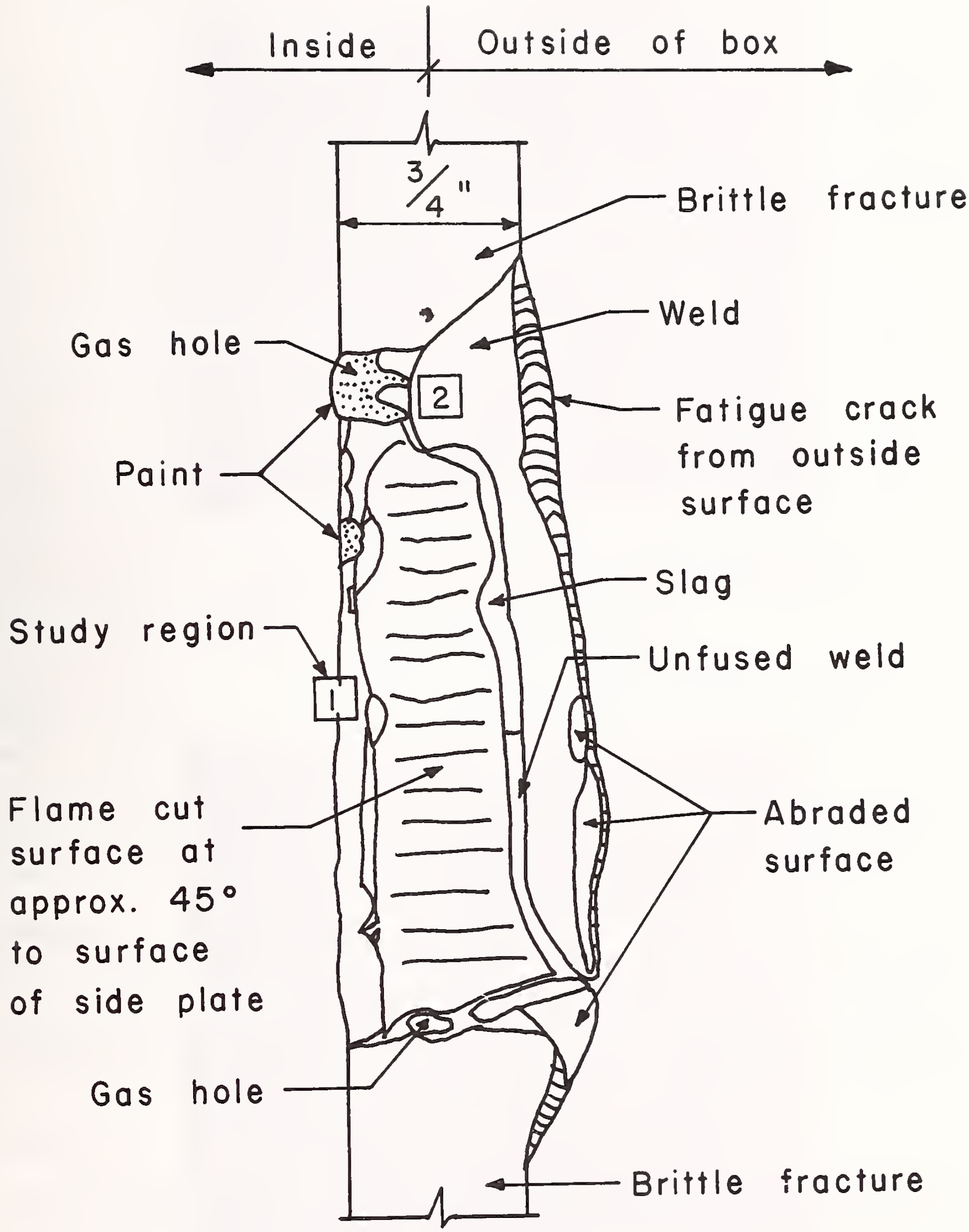


Fig. 3 - Schematic of origin of fracture in Bent No. 24

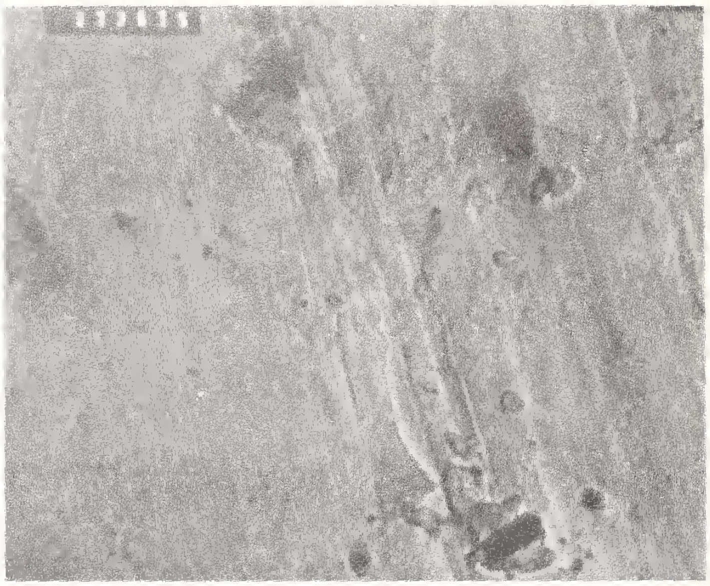


Fig. 4 - Fatigue crack growth striations
of Study Region No. 2, Bent No. 24
(By John Fisher, Lehigh University, X13,000)

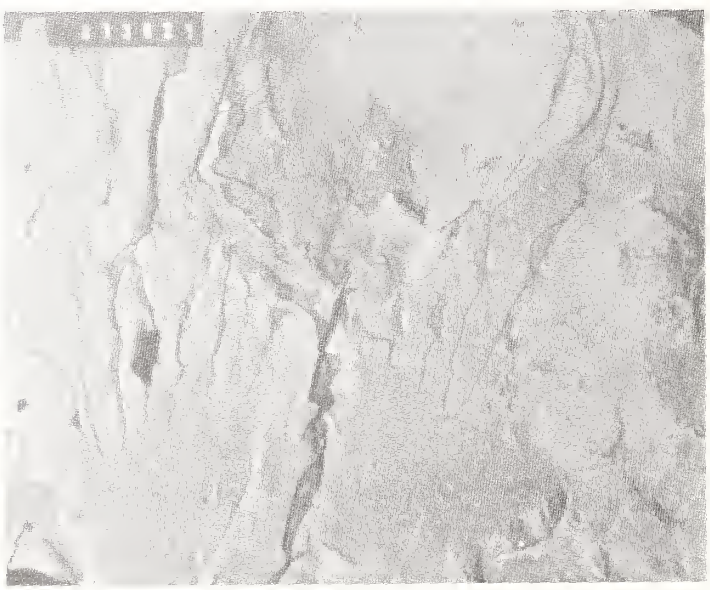


Fig. 5 - Cleavage fracture pattern at top
of Study Region No. 2, Bent No. 24
(By John Fisher, Lehigh University, X4000)

was not as abraded and dirty as the pieces from Bent Nos. 24 and 25. The examination was made by both scanning electron microscopy and transmission electron microscopy, by personnel at Lehigh University.

Replicas of the surface were made at selected Study Regions, shown for Bent No. 24 in Fig. 3. These replicas were examined with the transmission electron microscope. The examination of Study Regions 1 and 2 on Bent No. 24 indicated that fatigue crack growth initiated near the inside of the flame-cut surface and extended toward the outside. The growth was observed to extend at least 0.1 in. from the inside to outside in Region 1, and about 0.25 in. from inside to outside in Region 2. The crack growth rate was observed to be about 8×10^{-7} in. per cycle near the outside surface and increased to about 20×10^{-7} in. per cycle near the flame-cut inside region. Near the top of Region 2, evidence of cleavage fracture was apparent. Photographs of the fatigue striations and cleavage fracture are presented in Figs. 4 and 5.

Striations indicative of fatigue crack growth were observed in the Study Regions on Bent Nos. 25 and 26. The examination of the piece removed from Bent No. 25 also indicated that fatigue crack growth developed from the flame-cut area. These striations were observed to extend about 0.25 in. from their origin. The striations indicated a crack growth rate of less than 10×10^{-7} in./cycle. Thereafter, ductile dimples and cleavage facets were observed.

On Bent No. 26, the striations were most closely spaced near the outside surface. Toward the inside, the striations turned and tended to move down the side plate.

The scanning electron microscopy served to confirm the observations made by the transmission electron microscopy, but did not add substantial new information.

In summary, the examination of each origin of fracture showed that fatigue crack growth originated adjacent to the flame-cut areas and also at the exterior weld surface. The striations were all very finely spaced, indicating that very low rates of fatigue crack growth were experienced under cyclic loading. After the fatigue cracks extended and coalesced, a brittle (cleavage) type of crack extension developed.

Testing

Tensile strength, yield point and percent elongation in 2 in. were measured using round and flat coupons cut from the pieces taken from the bents.

Machining and testing of these coupons were performed in accordance with the methods prescribed by ASTM E8. The coupons were oriented so that the direction of fracturing in the test would be the same as the direction of fracturing observed in the plates. The steel was found to conform to the minimum mechanical property requirements of ASTM A36.

A chemical analysis was also performed on the steel. This analysis was made by the wet chemical method and atomic absorption techniques in accordance with ASTM E350 and other pertinent ASTM specifications. All samples conformed to ASTM A36 steel. The low silicon and aluminum contents indicated that the steel was not killed or deoxidized.

Charpy V-Notch specimens were machined from the test pieces. Three specimens from each piece were tested at each of the following temperatures: -50, -20, 10, 40, 70, 100 and 150°F (only two specimens from the piece from Bent No. 26 were tested at 150°F). The machining and testing were performed in accordance with ASTM E23. The results of these tests are shown in Fig. 6. They show good toughness with impact energies greater than the present American Association of State Highway and Transportation Officials' (AASHTO) minimum requirement of 15 ft-lbs at 40°F for Zone 2. Minimum service temperatures in Zone 2 are greater than -30°F but less than -1°F. Transition temperature behavior for this steel increases rapidly in the range between +40 and +70°F.

Compact tension tests were performed to obtain a measure of the critical stress intensity factor, K_c, which would produce a brittle fracture under the conditions existing at the time the cracking occurred in the bents. The specimens for these tests were cut from the piece taken from Bent No. 25, and tested in general accord with ASTM E399.

Three specimens were tested at each of the following temperatures: -30°F, -5°F, and +30°F. The method of loading produced a fracture in the specimens tested at -30°F and -5°F in about 1 second. Because of the increased toughness of the specimens tested at +30°F, a loading time between 2 to 4 seconds was required to cause fracture.

It was anticipated that valid K_{Ic} values, the critical stress intensity factor determined in accordance with ASTM E399, might not be obtained with specimens taken from the 3/4-in. thick side plates. Therefore, displacement was measured in two ways: at the mouth of the opening using an oscillograph, and also in line with the loading pins using an x-y recorder.

In all cases, cracking occurred at the maximum load. Of the nine specimens tested, one set of data was lost because the displacement gage ran out of travel before the specimen fractured. None of the load versus mouth opening displacement curves met the ASTM E399 requirement for linearity, and therefore, the measured displacements in line with the loading pins were used to indirectly obtain K_{Ic} by the J integral method.

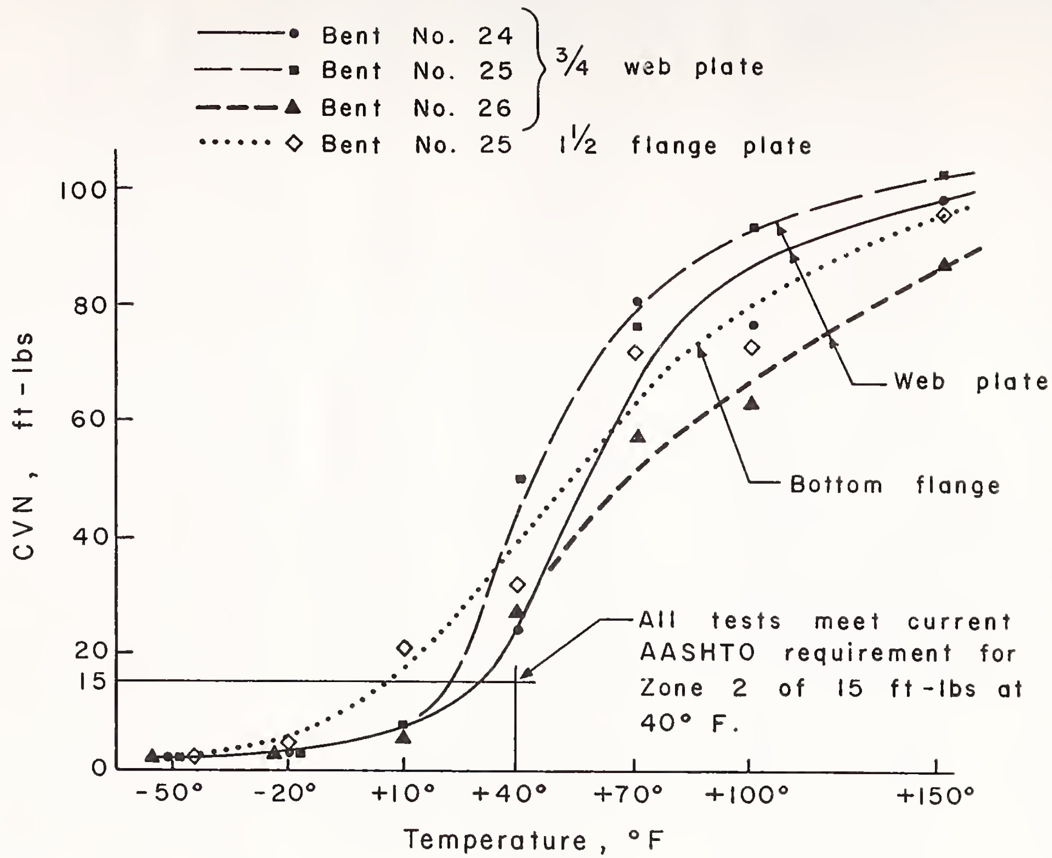


Fig. 6 - CVN Test Results (By Materials Research Laboratory, Inc., Glenwood, Illinois)

A plot of the K_{Ic} values versus test temperature is presented in Fig. 7. It is evident that there is a significant transition in the toughness of the steel at about 0°F. For the purposes of this investigation, the measured values of K_{Ic} are considered to be a conservative but realistic estimation of K_c, the critical stress intensity factor associated with the conditions causing the cracking in the piers.

A metallographic examination did not reveal any defects in the steel. The appearance of the grain structure near the surface was considered to confirm both the fatigue and fracture modes of crack propagation.

STRESS ANALYSIS

In the design of the Dan Ryan structure, a three-dimensional space frame analysis was used in order to include the effect of the curvature of the girders. The space frame model consisted of two bents, the four superstructure girders framing between bents, the stub girders cantilevering from the bents to the hinge points and the diaphragms between girders. The drop-in spans between hinges were analyzed separately as grids. Reactions determined from the grid analysis were applied to the space frame.

Various loading configurations on both single and double tracks were used to arrive at the maximum stress conditions. The magnitude of tensile and compressive stresses in the bents and girders was on the order of 20,000 psi. Magnitude of shear stress was 12,500 psi. These levels of stresses were permitted by the specifications of the American Railway Engineering Association in effect at that time.

Following the discovery of the cracks, the original designer undertook an independent investigation of the stresses in the bents, using an advanced three-dimensional space frame analysis.

Stresses in all bents under the Series 2000 Car loading were found to be less than the allowable stresses, with the exception of Bent No. 24 adjacent to the girder where the fracture occurred. At this location, the reanalysis indicated the side plate where the fracture originated was overstressed in shear under dead load, live load (two tracks), impact and centrifugal force. Specifically, the allowable shear stress was 12.5 ksi, and the calculated maximum shear stress was 14.9 ksi. The tensile stress range at this location for live load on two tracks plus impact and centrifugal force was approximately 3.2 ksi.

With the exception of the moderate overstress in shear of the web of Bent No. 24, the results of the reanalysis confirmed the original stress analysis.

In an effort to more accurately assess the stress conditions near the origins of the fractures, a computer analysis was performed by the

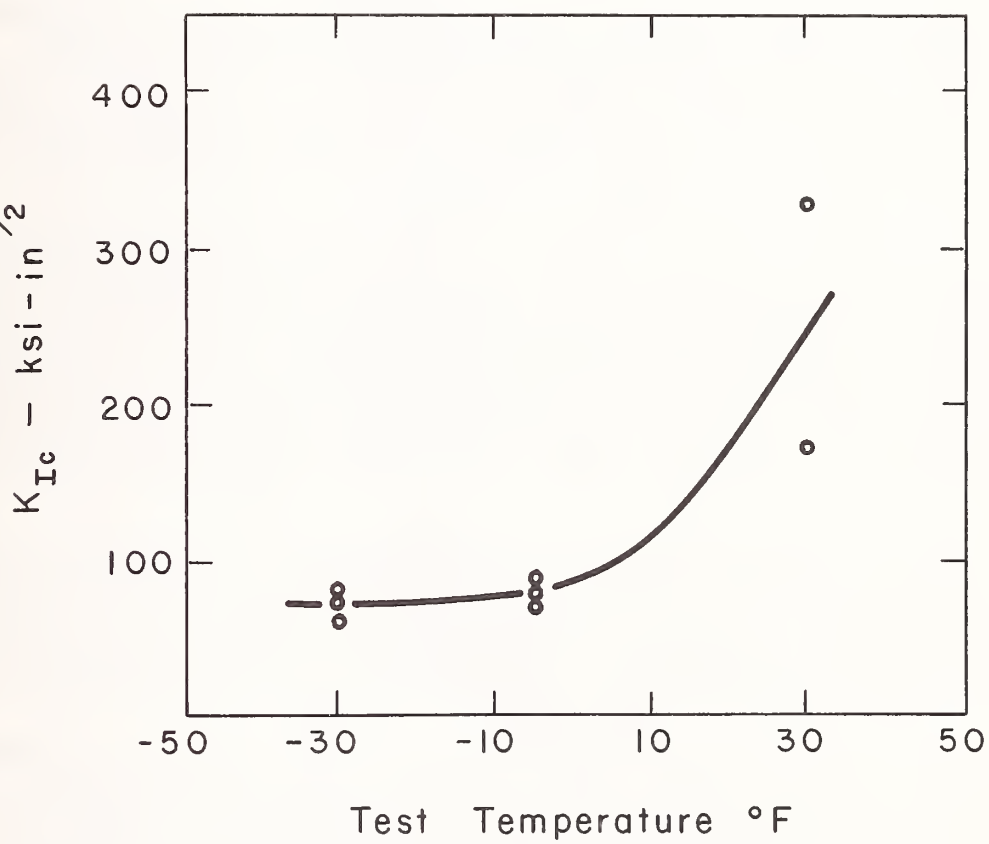


Fig. 7 - K_{Ic} Test Results (By Materials Research Laboratory, Inc., Glenwood, Illinois)

original designer using the NISA (Numerically Integrated Elements for System Analysis) program developed by the Engineering Mechanics Research Corporation.

Because of the large size of the model, an attempt to minimize the strain energy was not made. Therefore, the numerical values of displacements and of stresses obtained were regarded as approximate. The welded joints between the box side plates and the bottom flanges of the four curved girders were assumed to be completely fused; i.e., the thickness of all elements was taken as constant throughout the region of each element. Similarly, constant elastic properties of the material were specified for all elements.

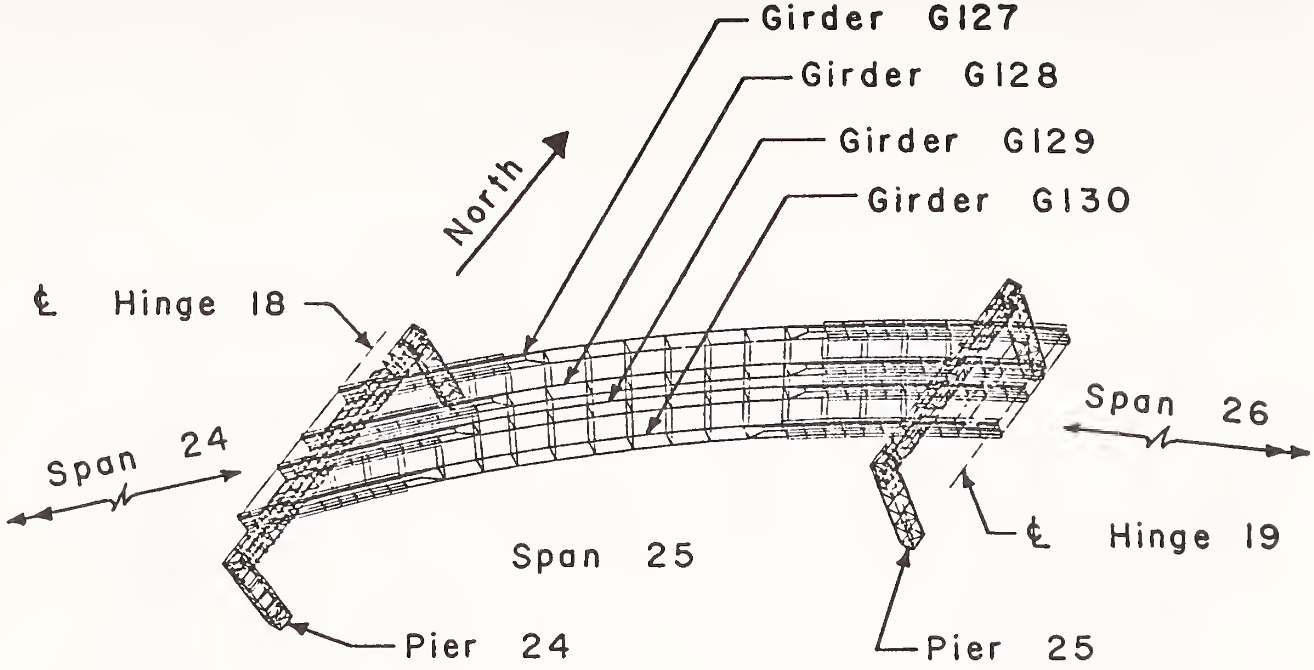
The structural model for the initial set of computations consisted of Bent Nos. 24 and 25, and the girders extending from Hinge 18 to Hinge 19, as illustrated by Model A in Fig. 8a. The model contained the same structural elements and support conditions as that used in the original analysis, to allow comparison of results. Model A consisted of 1583 four-node shell elements defined by 1590 corner nodes.

Uniformly distributed dead load was simulated by statically equivalent loads applied at the nodes along the top rows of girder web elements. Additional concentrated loads were applied at the end nodes of all four girders to account for the weight carried by the hinges in Span Nos. 24 and 26. Live load node forces were equivalent to train wheel loads (without impact). Two trains were placed in a position that leads to maximum tensile stress at the bottom flange of Bent No. 24 at the intersection with the exterior girder. The Series 4000 Car was used for live load.

Following the analysis of Model A, a second model representing the area of the intersection of the exterior girder with Bent No. 24 was prepared, as shown in Fig. 8b. Model B contained 165 four-node shell elements and 180 corner nodes. The boundary conditions specified for Model B were the displacements obtained from the analysis of Model A.

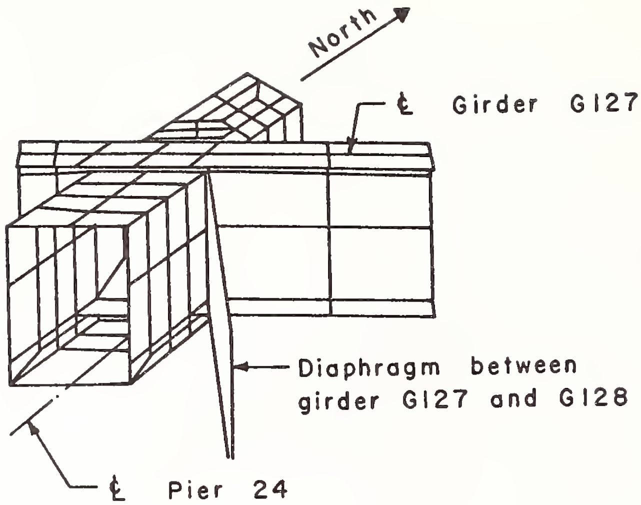
Analysis of the deformations in the area of the intersection between the box side plate and the bottom flange of the girder indicated that stress concentrations due to out-of-plane bending of the side plate are present. In order to determine the order of magnitude of these stresses a third model was prepared. Model C was programmed with a finer mesh than Model B, as illustrated in Fig. 8c.

Model C is a detailed representation of the side plate of Bent No. 24 in the vicinity of and including a segment of the bottom flange of the girder. This model contained 212 elements defined by 571 nodes. The elements used in this model are eight- and six-node isoparametric shell elements. All nodes along the boundaries of Model C coincide with the corresponding nodes of Model B. The displacements of these boundary

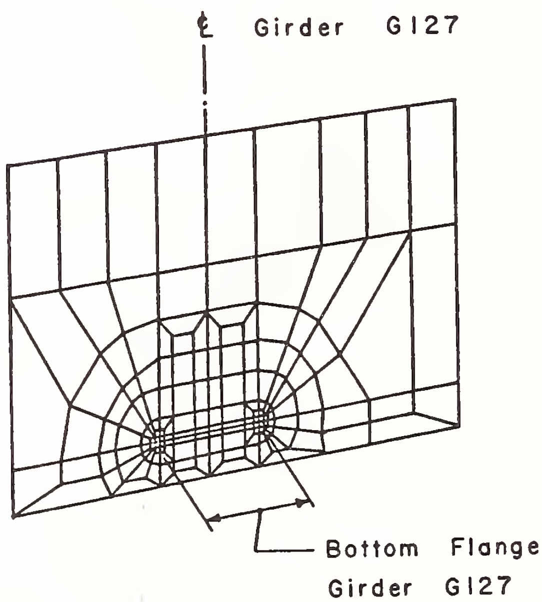


a) Model A

Fig. 8 - Finite element models (By DeLeuw Cather & Company)



b) Model B



c) Model C

Fig. 8 - (Continued)

nodes obtained from the analysis of Model B were used as the prescribed boundary conditions for Model C.

Stresses acting at the middle surface, at the outside face, and at the inside face of the web were obtained from the finite element analysis. The membrane stresses acting at the middle surface of the side plate were low (less than 10 ksi), even at the level of the girder flange. However, theoretical peak stresses due to out-of-plane bending of the side plate exceeded the nominal yield stress for A36 steel, both in compression on the outside face, and in tension on the inside face. The live load stresses were, in general, lower than the dead load stresses. Still, peak stresses in excess of 20 ksi were indicated. The peak live load stresses are of the same sign as the corresponding dead load stresses so that stress reversal does not occur.

DISCUSSION AND FINDINGS

Following the occurrence of the fractures in Bent Nos. 24, 25 and 26, an intensive review of the design, a careful inspection of these and other bents and piers, and a thorough testing program on representative samples of the steel were performed. The results of the design review and the testing program were summarized in the preceding chapters.

Substantial variation was found in the quality of welding of the junctions between the girder flanges and bent side plates in Bent Nos. 21 through 26, and the details of some joints did not conform to the details shown on the shop drawings. Numerous welding deficiencies were apparent. Cracks believed to be caused by fatigue were found in at least one junction in Bent Nos. 21 and 22 and in junctions other than where the fractures originated in Bent Nos. 24, 25 and 26. In Bent No. 23, the girder flange bears on top of the bottom plate of the bent, creating a different condition than in the other bents. No cracks due to fatigue were found in Bent No. 23.

Brittle fracture of structural steel members usually occurs when a critical combination of defect, stress intensity, and cold temperature exist simultaneously.

In Bent Nos. 24, 25 and 26, the defect was present in the form of a condition at the junction of the girder flange and bent side plate which was very much like a "built-in crack". It was a difficult area in which to weld, particularly in Bent No. 24 where the girders are skewed at an angle of approximately 45 degrees with respect to the box.

The analysis verified that high stress levels would be present at these locations, under both dead load and repeated applications of live load. High residual stresses due to welding would also be present. It should be noted that calculated and residual stresses which exceed the yield point will be redistributed by yielding of the steel. Thus, the cyclic train loading in a highly sensitive area caused fatigue cracking, which over a period of time increased the size of the defect and created a severe stress concentration at the tip of the crack.

Prior to the discovery of the fractures, temperatures in the Chicago area were extremely cold on several occasions. There were five occasions when the temperatures dropped below 0°F and the coldest temperature of the winter season to that time, as recorded at Midway Airport, was -7°F on December 10, 1977.

It seems likely that the first fracture occurred in Bent No. 24. The analysis shows that the level of shear stress was higher at the point of origin in this bent than in Bent Nos. 25 and 26. A link was missing on one of the girders located adjacent to Bent No. 24. The influence on the fracture caused by the missing link is not known, but regardless of the disengagement of the link, conditions that would cause fatigue cracking were present. Visual observations after the fractures were discovered also indicated that a heavier coating of rust was present on the fracture surface of Bent No. 24 than the other two bents.

The laboratory testing confirmed that the combination of stress concentration at the point of origin, fatigue cracking, and cold temperatures was sufficient to cause the fractures. However, the steel was found to meet current toughness requirements recommended by the American Association of State Highway and Transportation Officials for steels used in bridges.

It was concluded that the extremely severe stress concentration at the junction of the girder flange and bent side plate was the primary cause of the brittle fractures which occurred in the Dan Ryan Rapid Transit structure.

REFERENCE

(1) Final Report on Causes of Fractures in Bent Nos. 24, 25 and 26, Dan Ryan Rapid Transit, Department of Public Works, Chicago, 1979.

ACKNOWLEDGEMENTS

This paper is based on the report prepared by the Technical Committee and published in January, 1979. The Technical Committee consisted of:

Marshall Suloway
Commissioner of Public Works

Department of Public Works:	Louis Koncza, Chief Engineer
Chicago Transit Authority:	Thomas L. Wolgemuth, Mgr., Engineering
Regional Transportation Authority:	Dan Brescia, Mgr. of Engineering Department
DeLeuw, Cather & Company:	William G. Horn, Staff Vice President

The Technical Committee was assisted in the conduct of the investigation and preparation of their report by:

Department of Public Works:

George Eng	Deputy Commissioner
John M. Hanson	Consultant, Wiss, Janney, Elstner & Associates, Inc.

Regional Transportation Authority:

B. J. Ford	Division Director, Transportation
Don Buck	Director of Safety
John W. Fisher	Consultant, Lehigh University
Anton Tedesko	Consultant
Hannskarl Bandel	Consultant

DeLeuw, Cather & Company:

G. M. Randich	Senior Vice President

The examination of the fracture surfaces and material testing was performed by Wiss, Janney, Elstner and Associates, Inc., Lehigh University, Materials Research Laboratory, Inc., and Taussig Associates, Inc. The analysis was made by DeLeuw, Cather & Company.

BRIDGE WELDING AND FRACTURE CONTROL

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ABSTRACT: In the 1970's, at least twenty bridge girders suffered brittle fracture and over a hundred girders were found to be cracked; fortunately (or providentially) there was no catastrophic collapse. However, with an apparently increasing incidence of weld cracking and fatigue cracking, the probability of a catastrophic collapse occurring in the 1980's is very real indeed.

The range of occurrence of weld cracking in steel bridges has reached alarming dimensions. Why? Is there a problem with the design of welded connections? Is there a shortage of qualified welding engineers? Is there a problem with quality control and quality assurance in fabrication? Is there a people problem and/or deficiencies in the flaw-detection techniques? Is there a need for nondestructive testing engineers? Is there a base-metal weldability problem? Is there a problem with cosmetic corrections and weld repairs? The answer is an emphatic yes to each of these problems and more!

The situation is so serious that without prompt positive action on the part of all concerned, there will be more and more bridges made of concrete even where steel makes better engineering sense.

The paper takes a second look at some of the above problems using bridge failures from the 1970's as examples and then suggests some changes that will have to be made in the 1980's if the problems we now face are to be solved.

KEY WORDS: Fracture control; nondestructive inspection; quality control; welded steel bridges.

STATE-OF-THE-ART TECHNOLOGY, BRIDGES

Cracked welded-plate-girder bridges. The incidence of brittle fractures and weld cracking in steel bridges has reached alarming proportions in the 1970's. Over twenty bridge girders have developed brittle fractures, and in-field inspections of bridges under traffic have revealed hundreds of rejectable indications using ultrasonic testing (UT). Millions of dollars have been spent in bolt-splice repair of defective butt welds in fracture-critical members. The

Silver Memorial Bridge at Point Pleasant, West Virginia, the Paducah Bridge on I-24 between Kentucky and Illinois, and the Fremont Bridge in Portland, Oregon, are among the major bridges known to have UT-rejectable indications at the time of this writing. Intensive investigations are in progress to determine why and when the cracking occurred.

In just one State, inspection of 44 two-girder bridges with intermediate floorbeams and stringers revealed 24 bridges with cracked members. The cracks are occurring near the top flange at the floor-beam-to-girder connection plates; the general problem is believed to be out-of-plane deformation in the girder webs caused by end moments in the floorbeams. In the 24 cracked bridges, there were 290 cracked locations in negative-moment areas.

In one bridge which in 1973-74 had been given "in-depth" inspection including UT, in 1977 was found to contain 132 UT-rejectable indications of length ¼-in. or greater in 51 flange groove-weld butt splices. Many of the indications have since been confirmed to be weld cracking and the fact that many of the cracks were found in compression members as well as tension members suggests that the cracks were in the welds at the time of fabrication. Investigation is continuing. Another bridge contains a great number of fatigue cracks at the toe of fillet welds connecting cover plates to flanges. Some of the cracks were as long as 2 in. and over ½-in. deep as determined by UT. The recorded stresses were small (RMS stress range only about 1.3 ksi); a tremendous traffic count is said to explain the fatigue-crack growth. But there may be another problem which has not been adequately addressed - premature initiation of fatigue crack growth as a result of weld-toe cracking occurring at the time of fabrication and undetected by quality control (QC) or quality assurance (QA). Microcracking at the weld toes intensifies the stress-concentration factor inherent in the toe geometry and is exceedingly difficult to detect in nondestructive inspection.

Why have there been over twenty brittle fractures in bridge girders in the 1970's and why has there been so much cracking in bridge girders in the 1970's? Why are so many cracks found after the bridges are erected and under traffic? These and related questions are examined in the following paragraphs using as examples some actual bridge problems from the 1970's.

<u>The base-metal weldability problem</u>. Consider two examples. The Bryte Bend Bridge, Sacramento, California, June 13, 1970, developed brittle fracture of a main-load-carrying tension flange. The flange that fractured came from a heat of steel (C4913) that occurred in ten locations in the bridge. The steel was ASTM A517 grade H (modified), 2-¼ in. thick. Not only was the steel seriously lacking

in toughness but also it was highly susceptible to hot cracking. It was a weld hot crack that initiated slow crack growth during shop handling and erection of the girder. Subsequent inspection revealed numerous cracks in the standing bridge where cross bracing was welded to the A517-H flanges. There was also a serious cracking problem when the fabricator attempted to qualify his welding procedures by test; transweld cracking occurred in the submerged-arc-welded (SAW) butt splices, cracking that sometimes ran several inches into the base metal.

A second bridge containing the same steel, the Silver Memorial Bridge at Point Pleasant, West Virginia, replacing the bridge that collapsed killing 46 people in December 1967, has experienced serious weld cracking problems. This bridge is highly significant to the question of whether base-metal weldability is a problem. Two companies were contracted to fabricate the truss members for this bridge; the fabricator on the West Virginia side of the river used A514 grades F and H from the same steel producer supplying steel for the Bryte Bend Bridge, and the fabricator for the Ohio end of the bridge used A514 grades F and H from a different steel producer. Ultrasonic (UT) inspection of the groove-weld butt splices revealed numerous UT-rejectable indications, some so large that over a million dollars in bolted-splice-plate retrofitting was deemed necessary by the State of West Virginia. The statistics of defect location and incidence rate clearly showed that the A514 grade H chemistry as supplied by one of the steel producers was causing a significantly greater amount of weld cracking than the same ASTM type and grade supplied by the second producer. Clearly there was a weldability problem with the A517 grade H Steel supplied by one of the producers. The AWS D1.1 Structural Welding Code lists A514/517 as prequalified steel irrespective of grade and, therefore, under the AASHTO bridge specifications there is no requirement for weldability testing this steel.

<u>The design-of-welded-connections problem</u>. Consider two examples. When the tied arch at Paducah, Kentucky was completed, long behind schedule, weld cracking in the form of lamellar tearing had been so costly for the fabricator, he sued the State of Kentucky claiming over a million dollars in extras. The court found the design to be a major contributor to the problem and assessed damages at 80:20 in favor of the fabricator. In other words, the court found that 80 percent of fault was with the design and since the designer was an agent to the State, the State had to assume 80 percent of the cost. The remaining 20 percent was found by the court to be the result of bad welding. This problem was discovered before the bridge was opened. Now, just a few years later, the bridge is closed at the time of this writing, having been found to contain numerous UT-rejectable planar-type indications in the A514 tie-girder groove-weld butt splices and corner joints. The bridge has been in service less than 10 years.

When the Lafayette Street Bridge in St. Paul, Minnesota developed a brittle fracture, failure analysis revealed a gusset-to-web-to-stiffener connection which according to the contract plans was to be welded from both sides and without coping, so that the three welds were intersecting. With the gusset located only six inches above the flange, the fabricator found the joint design impossible so he used a backing bar, and then failed to get complete penetration. The ensuing fatigue cracking in the gusset-to-stiffener connection entered the girder web and triggered brittle fracture.

The "cosmetic-corrections" and weld-repair problem. Bridge after bridge with weld cracking has been found to have weld repair associated with or directly causing the cracking. The first crack found in the Silver Memorial Bridge was approximately 1 by 4 in. in a multiple weld repair. At least two repairs were made in the location that cracked. The crack that initiated brittle fracture in the Neville Island Bridge in Pittsburgh was in a massive repair of an electroslag weld. When weld repairs crack it should not be suprising considering the high restraint that is inherent in a local excavation, and considering the poor welding techniques often used, sometimes without knowledge of QC management.

"Cosmetic corrections" are particularly insidious because it is common practice for the corrections to be made without preheat and without benefit of nondestructive inspection. Often the welder decides what is to be done and how it is to be done. Consider, for example, minor undercut sometimes even within the limits acceptable under the Structural Welding Code (AWS D1.1); if the welder chooses to make the "cosmetic correction", a low-energy-input stringer bead is typically used with perhaps 30,000 J/in. and no preheat. If the flange happens to be 2 or 3 in. thick, that makes no difference to the welder, but to the steel it can mean untempered martensite. Then there are "cosmetic corrections" to oxygen-cut flange edges; if a gouge is produced in cutting, this is sometimes corrected by a short (tack-like) stringer bead across the edge of the plate. Again, this is typically done with low-energy input and without benefit of preheat. If sabotage were the intent, it could not be more effective! Since it is not a "repair weld", it may be done without knowledge of QC/QA. Such welding is likely to result in toe cracking and where the weld is transverse to the principal stress, the toe cracking is likely to initiate premature fatigue cracking and/or brittle fracture. Unfortunately, there are advantages in the concept of "cosmetic corrections," namely no paper work and no delays with inspection, preheating or post heating, so it will not be easy to eliminate the practice.

In the AASHTO Guide Specifications for Fracture Critical Nonredundent Steel Bridge Members, September 1978, there are several provisions specifically on repair welding. Repairs are placed in two categories: noncritical and critical. When repair involves removal of weld or base metal, surveillance by QC/QA is essential. Engineering decisions are involved as to the orientation of the excavation, the choice of electrode, the joint configuration (including angle and root opening), stringer-bead placement and preheat/interpass/postheat temperature. All too often when little attention is given to these considerations, weld repairs crack, necessitating multiple repairs in a given location. When hydrogen is the cause of cracking, if the repair itself does not crack, it is not unusual to find additional cracks nearby in weld locations previously accepted by QC/QA. Weld repair and cosmetic corrections are a major problem, as proven by cracks found in standing bridges.

The quality-control/quality-assurance problem. QC is virtually nonexistant in some shops and QA is often a passive hands-in-the-pockets operation, underpaid, understaffed, and underskilled. Standing bridges with cracking or brittle fracture always raise the question: When did the cracking initiate, was it in the fabrication shop or was it the result of fatigue? To answer the questions one should be able to refer to the fabrication QC/QA inspection records. Case after case can be cited where inspection was by radiography but the film was so poorly exposed or so poorly processed that it is virtually useless. Where was the QA inspector who was responsible for reviewing and approving the film? The advantage, we are told, in radiographic testing (RT) is in having a permanent record, but what good is the record if the techniques are so poor that the record cannot be read? In the case of one bridge that suffered brittle fracture, the RT files were lost. In another case, the file was found and rejectable indications were visible in the film; apparently no one had read the radiographs.

Then there are numerous cases where AWS D1.1 has been violated and apparently no one, neither QC nor QA, saw the violation. AWS D1.1, Section 9 specifically prohibits tack welds left in the finished bridge. The tie-girders of a large tied-arch bridge have recently been found to contain man holes cut in the webs of the tie girders where in closing the manholes after erection, a backing bar was tack welded on the inside of the box. The opening was closed in the final stage of erection. AWS D1.1, Section 9 requires that backing bars be spliced and inspected so as to assure that there will be no cracked groove-weld butt joints. The tie girders of a large award-winning tied arch under traffic have been found to contain numerous cracked butt splices in the steel backing corner welds. The defective welds were in violation of AWS D1.1 from the day they were welded. Where was QC/QA in that bridge? The Fort Duquesne Bridge in

Pittsburgh has a problem with lamellar tearing, together with groove-weld toe cracking, sometimes across the entire 48-in. width of flange. The design called for a cruciform joint configuration in 2-1/2 in.-thick ASTM A517 steel. The lamellar tearing can be seen by any pedestrian walking the bridge at the time of this writing. Why didn't QC/QA see the cracking at the time of welding?

In the 1970's, a footnote in the Structural Welding Code (footnote 27 page 73 of AWS D1.1-75, or page 103 of AWS D1.1-79) placed responsibility for inspection on the owner. For some members of the AWS D1.1 Committee, it was incredible that the fabricator should not be held responsible for continuing evaluation of the quality of his work, and yet under footnote 27 the "Inspector" was an agent of the owner. Moreover, in the 1970's there was no distinction between QC as a responsibility of the fabricator and QA as a prerogative of the owner. For reasons not altogether clear to the author, the prolonged arguments against clarifying the QC/QA functions included adament rejection of the terms QC and QA in spite of the fact that in other industries the terms are commonplace and well defined. Anyone interested in the compromise reached by the AWS D1.1 committee needs only turn to AWS D1.1-80 (1980) paragraph 6.1. A few State Departments of Transportation (DOT) including New York and California have long rejected the provisions of AWS D1.1 in this regard; in November 1979, FHWA issued a Technical Advisory clearly defining the roles of QC and QA, recommending that QC be specifically the responsibility of the fabricator and QA the prerogative of the owner (the State DOT). Moreover, it was recommended that QC for fracture-critical members (FCM's) include both RT and UT as complementary test methods, and that QA no longer be a passive watch-dog operation but use hands-on auditing of QC by UT.

In the late 1970's, intensive reinspection of several bridges under traffic containing fracture-critical members (FCM's) revealed instances of both cracking and rejectable workmanship defects (lack of fusion, slag and porosity) at the ends of submerged-arc-welded (SAW) groove-weld butt joints. Many of the bridge members had been subjected only to RT. In the 1970's AWS D1.1 had no provision for minimizing scattering off the cassette into the image area, which tends to obscure defects at the plate edge; when the end of the butt splice corresponds to the start of welding, there appears to be a greater probability of defects, perhaps associated with inadequate or improper starting tabs or preheat. Linear-elastic fracture mechanics shows that the stress intensity associated with an edge crack can be dangerously high and, therefore, is likely to initiate premature fatigue crack growth and/or brittle fracture.

Another type of cracking seriously neglected by QC/QA and yet dangerous from the standpoint of promoting premature fatigue crack growth is fillet-weld toe cracking. This type of cracking is insidious because inspection for such cracking is usually only done by magnetic particle testing (MT) which may produce a buildup of iron powder at the geometric discontinuity inherent at the interface of the plate surface and weld crown. Dye-penetrant and/or eddy-current inspection would be far more effective for detecting weld toe cracking but are seldom used. MT with prods must be done before painting (bare metal contact); both MT with prods and AC-yoke MT must be done in two directions if both toe cracking and transverse weld cracks are to be detected.

The critical-crack-size/nondestructive-testing problem. If linear elastic fracture mechanics (LEFM) is to be utilized, nondestructive testing must be capable of measuring both the length and depth of any given indication. Among the weld defects that are of greatest concern are the planar defects such as lack of fusion, incomplete penetration and cracking. The current AASHTO Guide Specifications for Fracture Critical Nonredundant Steel Bridge Members (September 1978) specifies Charpy V-notch impact minimum values for qualifying the welding procedures used in all weld joints in "fracture critical" members. The Charpy requirements for weld metal joining A36, A588, and A572 steels is 25 ft-lb at minus 20°F and for weld metal joining A514/517 steel is 35 ft-lb at minus 30°F.

Based on the Barsom/Rolfe relationship

$$K_{Id}^2 = 5(CVN)E$$

where CVN is the weld-metal Charpy ft-lb value and E is Young's modulus, one can estimate the critical stress intensity corresponding to the specified Charpy values, and from the critical stress intensity estimate the critical crack sizes. Table 1 gives the calculated flaw dimensions for two flaw shapes when the planar indication is embedded in the weld. Note that the critical depths in the weld-residual-stress field are less than one inch. With the ultrasonic testing techniques presently used by the bridge-building industry, there is a serous problem in quantitatively sizing a flaw when the flaw depth is less than the UT-beam width.

Without the ability to measure both the length and depth of a given planar indication, it is not possible to utilize LEFM. Furthermore, even with quantitative flaw sizing, without a realistic estimate of the largest flaw that may escape detection in nondestructive testing, it is not possible to specify the required toughness. Furthermore, to allow for possible error in flaw measurement and to allow for some crack growth before instability, the nondestructive techniques and personnel skills must assure that the largest flaw escaping detection is no more than say 50 percent of the critical crack size. To assure that defects will be found that are smaller than the critical crack depths listed in Table 1, either greater toughness must be specified or nondestructive testing techniques and personnel skills will have to be improved.

It is the shockingly large cracks that have been found in standing bridges that attest to a problem both with nondestructive testing and welding as presently conducted.

10-YEAR FORCAST

Can the incidence of weld cracking during and after fabrication be reduced? To do so during fabrication will require the services of a welding engineer and careful consideration of sometimes neglected but vital details such as (1) cracked tack welds (weld metal and/or weld heat-affected zone), (2) consistent implementation of low-hydrogen practice, (3) control of "cosmetic corrections" and weld repairs, (4) control of cooling rates to preclude both brittle weld-transformation products in the heat-affected zone and loss of toughness in the weld metal, (5) use of welding consumables that provide tough, crack-resistant weld metal, (6) control of welding procedures to minimize planar-type defects, (7) designer review of welded connections at the shop-drawing stage, and (8) attention to the inspectability of welded connections.

And to markedly reduce the incidence of cracking after fabrication it will be necessary to have the services of a nondestructive testing engineer (at least an ASNT Level-III person) and qualified ASNT Level-II inspectors capable of finding and sizing buried weld defects.

To realize these requirements in the 1980's, some changes will be necessary. For the last two decades our colleges and universities seem to have been oblivious to the need for solidly educated welding engineers and welding metallurgists.

Attempts to get educators' attention by published articles comparing U.S. and USSR curricula and the numbers graduated in welding engineering has been of no avail. With an average of only about 20 BS welding engineers in the USA per year for the last 20 years, as compared with 2000 each year in the USSR, and currently about 800 majoring in materials science and metallurgical engineering each year in the USA and Canada as compared with 8000 each year in the USSR, it is not surprising that welding engineers are seldom found in plants fabricating bridge girders. The FHWA for the last 2 years has advertized for a welding engineer, offering a qualified person

$30,000 per year plus excellent fringe benefits, and is still looking.

Only two schools in the USA offer a BS degree in Welding Engineering, Ohio State and LeTourneau (Longview, Texas). The author knows of no college or university in the USA offering a BS in Nondestructive Testing. If in the 1980's we are to minimize the incidence of weld cracking, and when occasionally it does occur, if we are to be assured of finding and sizing the cracks, it will be necessary to have an adequate supply of well-trained welding engineers and nondestructive testing engineers. This is true not only for more efficient, better weld fabrication but also if we are to regain the lead in research and development. Without the latter we will be coasting; Omer Blodgett of Lincoln Electric tells his design classes that in todays rapidly advancing technology, to coast is to go down hill.

Assuming that our college and university administrators see the light and deliver well-trained welding engineers and nondestructive testing engineers in sufficient quantity, let's look at some of the changes needed to reduce the incidence of weld cracking as enumerated above.

(1) Control of tack welds. AWS D1.1 as presently written is ambiguous and imcomplete in this connection. In bridges, tack welds are not allowed in the completed structure and, therefore, must be removed or overlayed with a continuous weld. When tack welds are not completely consumed by the final weld, cracks that are produced in the tack weld or in the heat-affected base metal under the tack weld are initiation sites for fatigue or brittle fracture in service. Tack welding is inherently a low-energy-input procedure and, therefore, is inherently prone to produce weld heat-affected-zone (HAZ) cracking. Moreover, the high hardness that attends low-energy-input welding procedures in hardenable steels, like A588 and A514, makes hydrogen cracking more likely. If preheat is used in tacking, the chances of both HAZ hardening and hydrogen cracking will be reduced. To accomplish this AWS D1.1 par 3.37 will have to be revised to read:

> 3.37.1 Tack welds should be subject to the same quality requirements as the final welds except that (1) minor (superficial) discontinuities such as undercut and unfilled craters need not be removed before final welding.
>
> 3.3.7.2 Tack welds, whether incorporated into a final weld or not, shall be subject to the same procedural and consumables requirements as the final welds -

(1) when cracking and/or porosity are visible in the surface of a tack weld, the tack weld shall be removed before final welding;

(2) whether incorporated into a final weld or not, tack welds shall be preheated to 150°F or in accord with Table 4.2, whichever is higher.

3.3.7.3 Tack welds which are not to be incorporated into final welds shall be removed and the tack-welded surfaces ground smooth and flush, and the surfaces inspected with MT and/or DT.

There are many changes to be made in the Structural Welding Code; the above is cited in detail as an example of what will have to be done in the 1980's to tighten the present specification.

(2) Use of low-hydrogen practice. The use of low-hydrogen electrodes is only the first step in avoiding delayed (hydrogen) cracking. Low-hydrogen practice involves control over moisture and contaminants (such as hydrocarbons) at every step of fabrication. Weld plate surfaces and welding wire must be free of rust and hydrocarbons. Rust may contain chemically combined water which can not be driven off by preheating. Grease and oil and drawing lubricants (on wire) are hydrocarbons which break down in the arc to form hydrogen. E6010 and E7010 electrodes should never be allowed where low-hydrogen practice is specified. After consumables (electrodes and fluxes, including FCAW wire) are exposed to the ambient shop environment for times in excess of the code-stated limits, low-temperature (300 to 400°F) baking in a low-vacuum chamber will permit multiple rebaking without damage to the consumables. Properly used holding ovens for flux and electrodes positioned near the welding station are essential to low-hydrogen practice. Preheat/interpass and postheat temperatures should no longer be arbitarily specified but calculated based on the critical cooling rate to avoid undesirable weld-transformation products. Post-heating of welds and weld repairs will be done with temperature and time calculated to remove all but very low levels of residual hydrogen. Carbon equivalent calculations will be made for each heat of steel to assure that an adequate preheat/interpass temperature is used to avoid excessive HAZ hardness.

Obviously low-hydrogen practice will only be implemented in the 1980's with a sufficient number of (1) certified and qualified welding inspectors and (2) welding engineers.

(3) Weld repairs and cosmetic corrections. The latter constitute a situation that is presently out of control. Shop personnel are making engineering judgements as to what corrections need to be made

and how the corrections are to be executed. If the incidence of cracking is to be reduced in the 1980's, cosmetic corrections as such will be prohibited and treated as weld repairs. Weld repairs will be given extraordinary attention by both welding and nondestructive testing engineers, with mandatory weld-qualification-procedure testing for repairs made to fracture-critical members. All critical weld repairs will be post heated with temperature and time calculated to remove all but a very low level of residual hydrogen.

(4) Control of weld-transformation products. With knowledge of the minimum cooling rate for avoiding low-toughness, high-temperature transformation products and the maximum cooling rate to avoid the martensite transformation (hard, brittle HAZ), the welding engineer will calculate the energy input and preheat temperature that will result in cooling rates within an allowable range.

When steel heats are found to have high hardenability; with carbon equivalent (CE) in excess of say 0.60 based on the IIW formula, weld transformation to martensite will be avoided by interrupting the cooling cycle just above the M_s temperature (start of the martensite transformation) using the chemical composition for calculating the M_s temperature. The duration of the postheating will be based on the time to isothermally transform to lower bainite.

(5) Tough crack-resistant weld metal. As long as human beings are making welds, there is the chance of error and/or omission and problems with workmanship. Thus, the weld deposit is the most likely place to find defects, defects ranging from relatively innocuous porosity to planar defects like lack of penetration and cracks. If such defects are to remain dormant or at worst grow very slowly, the weld metal must be tough so that the critical crack size is no smaller than the plate/weld thickness.

(6) Sizing of defects. Ultrasonic testing will have to be updated both with regard to instrumentation and certification of personnel. Inspectors will be certified for assignment to fracture-critical members on the basis of a performance test administered by the State Department of Transportation. Automated scanning devices will be utilized together with computer analysis of the recorded data. B- and C-scan testing will be done to determine (1) the cross-sectional position of a reflector and (2) plan-view contour maps of echo amplitude. Two-probe time-delay testing will be used for defect sizing; a diffracted longitudinal-wave pulse of ultrasonic energy from the defect extremity will be accurately timed enabling the position of the extremity to be defined in the vertical plane.

(7) Crack-tough steel for bridges. Fracture-critical members (FCM's) will be fabricated using steels purchased with guaranteed minimum Charpy V-notch impact values at the lowest anticipated service temperature (LAST) and a guaranteed ASTM E208 nil-ductility transition (NDT) temperature 30°F or more below the LAST. Crack-arrest properties at the LAST will be specified. When the designer anticipates problems with lamellar tearing, Z-direction properties will be specified. Several new steels will be used (new only to bridge construction) such as ASTM A633 grades A thru E, incl, A678 grades A, B and C, and A710 grade A classes 1, 2, and 3. The latter, with 0.07 carbon maximum, will be specified where FCM's are to be welded in the field; otherwise, no welding will be allowed during erection of FCM's. The Charpy V-notch test will be used only in the steel mill for checking plate compliance with the minimum specified properties. The precrack Charpy impact test will be used by QC/QA to verify that the NDT temperature is at least 30°F below the LAST.

Crack toughness will be specified for both weld heat-affected zone (HAZ) and fusion zone. The microstructure in the HAZ with lowest crack initiation/propagation resistance will be determined using composite precrack Charpy impact testing as a requirement of weld-procedure-qualification testing for FCM's.

Just about everyone will admit that it is better and less costly to do a job right the first time. But short cuts (cutting corners) can be very very tempting from the standpoint of cost and delivery schedules. But when a fabricator runs into weld cracking problems, it frequently means extraordinary cost escalation and loss of production time. When cracks escape QC/QA and are found after a bridge is in service, the cost is usually passed to the tax payer. Millions of dollars are lost each time this happens, in inspection, retrofit and manhours lost in redirected traffic. And in the event of a catastrophic collapse with possible loss of life, the cost is incalculable.

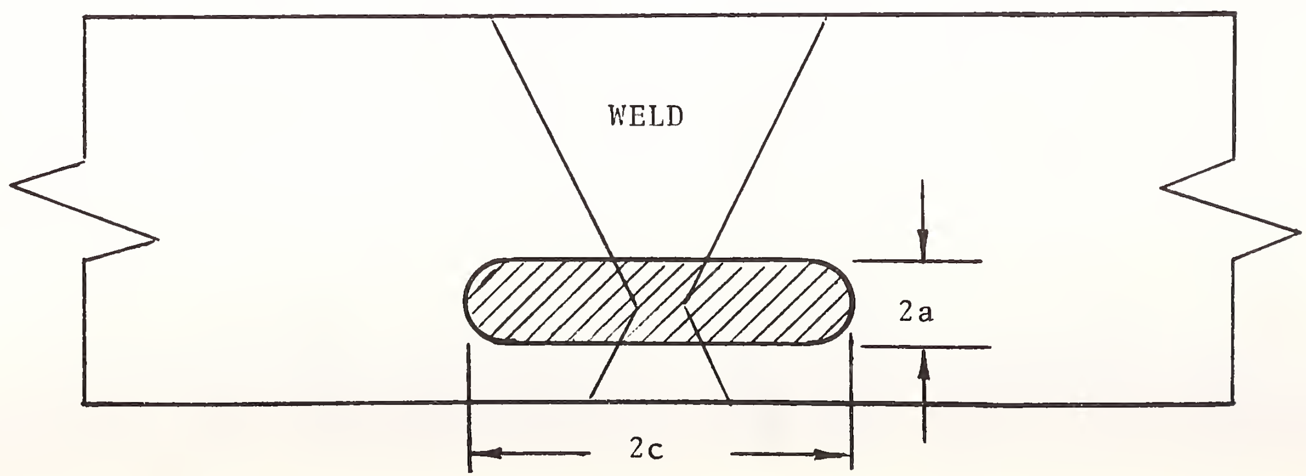
TABLE 1

CRITICAL CRACK SIZE IN A588

ASSUME: YIELD-POINT RESIDUAL STRESSES
65-KSI WELD METAL

EMBEDDED FLAW $a = KIc^2Q/1.1^2\pi S^2$

CVN (FT-LB)	KIc (KSI-IN½)	WELD FLAW DIMENSIONS a/2C = 0.1		a/2C = 0.3	
		DEPTH	LENGTH	DEPTH	LENGTH
15	47	0.25	1.2	0.38	0.6
25	61	0.42	2.1	0.65	1.1
35	72	0.58	2.9	0.90	1.5



THEORY AND DESIGN OF INSTRUMENTATION
FOR BRIDGE INVESTIGATION

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The U.S. Department of Transportation is currently responsible for approximately 600,000 bridges. Approximately 100,000 bridges per year require maintenance.

According to Escalante and Ito[1/], the rate of deterioration of concrete structures has increased considerably over the past 10-15 years: The 1969 annual total cost estimated for restoration and protection of bridges on the interstate road system was $2.6 billion.

Commercial instrumentation is available for evaluating the deterioration of small metal probes that can be imbedded in concrete in close proximity to the metal of interest. This may be adequate for future concrete bridge footings, but does not address the problem of testing older footings.

An "Acoustic Crack Detector" mounted on a back pack is currently used with marginal success. Radiation gaging has been tried and found to give inaccurate diagnostic information.

The fabrication of a panacean detection instrumentation is highly unlikely for the diagnosis of bridge footings. However, theoretical calculations have indicated that hybrid instrumentation using both radiation and electromagnetic interrogation for measurement of the integrity of bridge footings is feasible.

This paper will address the theory of design of such instrumentation.

1/ NBS Special Publication 550: A Bibliography on the Corrosion and Protection of Steel in Concrete

Development of standard tests and testing methods for the evaluation of construction materials has been of interest since the turn of the century. Congress created the National Bureau of Standards in 1901. The American Society for Testing Materials (ASTM) was formally organized and legally chartered by the State of Pennsylvania in 1902. The two of the first four original technical ASTM Committees formed were Committee A-1 on steel, and Committee C-1 on Cement.

The objective of this set of experiments is to theoretically derive the mathematical formulations necessary for the optimization of radiation wavelengths employed for gauging of composite materials: concrete reinforced by steel. Firstly, consider the Beer-Lambert Law applied to a single component matrix.

$$I = I_o \, \epsilon^{-\mu x}$$

Where

I = Transmitted Radiation Intensity

I_o = Incident Radiation Intensity

μ = Linear Attenuation Coefficient (cm^{-1})

x = Matrix Thickness (cm)

$$\mu = \frac{\rho N \sigma}{A}$$

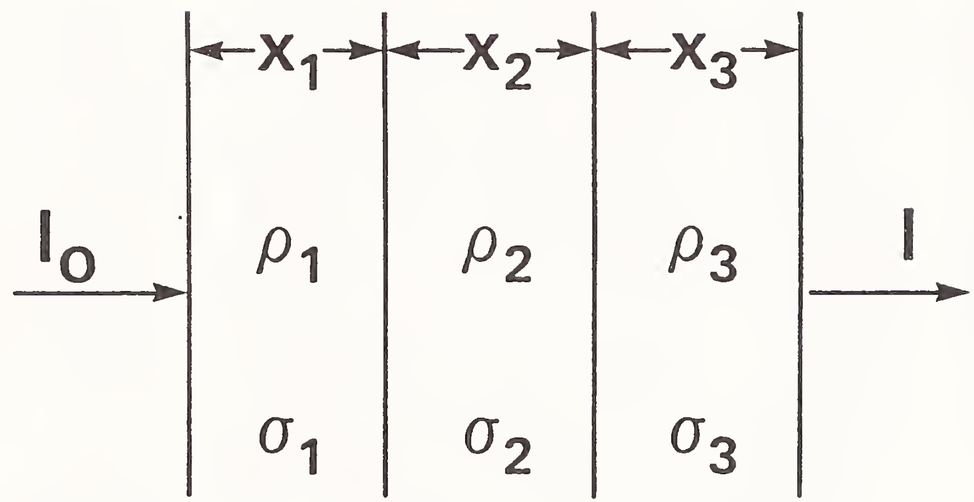
ρ = Matrix Density

N = Number of Atoms $Mole^{-1}$

σ = Total Cross Section in Barns

A = Effective Atomic Weight in Grams

NOW CONSIDER A MULTICOMPONENT MATRIX



THEN: $I = I_o \, \epsilon^{-(\mu_1 x_1 + \mu_2 x_2 + \mu_3 x_3)}$

Example Instrumentation to Prove the Theory

A gamma ray transmission instrument was developed to detect blockage in molten material transfer lines. 1/

Periodically, problems are encountered with blockages in the lines used to transfer molten material from one location to another in industrial plants. The lack of productivity during the downtime required to locate the blockages can represent a significant economic impact to the companies affected. These periods can last as long as a week and can represent a dollar production loss of $50,000/hr.

Blockages occur as a result of impurity buildup or freezing inside the transfer lines. Various compounds present as impurities comprise the blockages since these compounds often possess higher melting temperatures than that of the molten material. The impurities rapidly solidify restricting the material flow. Finally, complete blockage occurs. (See Figure I).

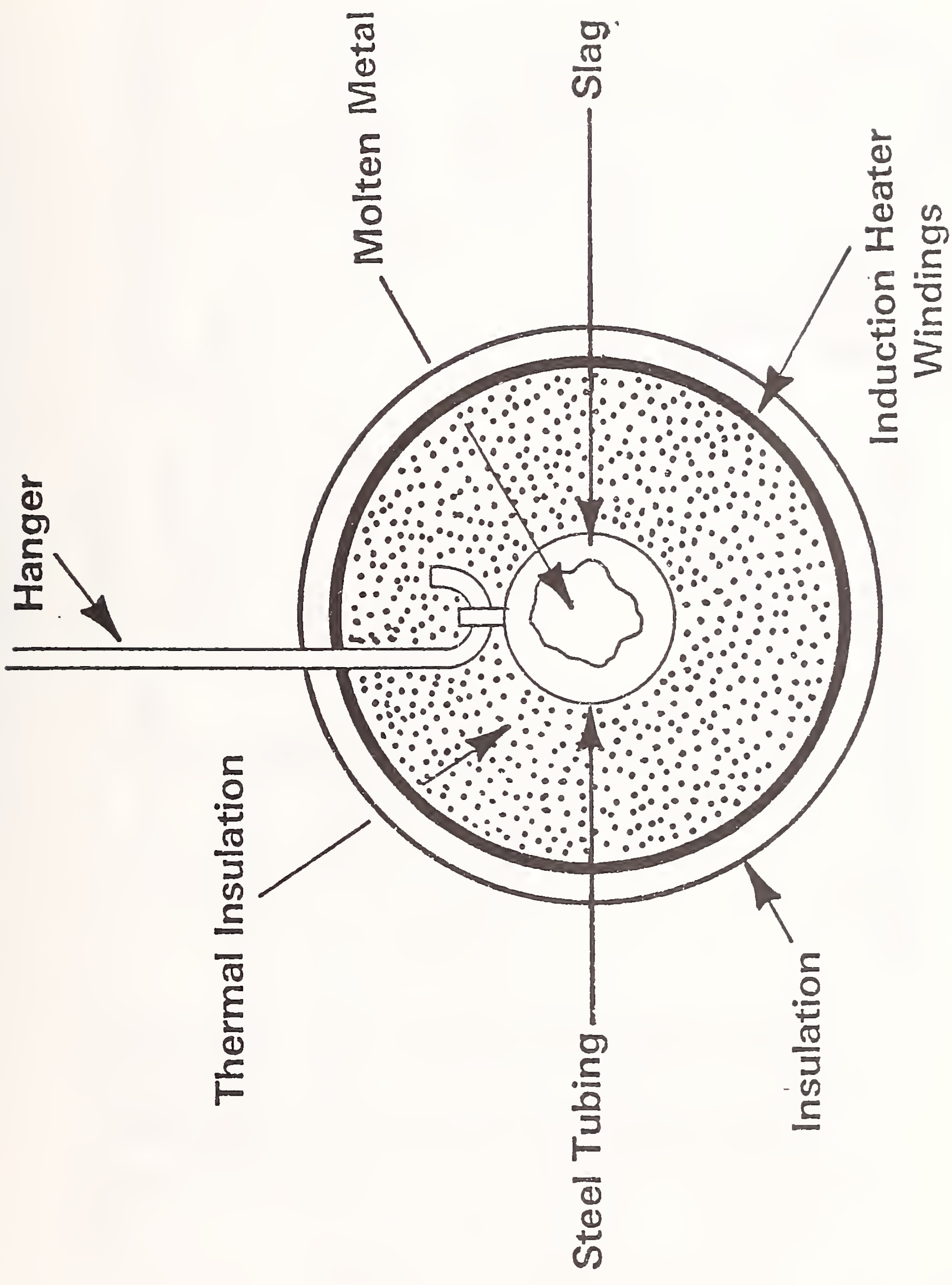
A blocked line can be cut on a "best guess" basis and a blockage is located by inserting a tape into the line. Oftentimes the lines must be cut and probed in several places before the location of the blockage can be determined. It is possible to displace impurities with the tape and temporarily remove the obstruction. The blockage can reoccur whenever material flow is re-established in the lines.

An instrument for accurately and rapidly locating blockage based on the gamma ray transmission technique is shown in Figure II. A highly collimated beam of gamma rays from a small radioactive source is transmitted through the blocked transfer lines. General trends and localized anomalies in the blockage deposits are detected by observing variations in the transmitted gamma ray beam intensity. The beam intensity represents a quantitative measure of the thickness of blockage deposits present in the pipe. (See Figure III).

Now consider concrete that has been attacked by sea borers. Sea borers produce channels in the concrete which result in damage to the structural integrity of pilings below the water line. Also consider the semi-random placement of the steel reinforcing rods shown in Figures IV, V.

We feel that a cross fertilization of technology is needed to solve these problems: Radiation transmission combined with ferro-magnetic measurements aided by a mini-computer should yield results defining the integrity of the structure under test.

1/ D. A. Garrett, GAMMA RAY TRANSMISSION GAGING FOR THE DETECTION OF BLOCKAGES IN MOLTEN MATERIAL TRANSFER LINES Presented at the Spring Conference/Seminar of the American Society for Nondestructive Testing April 2 - 5, 1978, New Orleans, Louisiana



Hanger
Molten Metal
Slag
Induction Heater
Windings
Thermal Insulation
Steel Tubing
Insulation

FIGURE 1

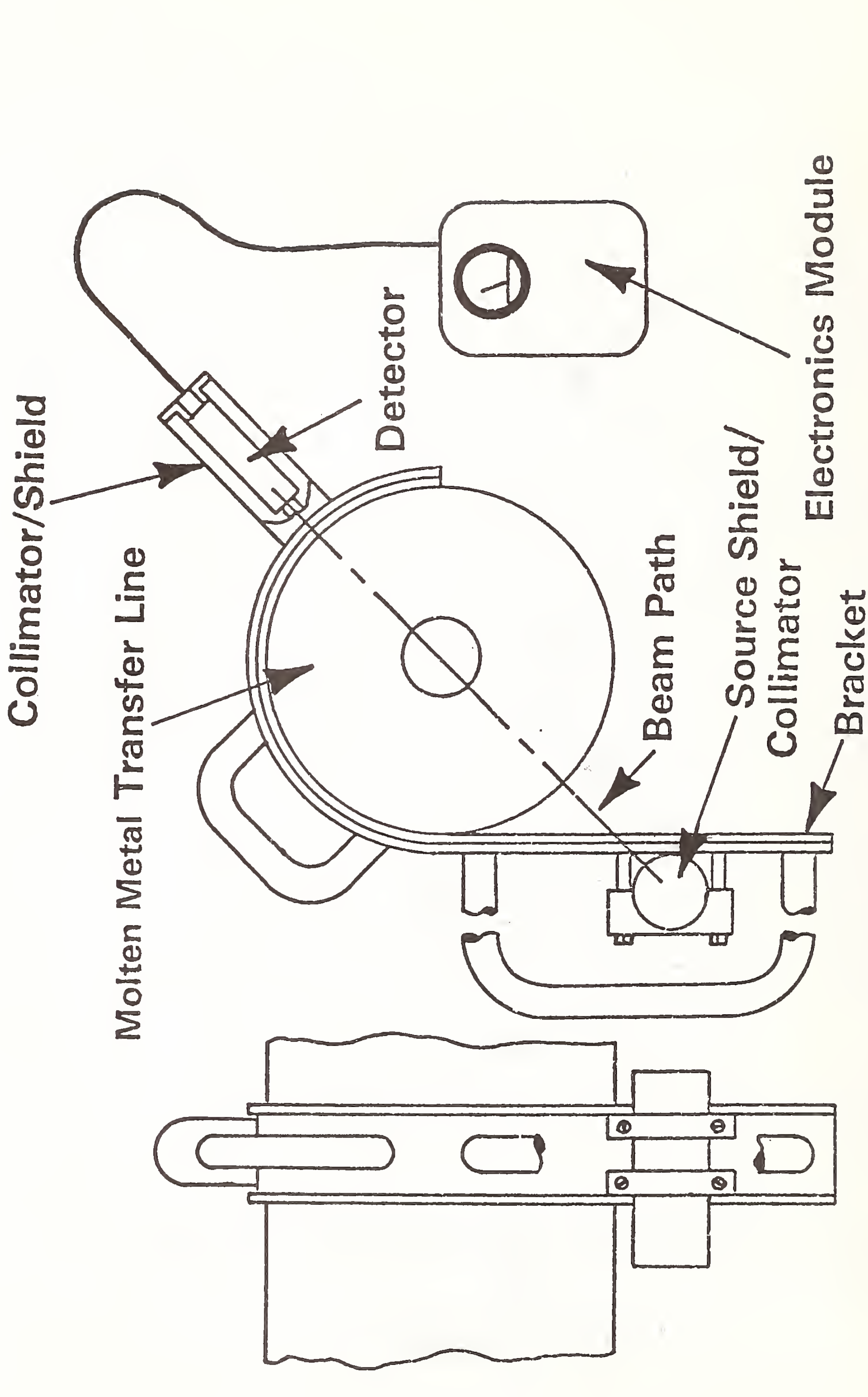

Collimator/Shield
Molten Metal Transfer Line
Detector
Electronics Module
Beam Path
Source Shield/
Collimator
Bracket

FIGURE II

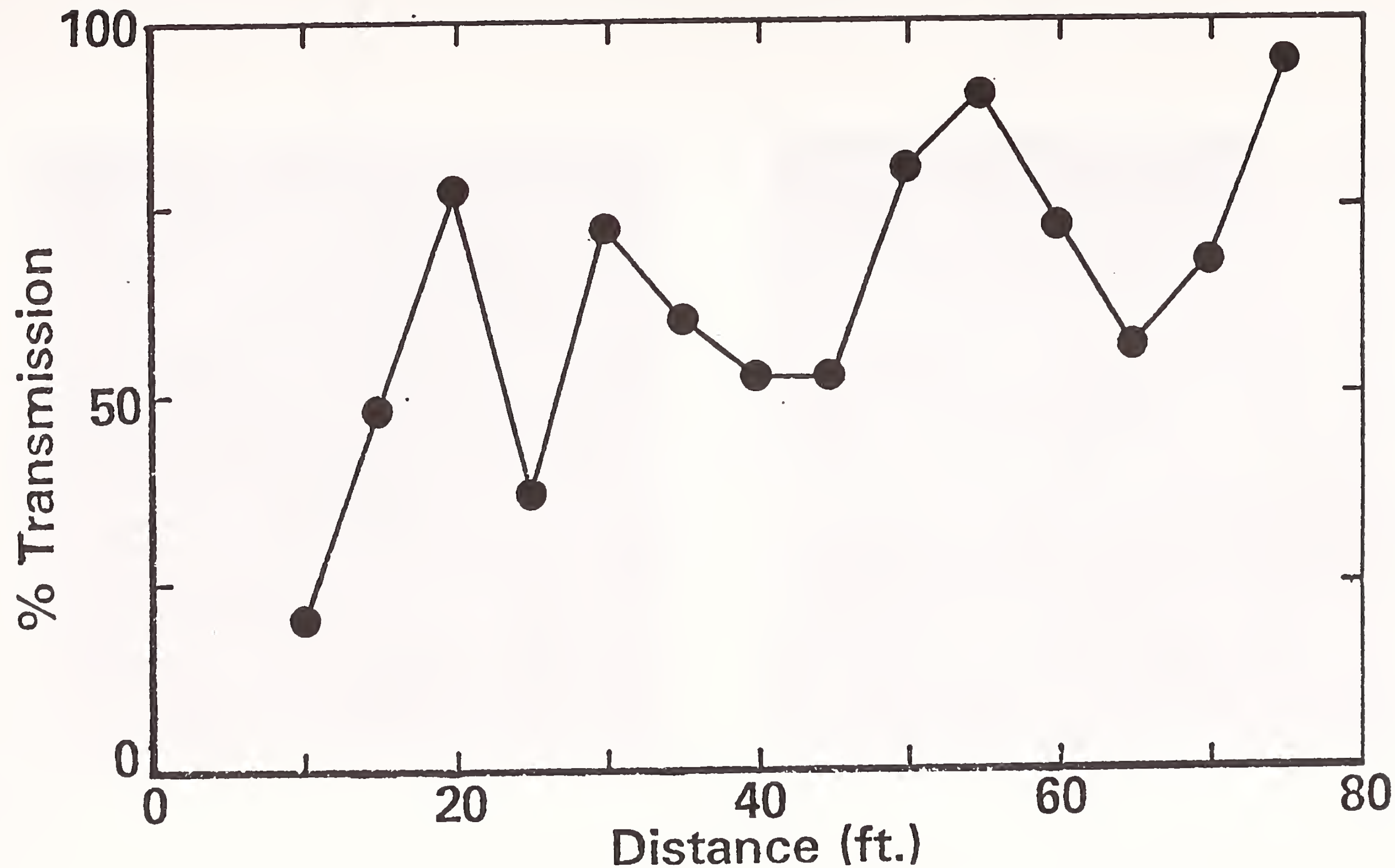


FIGURE III

Figure IV

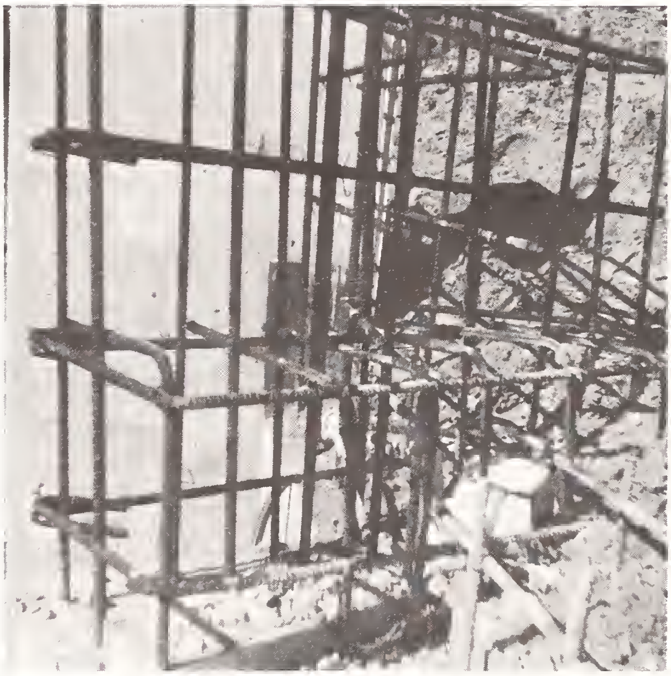


Figure V

SESSION IV

PIPELINE TRANSPORTATION SYSTEMS

Chairman: M. Lauriente, Department of Transportation

DUCTILE FRACTURE ANALYSIS OF PIPELINES

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Abstract: In this paper after briefly discussing the evolution fracture failure in pipelines and reviewing the related basic concepts, the problem of ductile fracture is considered. It is assumed that the thickness-to-radius ratio of the pipe is sufficiently small so that structurally the pipe may be treated as a cylindrical shell. It is also assumed that the pipe wall contains a part-through crack around which yielding takes place through the entire wall thickness. In the analysis the plastic deformations are approximated by a perfectly plastic layer and the bending theory of shallow shells is used to solve the problem. The questions regarding the fracture initiation along the crack front, the progressive growth of the crack, the plastic necking of the net ligament, and the pipe rupture leading to leak or break are then considered. The concept of crack opening stretch is used in the analysis and the discussion of ductile fracture.

Key words: Pipeline fracture; ductile fracture; part-through crack; crack opening displacement; crack initiation; progressive crack growth; plastic necking instability; leak vs. break.

Introduction

Depending on the thermo-mechanical behavior of the material and the nature of the applied loads and the environmental conditions, in the design of pipelines, tank cars, and a great variety of other shell structures it is often necessary to consider fatigue or corrosion crack propagation and fracture among the possible modes of failure. This requires, in addition to the application of standard failure theories specified by the existing design codes, the treatment of the problem of acceptance and safety from the viewpoint of fracture mechanics. In analyzing the fracture failure in such structures one is likely to encounter nearly all types of fracture ranging from the subcritical growth of initial cracks involving practically no plastic deformations to the final rupture of the pipe wall resulting from the net ligament plastic necking instability. Generally, in pipes, containers, and pressure vessels the fracture failure may evolve in the following manner: (a) A fatigue or a corrosion fatigue crack may initiate around a localized imperfection, generally a surface flaw, having the worst possible combination of geometry and loading conditions. (b) Next, this dominant flaw may grow subcritically taking roughly a semielliptic shape. At the initial stages of this phase the cylinder wall would be mostly elastic and the plastic deformations would be confined to a relatively small region along the crack front only (Figure 1a and b). (c) As a consequence of increased crack depth the net ligament through the wall would become fully plastic and the rate of subcritical crack propagation would increase markedly (Figure 1 c). (d) Finally, under the peak load the net ligament would become unstable and rupture. Depending on the combination of factors which consists mainly of relative magnitudes of the fracture resistance of the material and the crack driving force, the resulting through crack would either be arrested or would continue to grow in an unstable fashion. These phenomena are respectively known as the "leak" and the "break".

In the initial phase of the fracture process outlined above a predictive analysis of the problem of fatigue crack propagation can be carried out provided the stress intensity factor along the crack front is known. This requires the solution of the three dimensional elasticity problem for a vessel with a part-through surface crack. At the present time the problem appears to be analytically intractable. The existing solutions of the problem are based on either the alternating method (see, for example, the review articles by F.W. Smith, and R.C. Shah and A.S. Kobayashi in [1]), or the finite elements [2-4], or the method of boundary integral equations [5].

At the third and fourth stages of the fracture process (Figures 1 c, d, and e) it is understood that prior to net ligament rupture the pipe wall around the crack would undergo large scale plastic deformations. Consequently, the elasticity solutions would not be adequate to study these phases of the fracture phenomenon. In a large group of important structures in which the thickness to radius ratio is relatively small, the last phases of fracture usually involve relatively long and deep surface cracks with full plastic deformations along the entire net ligament and through the container wall around the crack tips. Particularly in structures such as the great variety of straight and bent piping and pressurized containers used in transportation in which fluctuating bending stresses are superimposed on stresses caused by the internal pressure, relatively long surface cracks before rupture appear to be a rule rather than an exception. By making two major assumptions the fracture problems arising in the failure analysis of such structures may be brought under control and treated analytically. The first assumption is that because of the relatively long crack length and small thickness-to-radii ratios the structure can be adequately modeled as a "shell". The second assumption involves the use of a modified version of the conventional plastic strip model to account for the plastic deformations in the crack region. In a structural component undergoing ductile fracture which involves progressive crack growth, it is generally agreed that the size of the energy dissipation zone R_D shown in Figure 2 is an increasing function of the crack length. Consequently, in the case of slow progressive ductile fracture the fracture resistance of the solid increases with the increasing crack length (hence, the concept of "crack extension resistance curve" or the R-curve). In progressive ductile fracture one may generally distinguish four different regions in the material, namely the basic energy dissipation zone R_D, the wake region of the residual stresses R_W, the crack

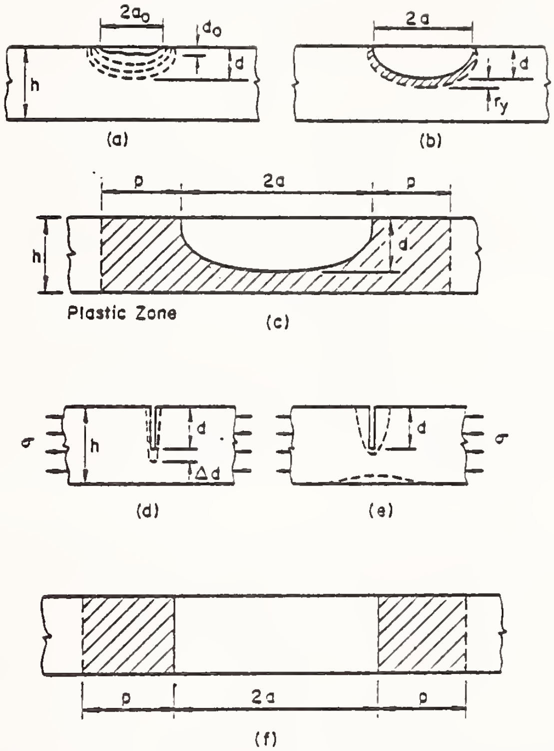

Figure 1. Evolution of a through crack

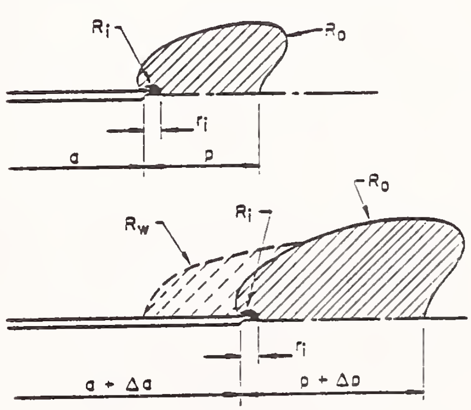

Figure 2. Progressively growing ductile fracture

initiation zone R_i, and the remaining elastic zone R_E (Figure 2). There is some experimental evidence to the effect that at the tip of a growing crack the "crack opening angle" or the "crack opening displacement" remains constant. Even though this notion has not been fully documented and universally accepted, it may still be used to support the conjecture that in order for the fracture initiation to take place at the crack front, some local strength parameter must reach a critical value- and, in the absence of a more suitable load factor, the crack opening stretch may be considered as the measure of this parameter. This is a necessary condition for fracture propagation and is related to the formation, growth and coalescence of holes in the small fracture initiation or "fracture process zone" R_i at the crack front. On the other hand continuous (stable) growth and instability of fracture require that the condition of global energy balance be satisfied. The latter constitutes essentially the sufficient condition for fracture propagation. The R-curve approach to ductile fracture is a typical application of this concept.

The type of ductile fracture resulting from progressively growing cracks would occur usually in structural components having a through crack or having a part-through crack with a relatively small crack depth to wall thickness ratio (Figure 1 d and f). However, for some crack-structure geometry and material combinations the crack opening stretch may reach and far exceed the local critical value and yet the condition of global energy balance may not be satisfied. Some plate and shell structures of high toughness materials having relatively long and deep surface cracks may fall into this category. In this case the net ligament would generally undergo plastic "necking", and the net ligament rupture may result from a "necking instability" under increased membrane loads. In the following sections the problem of ductile fracture in pipes containing surface cracks will be discussed by using these concepts with the crack opening stretch as the primary correlation parameter. It should perhaps be pointed out that the concept of "tearing modulus" recently introduced by Paris appears to be a very effective way of characterizing and classifying structural materials with respect to their ductile fracture resistance. However, the concept is confined to certain types of J-controlled progressive fracture and is not applicable to net ligament plastic necking instability.

Ductile Fracture of Cylinders with a Part-Through Crack

Let a relatively thin-walled pressurized cylinder contain a surface crack which is sufficiently deep and long so that under the given pressure the net ligament and the cylinder wall around the crack are plastically deformed. Using a plastic strip model which is basically a generalization of the conventional Dugdale model used in plane problems, and the bending theory of shells one can calculate the crack opening stretch along the leading edge of the crack [6]. Figures 1c and 1f show the geometries of the part-through and the through crack, respectively. A sample result for the crack opening stretch δ calculated at the tips of an axial through crack on the mid-surface of the cylinder is shown in Figure 3 where $N_O = p_O R$ is the membrane resultant in hoop direction, σ_Y is the flow stress representing the yield behavior of the material, and the shell parameter λ and the normalization factor d_1 are given by

$$\lambda = [12(1-\nu^2)]^{\frac{1}{4}}\ a/\sqrt{Rh}\ ,\ d_1 = 4a\sigma_Y/E \qquad (1)$$

R, h, and 2a being the mean shell radius, the wall thickness, and the crack length, respectively. The curve $\lambda=0$ corresponds to the flat plate results.

One can use such results in three different ways. First, if one assumes that the critical crack opening stretch can be used as the local crack initiation criterion, then based on Figure 3 and on similar results obtained from a part-through crack solution a set of design curves may be prepared which would give the load carrying capacity of the cylinder against crack initiation. These curves are shown in Figure 4 where the varying parameter δ/d_1 correspond to the prescribed critical crack opening stretch ratio (which is a property of the material). Note that the results are general and may be used for any material for which the critical stretch falls between 0.4 d_1 and 12 d_1. Some sample results for a part-through crack are shown in Figure 5. Here δ_c is the maximum crack opening stretch which is at the deepest penetration point (or the midpoint) of the crack. The dots on the right hand side of the figure correspond to the load carrying capacity of the cylinder having a through crack of the same length 2a. The fact that they are generally somewhat greater than the extrapolated values obtained from the part-through crack solution may indicate that in some cases at least in principle leak before break is possible.

One may also use the crack opening stretch concept to model the fracture process in a cylinder with a fully yielded net ligament and to determine the failure pressure. Even though it is very difficult to predict whether a given fully yielded net ligament will undergo progressive fracture propagation or plastic necking, experiments indicate that for a progressive crack growth characterization of a fracture process the crack opening stretch must be lower than a certain value. For example, the results given in [7] suggest that for progressive fracture propagation the crack opening stretch at the leading edge of the crack needs to be less than approximately 0.04 (h-d), h-d being the thickness of the net ligament. On the other hand the results of the tests on pressurized steel cylinders reported in [8] show that for conditions near the rupture value of pressure the opening stretch δ_c at the deepest penetration point of the crack is generally greater than 0.1(h-d). Therefore, in cylinders with a part-through crack it is possible to encounter both a progressive crack growth in the fully-yielded net ligament and the plastic necking of the net ligament. In the safety analysis of this type structures the most important problem is quite clearly the estimation of the instability value of the pressure. There seems to be no simple and unique methods currently available to calculate

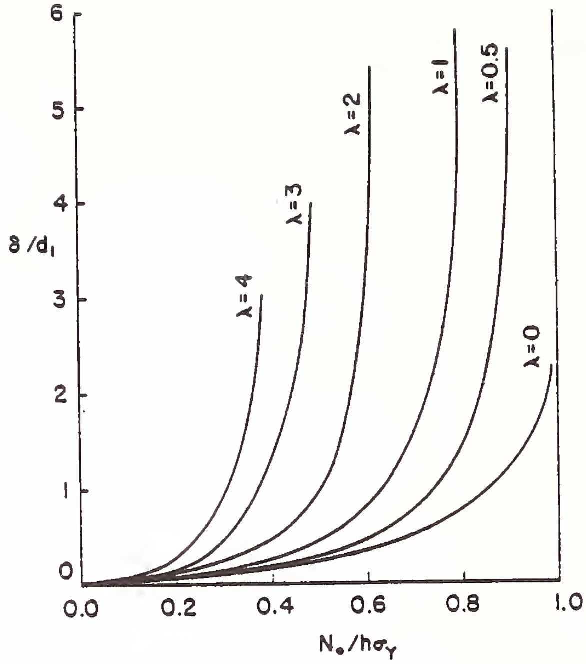

Figure 3. Crack opening stretch at the tip of an axial through crack in a cylindrical shell.

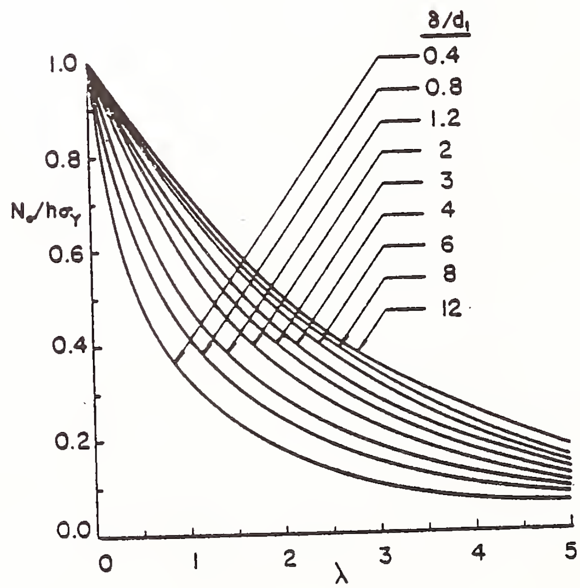

Figure 4. Load carrying capacity of a pressurized cylindrical shell with an axial through crack against crack initiation.

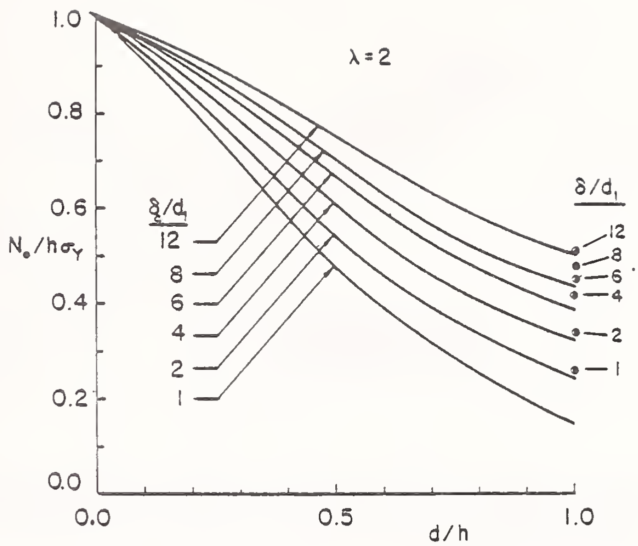

Figure 5. Load carrying capacity of a pressurized cylinder with an axial part-through crack.

this pressure. For relatively thin-walled plate and shell structures having a part-through crack with a fully yielded net ligament the two methods outlined below appear to be very promising.

The first method refers to the calculation of the instability pressure for the progressively growing part-through crack. Examining the crack opening stretch curves given by Figure 3 and similar results obtained for the part-through crack (see, for example, [9]), it may be observed that for a given crack geometry when the membrane stress resultant $N_O = p_O R$ reaches a certain value, a slight increase in the pressure p_O may cause a very large increase in the crack opening stretch δ. Physically this implies some kind of a plastic instability. Thus, one may assume that the instability load corresponds to having an arbitrarily high relative slope in the δ vs. N_O curves. Fixing this slope, for example, at an arbitrarily high value of 20 in the units shown in Figure 3, for each crack depth d one may obtain a curve giving the instability load as a function of the crack length 2a (or the shell parameter λ). For an axial crack in a pressurized cylinder these results are shown in Figure 6. One may note that, aside from the dimensions of the crack and the cylinder, the only information needed to obtain these curves is the value of the "flow stress" σ_Y characterizing the yield behavior of the material. Figures 7 and 8 show the comparison of the theoretical results given in Figure 6 with the results of burst tests obtained at Battelle Laboratories [8] for steel pipes (R = 18 in; h $\cong$ 0.4 in.). Considering the complexity of the problem the agreement appears to be quite good.

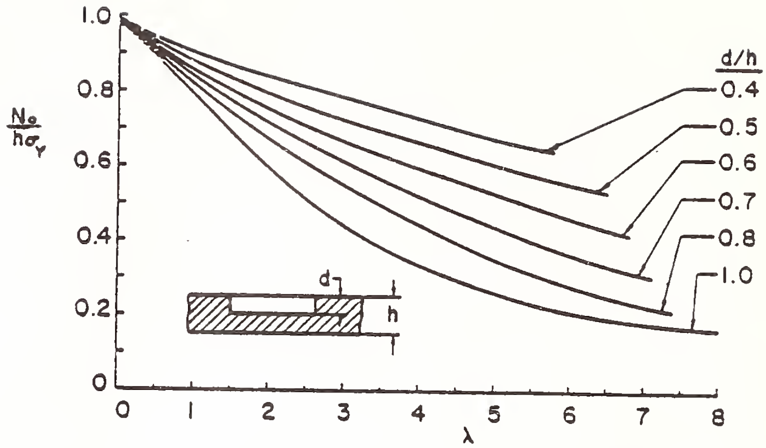

Figure 6. The instability load $N_0 = Rp_0$ for a pressurized cylinder with a progressively growing part-through crack (d<h) or a through crack (d=h).

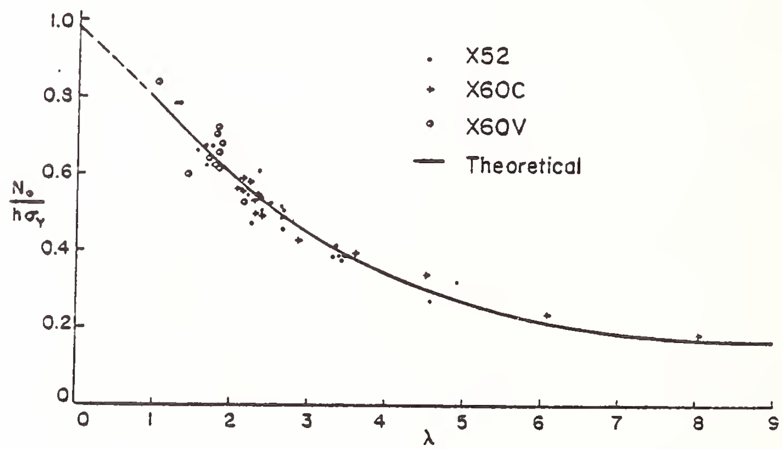

Figure 7. Comparison of calculated (Figure 6) and experimental [8] instability loads in steel pipes with an axial through crack.

The second method concerns the calculation of the instability load in cylinders having a part-through crack for which the net ligament undergoes plastic necking. In this case as the pressure is increased, unlike the progressively growing part-through crack problem in which there is no significant change in the initial crack depth d_0 until the crack initiation and propagation begin, the crack depth d_0 also increases. For a given pressure below the instability value, the crack

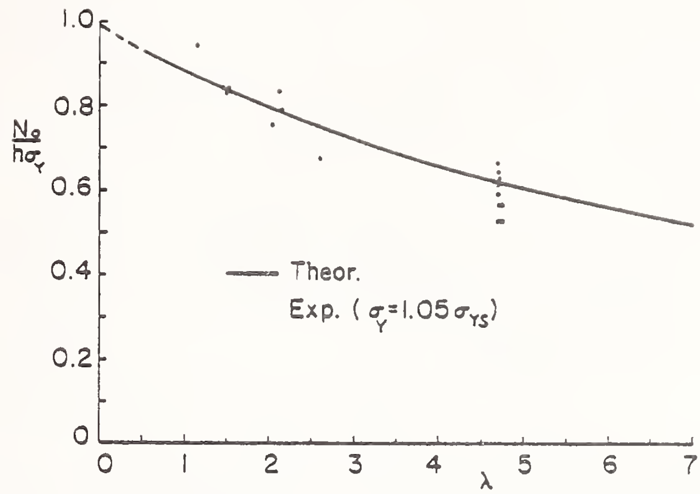

Figure 8. Comparison of calculated (Figure 6) and experimental [8] instability loads in pipes with an axial part-through crack.

depth reaches an equilibrium value, there is a stress redistribution around the crack region, and the stress in the net ligament becomes the plastic stress corresponding to the prevailing strain. The fracture instability of the net ligament is clearly related to the question of whether or not there exists an equilibrium value of the crack depth d. Under the given pressure if d has an equilibrium value, the net ligament may undergo some plastic necking but no rupture would take place. On the other hand, if d has no equilibrium value then fracture will occur and the (smallest) corresponding pressure is the fracture instability pressure.

In order to investigate whether or not an equilibrium value of the crack depth d exists for a given pressure one may express d as follows (see [6]):

$$d = d_o + \alpha(\varepsilon_z)\delta_z \tag{2}$$

where d_o is the initial crack depth, α is a function of the average net ligament strain ε_z, and δ_z is the average stretch in the net ligament. ε_z and δ_z are functions of the current value of the crack depth d. δ_z is calculated from the elastic-plastic shell theory and hence is a known function of d. The nonlinear equation (2) can be solved for d by a successive approximation technique by assuming $d(0) = d_o$ and

$$d(N+1) = d_o + \alpha(\varepsilon_z(N))\ \delta_z(N) \quad , \quad N = 0,1,2,... \tag{3}$$

Now, if d has an equilibrium value it can be obtained from (3) as the limit for $N\to\infty$. Clearly for the given pressure if the successive

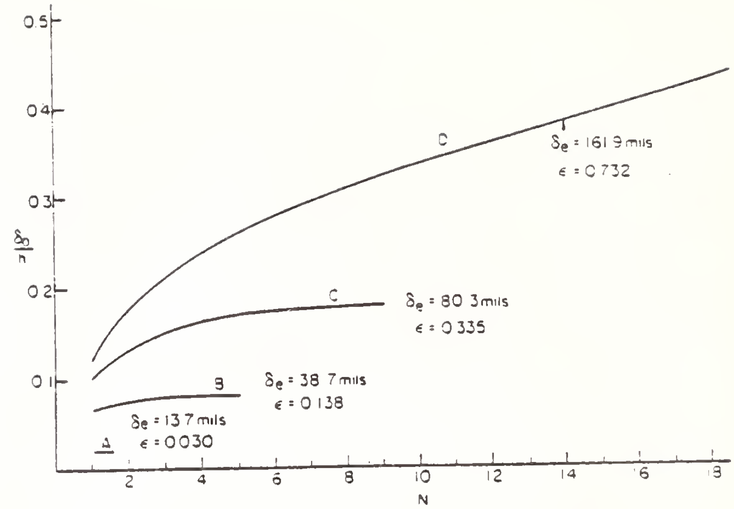

Figure 9. Results of the successive approximation calculations to determine the burst pressure (N is the number of iterations).

approximation formulated by (3) is convergent giving a finite value which is less than the thickness h, then there would be a load redistribution around the crack accommodating a stable net ligament thickness. On the other hand the divergence of the successive approximation technique (as defined by d exceeding h) would mean the fracture due to net ligament plastic necking instability.

As an illustration of the technique to estimate the failure pressure in cylinders we consider the tests carried out on full scale line pipes at Battelle [8]. Following is the relevant information regarding the tests.

Dimensions: $R = 18$ in, $h = 0.403$ in, $d_0 = 0.201$ in,
$2a = 3.8$ in, $\sigma_{YS} = 64.6$ ksi;

Test Pressure p_0(ksi):	1.0	1.2	1.25	1.29
δ_e(mills):	29	60	80	Failure

Here δ_e is the measured crack opening displacement on the outside surface of the cylinder and at the center section of the crack corresponding to the pressure levels shown. In applying the technique it was assumed that

$$\sigma_Y = \sigma_{YS} + 2 \text{ ksi}, \ \alpha(\varepsilon_z) = 0.46 + 0.54\left(1 - \frac{0.1}{\varepsilon_z}\right). \tag{4}$$

The calculation was carried out for the pressures 1.0, 1.2, 1.25, and 1.267 ksi. The result is shown in Figure 9. The ordinate in the figure is the crack opening displacement δ_0 on the middle surface of the cylinder and at the center section of the crack. Which one of the quantities δ_0, δ_e and d is used as an ordinate in the figure is, of course, immaterial. It is seen that the calculations for the first three pressures are convergent meaning that these pressures are below the instability or failure value. For p_0 = 1.267 ksi the calculations diverge, implying that this pressure is equal to or greater than the failure value. The figure also shows, for each pressure, calculated values of the average net ligament strain ε_z and δ_e. Note that the iteration technique predicts the failure pressure quite accurately. However, particularly at smaller pressures, the predictions for δ_e are not good. This is somewhat understandable, as the model is based on large scale yielding and large plastic strains which may not in fact exist at lower pressures.

Acknowledgement: The research on ductile fracture of pipelines at Lehigh University is being supported by the Department of Transportation under the Contract DOT-OS-82007.

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RADIOGRAPHIC VARIABLES AND WELD FLAW ANALYSIS

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Abstract: An alternative to existing workmanship codes, where failure predictions are based on a priori experience, is a fitness-for-purpose criteria where failure predictions are based on fracture mechanics analysis. In this alternate "code", critical flaw dimensions form the coordinates for a decision curve separating failure-no failure regions. The critical flaw dimensions under this system are the length and through-the-wall depth of the flaw, at least for the particular case of pipeline girth welds. This requirement to accurately dimension flaws puts additional quality control on the radiographic inspection procedures. The flaw depth, in particular, cannot be directly determined by scaling the radiograph. We need to measure the image contrast presented by the flaw and relate this to metal thickness reductions. Contrast however, is considerably influenced by the radiographic variables. Some of these are under the radiographer's control; some are not. The variables over which some degree of control can be exercised include x-ray kilovoltage and filtration, film and screen selection, and film processing techniques. The variables over which the radiographer has little control are mainly of a geometric nature - that is, they are related to the exposure geometry. These include x-ray source size, source to film distance and material thickness for example. These geometric quantities also determine the radiographic definition or "unsharpness" of the image. The limits on accuracy of flaw depth analysis due to these variables will be analyzed in this paper.

Key Words: Weld flaw inspection; radiographic nondestructive testing; flaw analysis from radiographs; pipeline radiographic inspection; flaw depth determinations.

Introduction

The existing workmanship code for liquid and gas pipelines is A.P.I.1104 [1]. This has been the standard for most pipeline construction in the United States for many years. The code undergoes continuous review by responsible representatives from API, AGA, AWS, etc. A detailed radiographic procedure is given in the code. With the exception of undercut however, flaw depth measurements are not specified in the code.

A newer technology - fracture mechanics - may form the basis of an alternate construction code. If this approach is followed, flaw dimensioning, particularly the length and depth of the flaw are important parameters of the analysis. To the extent that such a "code" is used the need for more accurate flaw sizing becomes significant.

Radiography, of course, is not the only nondestructive testing technique that is applicable to weld flaw detection and dimensioning. Each of the available techniques have advantages and disadvantages in this application. By and large, however, only radiography and ultrasonics predominate in field use. Perhaps this is because they are the most suitable for internal flaw detection and analysis.

Table I is a listing of the advantages and disadvantages of several nondestructive inspection methods that have been used in the field for pipeline weld flaw detection and evaluation.

Table 1

FIELD METHODS FOR FLAW CHARACTERIZATION

TECHNIQUE	ADVANTAGES	DISADVANTAGES
RADIOGRAPHY (FILM DENSITY CHANGE)	(1) ESTABLISHED FIELD PROCEDURE (2) ACTUAL IMAGE OF FLAW (3) CONVENIENT PERMANENT RECORD (4) LENDS ITSELF TO IMAGE PROCESSING (5) FLAW "DEPTH" OBTAINABLE (6) GREATEST DYNAMIC RANGE (>10^6)	(1) REQUIRES EXPERIENCED FILM INTERPRETATION (2) RADIATION HAZARD (3) EQUIPMENT FAIRLY EXPENSIVE (4) PROVIDES ONLY A THROUGH THE WALL IMAGE (5) CRACK DETECTABILITY STRONGLY DEPENDENT ON GEOMETRY (ORIENTATION) TO BEAM DIRECTION (NOT A STRONG POINT)
QUANTITATIVE ULTRASONICS	(1) ESTABLISHED QUALITATIVE METHOD (2) HIGHLY DEVELOPED TECHNICAL BASE (3) MEASURES OVERALL FLAW DEPTH (4) GOOD DYNAMIC RANGE (100 DB)	(1) REQUIRES HIGHLY SKILLED INTERPRETERS (2) TIME CONSUMING (3) DOES NOT DO WELL ON SMALL 3-DIMENSIONAL FLAWS (4) STRONGLY AFFECTED BY PIPE BASE METAL QUALITY (E.G. LAMINATIONS)
MAGNETIC PARTICLE (PENETRANT)	(1) MORE SENSITIVE THAN RT OR UT TO SURFACE FLAWS (2) DOES NOT REQUIRE HIGHLY SKILLED INTERPRETERS	(1) FLAW MUST BE AT OR NEAR SURFACE (2) DOES NOT MEASURE FLAW DEPTH (3) NOT PRACTICAL FOR LARGE AREAS (4) LIMITED TO FERROMAGNETIC MATERIALS
ELECTRIC CURRENT & EDDY CURRENT	(1) DOES NOT REQUIRE HIGHLY SKILLED INTERPRETERS (2) CAN MEASURE FLAW DEPTH	(1) REQUIRES GOOD PROBE CONTACT TO SURFACE (2) NOT PRACTICAL FOR SCANNING LARGE SURFACE AREA (3) BETTER FOR FLAW MEASUREMENT THAN FLAW DETECTION

A review of this table shows why radiography remains as the method of choice for nondestructive inspection of girth welds. In the remainder of this paper we will describe some of the characteristics of the radiographic process and their application to weld flaw analyses.

Girth Weld Radiography Details

Large pipelines use a panoramic x-ray source placed internally on the pipe axes and usually aligned in the plane passing through the center of the girth weld. Radiation is emitted in a 360° azimuthal direction. The beam cross section is fan shaped. Films and screens are placed in a light tight cloth package externally around the girth weld. The exposures, processing, and interpretations are carried out in the field.

In some cases the reading is made while the film is still wet. Frequently, these interpretations are made by the radiographer and perhaps a welding engineer or contractor, neither of whom are necessarily "readers". Recent studies reveal that a certified reader when combined with good radiographic quality control increases the probability of detecting a flaw - particularly cracks - the most difficult flaws for radiographic detection. [2] The reader certification should be carefully controlled and guidelines for this are given in reference 2. The problem of reader certification should be taken into account along with the problems of image quality control to be discussed here.

Image Quality

Table 2 lists the most common x-ray variables associated with the radiographic procedure. These variables will be discussed as they relate to flaw detection and sizing which of course are intimately related to the degree of image quality achieved.

Table 2

X-Ray Factors Affecting Image Quality

1.	X-Ray source size	(R)
2.	Kilovoltage & beam quality	(C)
3.	Film density of image	(C)
4.	Processing	(C)
5.	Exposure Geometry	(R,C)
6.	Film/screen type	(R,C)
7.	Operator care & experience	(R,C)

The radiographer has some degree of control over all of these variables except probably items (1) and (5). The notation R and C indicate that the parameter is identified more with resolution (R) or contrast (C) although in the final analysis these two quantities cannot be completely uncoupled. Resolution and contrast are the two main factors contributing to overall image quality. These will be addressed in a general way in the text following.

Image Resolution

There are several ways to define or describe image resolution. In optics, there is usually done in terms of the degree of separation of the intensity distribution of two light sources. We choose to define it, for present purposes, in terms of "unsharpness". This is sometimes given the more formal designation - radiographic definition. We note of course that the modulation transfer function is another way to define or measure this quantity but unsharpness has the advantage of general acceptance in industrial radiographic facilities. Therefore, we will continue to use this term in this paper.

Unsharpness refers to how well a sharp edge of an object is imaged by the radiographic system. Scanning across the image with a precision microdensitometer produces a trace which is referred to as the edge spread function. The width of the edge spread along the "X" axis (abscissa) is defined as the "unsharpness" of the image [3]. The degree of "edge spread" is a function of several quantities. These include x-ray source size, specimen thickness and source to film distances (geometric factors) and film type, presence or absence of intensifying screens, film/screen/specimen contact and object or x-ray source motion. The geometric quantities are generally not controllable. The latter quantities to a large extent are controllable. Exposure geometry is usually dictated by the system to be radiographed. In a later section, under Flaw Detection and Sizing, the relationship of unsharpness to dimensioning will be considered.

Image Contrast

The radiographer can exercise some control over contrast. Film contrast can be improved by proper choice of kilovoltage, film, and exposure. Consider Figure 1 for example, where the ordinate (gradient) is the contrast produced on the film as a function of image density. A denser

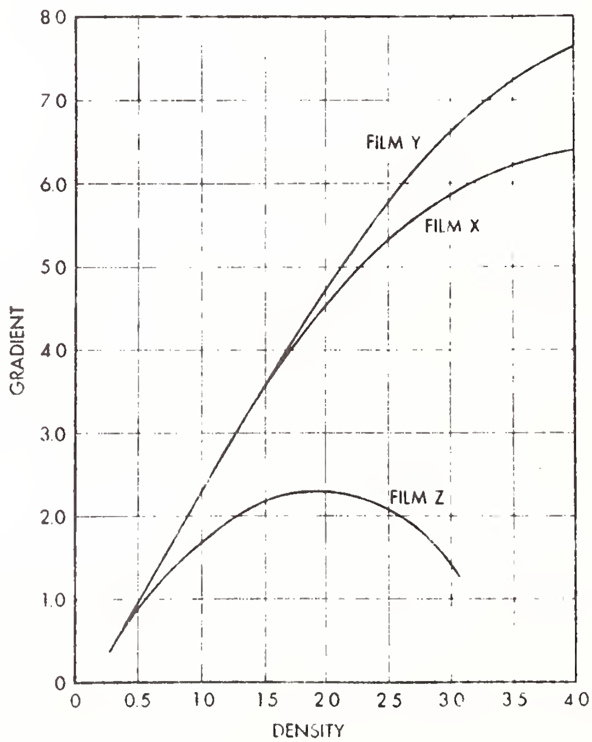

Fig. 1. Density vs. Gradient for several film types. Reference "Radiography in Modern Industry"

radiographic image has better contrast for a given film type than the same image at a lower density. For this reason, most industrial radiographs are exposed to achieve densities of 2 or greater.

Another factor affecting image contrast is beam quality which relates to the kilovoltage applied to the x-ray tube. A more accurate designation of beam quality is the photon energy spectrum produced by the x-ray tube. The effect of beam quality on contrast is quite easily seen if one considers a typical exposure chart for radiographing a material. On these charts or graphs exposure is plotted as a function of material thickness for different kilovoltages (see Figure 2). Taking the exposures required

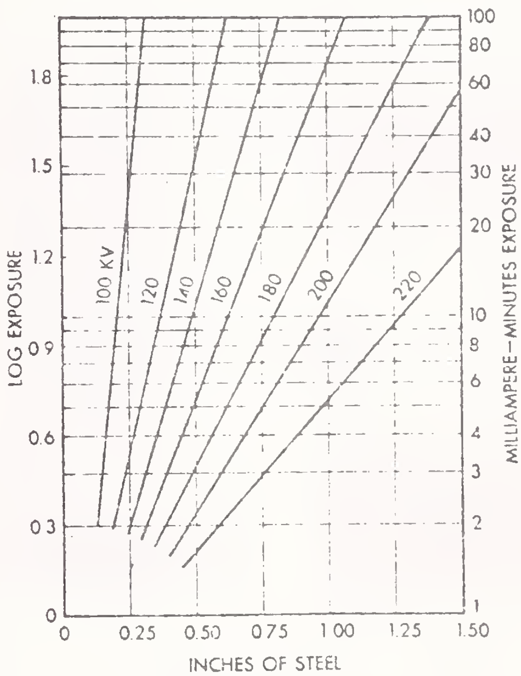

Fig. 2. Typical exposure chart for steel with kilovoltage as a parameter. Reference - "Radiography in Modern Industry".

for two steel thicknesses for example, we can construct a table as follows:*

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*Radiography in Modern Industry, Third Edition, pg. 68.

kV	Steel Thickness	Exposure for a Given Density	Relative Intensity	Intensity Ratio
160	3/4"	18.5 mAM	3.8	3.8
	1"	70.0 "	1.0	
200	3/4"	4.9 "	14.3	2.5
	1"	11.0 "	5.8	

The higher intensity ratio at the film implies a greater contrast for the lower kilovoltage but at a considerable cost in exposure needed.

Processing also affects contrast as is shown in Figure 3. In general, it is wise to follow the film manufacturer's recommendations on processing since they will try to optimize the contrast sensitivity for their own films. In general, the base and fog level of the film is generally affected adversely by greater processing time or temperature and this can affect image quality or x-ray sensitivity.

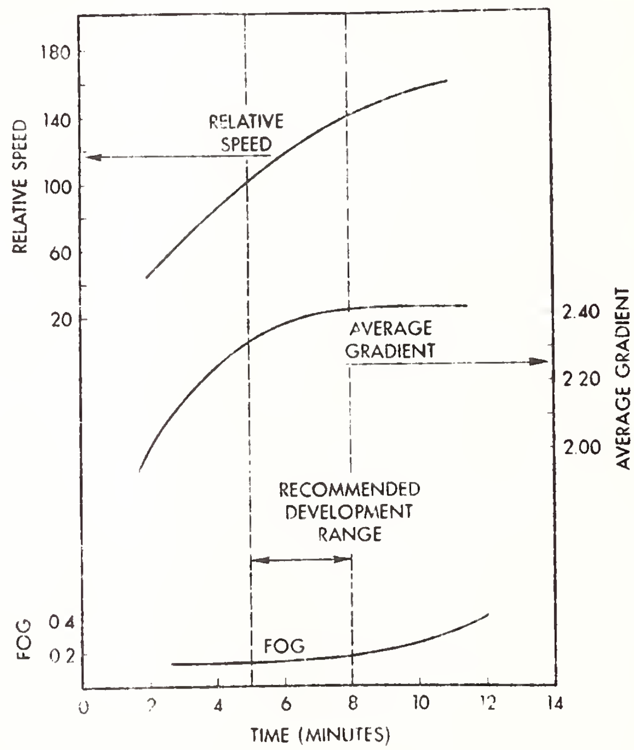

Fig. 3. Relative speed, average gradient, and fog level as a function of manual processing time for a typical film. Reference "Radiography in Modern Industry".

Since the factors just described also affect total exposure time for a given specimen, this must be taken into account in the overall selection process.

Flaw Detection and Sizing

Weld flaw detection and analysis is considerably improved by reader training and experience. Scanning girth weld radiographs requires a high degree of concentration in order to reduce the probability of not detecting a flaw because of a momentary attention lapse. Flaw detection is also complicated by the weld bead ripple in the image which can mask a serious flaw or render its sizing more difficult. Such a flaw can be missed on a field radiograph, particularly if its exposure geometry is such as to place it in a shaded region of the weld image. Figures 4 and 5 are illustrations of the failure to detect a flaw - in this case a crack - on the field radiograph. This flaw was later revealed in a laboratory radiograph and in radiographs taken after the weld root and crown overbead had been removed.

Fig. 4. Photo of typical section of a field radiograph. There is a crack in this area.

Fig. 5. Crack in weld section shown in Fig. 4. This was not revealed until root and crown overbead were removed.

Most of the other types of weld flaws are more easily detected and sized by radiography. In fact, for three dimensional flaws, such as slag and porosity, radiography is the method of choice. There are several procedures that have been developed for flaw sizing. These include:

1. Comparative Visual Reference System [3] [5]
2. Direct Densitometry [4] [5]
3. Penetrameter/Flaw Visual Reference Method [5]

The comparative visual reference system utilizes a film step wedge which relates image contrast to change in metal thickness. The details of producing the step wedge are discussed in reference [3]. The probable accuracy of the method is discussed in [5]. Direct densitometry is used when the flaw is large enough in projected area to exceed the size of the densitometer aperture. One can relate four densitometry readings to the flaw depth by using the formula cited in the reference. Two of the readings are taken on and adjacent to the penetrameter/shim image and two are taken on and adjacent to the flaw. The third method listed above is really a variation of method number 1. It requires a very experienced reader who can relate the contrast of the penetrameter/shim image, relative to the base metal, to the contrast of the flaw relative to its surroundings. Fortunately, most flaws relative to their surroundings fall in the same range of contrasts as is produced by the penetrameter/shim thickness contrast relative to the base metal.

The basic problem with the comparative reference system is the calibration of the film step wedge. This should be done at the same time and location as the field radiographs but generally this is not the case. Later exposures in the laboratory attempt to simulate the field conditions but some are not duplicable. Therefore, to assess accuracy of the methods used when the original exposure conditions are not known, requires a series of empirical tests similar to the following recipe.

For a given geometry there are four parameters to assess; kilovoltage (beam quality), processing, film type and x-ray exposure (beam current/exposure time product). Therefore, each parameter is assessed independently to the extent this is possible. For example:

1. Fix processing, film type, x-ray exposure and vary kilovoltage. Assess contrast changes on image.

2. Fix kilovoltage, processing, x-ray exposure, and vary film type. Assess contrast changes on image, etc.

Some of these steps can be eliminated if reliable documentation of the exposure conditions in the field is available. However, once these assessments have been made and the reader of the field radiograph has determined the contrast for a flaw (relative to its surroundings) it is possible to estimate how much this contrast would change due to the variables cited above.

Unsharpness, which was previously considered in the section on resolution, is a geometric condition which compounds the analysis described above. In the case of girth weld radiography with an internal source this quantity can be significant and must be considered in the analysis. For example a 40 inch diameter pipe with 5/8 inch wall thickness when radiographed with an x-ray source whose size is 5 mm (.2") will yield an unsharpness value of .006" based on the formula given in ASTM E-94 [6]. This can be a serious problem for sizing defects that are (e.g.) of the order of .040" in width, length or depth. Note that API 1104 specifies a 2T image quality on pipeline inspection unless otherwise specified. This implies the ability to detect a flaw of the order of .025" for a wall thickness of 5/8" or .050" if 4T quality is acceptable. Yet the unsharpness can be of the order of 10 to 25 percent of these values simply due to geometrical restrictions. The only parameter available to reduce the geometrical unsharpness is the source size which is generally an option not available on field x-ray systems.

This is a brief discussion of some of the problems connected with flaw detection and sizing. Our goal is to reduce the variability presently existing in the radiographic inspection of pipeline girth welds when flaw sizing is a critical need.

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DEVELOPMENT OF WELDING CONSUMABLES FOR ARCTIC PIPELINES

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Abstract

In order to meet the severe requirements of Arctic pipe quality, a series of new submerged arc welding fluxes and wires were developed over the last several years for commercial line pipe production. Additional efforts are now underway at Linde's Materials Technology Laboratory to improve upon these new fluxes and wires.

The current specification calls for weld metal impact requirement of 70 ft. lbs at 0 °F. In order to achieve this in practical commercial applications, pipe manufacturers are seeking weld metal impact values of the order of 80 to 85 ft. lbs. at 0 °F. Such high impact properties of weld metal have to be met in multiple wire, high current, high speed submerged arc welding conditions which is typical of the current pipeline seam welding practice.

In order to attain such a goal it is imperative that control of weld metal microstructure (achieved by controlling heat input and chemical composition of weld metal) and substantial reduction of inclusion level in the weld metal are essential. Meeting the high weld metal impact property requirements is further complicated by high dilution from the base metal (60 to 70 percent is typical) which introduces some undesirable elements into the weld metal.

This presentation covers various experimental investigations from slag-metal reactions to weld metal mechanical properties of HSLA line pipe steels for Arctic application.

SESSION V

MOTOR CARRIERS

Chairman: K. Pierson, Bureau of Motor Carrier Safety

1978 ROADSIDE VEHICLE INSPECTIONS[1]

Presented by

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Washington, D.C. 20590

Foreword

The safety inspection of vehicles and drivers, conducted during periods when actual highway transportation operations are underway, is one of the activities of the Federal Highway Administration's Bureau of Motor Carrier Safety intended to increase the safety of interstate commercial motor vehicle transportation on the Nation's highways. This report is a compilation of the results of the Bureau's roadside vehicle inspection activities conducted at various locations throughout the country during calendar year 1978.

Introduction

The Federal Motor Carrier Safety Regulations (FMCSR) and the Hazardous Materials Regulations (HMR) are applicable to the transportation operations of motor carriers of passengers or property engaged in interstate or foreign commerce. The Regulations are designed to promote public safety by minimizing the risks associated with these transportation operations and thus contribute to efforts to reduce highway accidents. The Federal roadside safety inspection program is designed to identify vehicle defects and driver conditions which could cause accidents, and to remove vehicles and drivers (that are deemed imminently hazardous) from the highways.

Roadside vehicle/driver safety inspections may be conducted by one Federal Motor Carrier Safety Investigator working with State authorities, or the inspections may be on a larger scale, with several Federal and State personnel working in one or more locations. Occasionally, other elements of the Department of Transportation as well as other Federal agencies participate--such as the Department of Defense, the U.S. Coast Guard, the Federal Bureau of Investigation, the Nuclear Regulatory Commission, and the U.S. Customs Service[2].

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[1]This paper is reprinted from a publication of the Bureau of Motor Carrier Safety, Federal Highway Administration, Washington, DC 20590.
[2]Certain vehicles and drivers, because of either the type(s) of cargos or area of operations are exempted from certain Federal safety requirements; e.g., "exempt intracity" operations.

Vehicles may be inspected on either a selective or random basis. The selective method involves a visible and audible prescreening by the Federal inspector to determine if there are obvious signs of defects. If during this cursory examination a deficiency is noted, a thorough inspection of the vehicle and driver documentation is performed. The selective prescreening inspection procedure is intended to identify as many defective vehicles as possible during the inspection period. The prescreening inspection procedure is not intended to produce statistically valid measurements about the number or percent of defective vehicles on the highway. Vehicles may also be inspected on a random basis to provide statistically valid data for use in determining Federal Motor Carrier Safety Program emphasis areas, and for statistical reports. Inspectors are instructed to perform thorough inspections on vehicles in accordance with a preplanned vehicle selection procedure designed to insure that the vehicles are selected on a random basis.

Regardless of the selection procedure used, vehicles found to have defects sufficient to warrant being classified as imminently hazardous to public safety on the highway are declared out-of-service, and may not be operated until the serious defects have been repaired. If a vehicle is inspected and found to have lesser defects that are not imminently hazardous, those defects are noted on the inspection report and may be repaired as soon as possible away from the inspection site.

In addition to the inspection of the vehicle, the required credentials and documents of the driver are also inspected. Most drivers[3] are required by Federal regulation to have a valid operating license issued by the State authority, a valid medical certificate, and a current driver's daily log.

The driver's log contains entries showing the number of hours worked, the types of duties performed, the mileage traveled, and other identifying information. The purpose of the daily log is to assure compliance with Federal regulations limiting the driver's hours of service, and to reduce the risk of accident involvement due to excessive hours on duty or driving. Drivers who are found to be in violation of Federal hours of service limitations at the time of the roadside inspection are placed out-of-service.

In all cases where the inspection report notes defects or driver violations, the motor carrier is required to certify within 15 days to the Bureau of Motor Carrier Safety that the defects have been corrected.

[3]Drivers operating in "exempt intracity" operations or who are driving vehicles with a gross weight of not more than 10,000 pounds and not more than two axles are exempt from some or all of these requirements.

The Bureau of Motor Carrier Safety (BMCS), in continuing its effort to improve highway safety, conducted 27,601 roadside vehicle inspections during calendar year 1978. The total number of vehicles inspected during this period represents an increase of 87 percent over calendar year 1976, and an increase of 50 percent over the number of inspections performed in calendar 1977. All vehicles inspected during 1978 were selected for inspection using the selective prescreening procedure.

Inspections of Interstate Property Carriers--Trucks

Ninety-four percent (25,965) of all vehicles inspected were operated either by for-hire or private carriers of property. For-hire carriers of property are classified into two types, "authorized" and "exempt." Authorized carriers are those who perform transportation services authorized by a certificate or permit issued by the Interstate Commerce Commission (ICC). Exempt carriers are those who transport for hire those commodities that are exempt from ICC economic regulations. In most cases these commodities are agricultural products, including produce, livestock, and grain. Private carriers of property are those who engage in transportation as an adjunct to another business enterprise.

An additional classification of carriers, "other," used in the report includes those transporting the mails operating under contract with the U.S. Postal service, carriers in foreign commerce, and carriers whose classification was unknown at the time of the inspection.

Distribution of Out-of-Service Defects by Carrier Type

Of the total 25,965 property carrying vehicles inspected, 10,423 (40 percent) were found to have one or more serious mechanical defects which resulted in the vehicle being marked and declared out-of-service by the BMCS inspector. Table 1 displays the distribution of the out-of-service actions by carrier type, and the average number of defects per vehicle.

Table 1--Out-of-Service Actions by Carrier Type

Carrier Type	Total Vehicles Inspected	Total Out-of-Service	Average Number of Defects per Vehicle
Authorized	11,424	4,805	3.37
Private	9,990	3,667	3.38
Exempt	1,871	809	3.85
Other	2,680	1,142	3.86

The average number of defects discovered per vehicle inspection in calendar 1978 was 3.46. In calendar 1976 the average number of defects discovered per vehicle was 3.23, and in calendar 1977 the average number of defects per vehicle was 3.31.

Defects Discovered by Type and by Classification of Carrier

The 25,965 property carrying vehicle inspections disclosed a total of 89,998 defects, 16,226 of which are classified as being imminently hazardous and out-of-service. The greatest number (36,385) of defects discovered was in the lighting/electrical systems of these vehicles, and 2,191 (6 percent) of these lighting/electrical defects were serious enough to cause the vehicle to be placed out of service. The largest number of out-of-service defects was found in the vehicles' service brake systems. Of the total of 17,293 defects discovered in the service brake systems, 8,128 (47 percent) were out-of-service. The largest percentage of out-of-service defects within the service brakes related to brake hoses or lines (33 percent), followed by inoperative brake energy warning devices (21 percent), leaking brake chambers (10 percent), and cracked brake drums (4 percent). Table 2 shows the distribution of the out-of-service and less serious defects by defect type and carrier classification.

Inspection of Interstate Passenger Carriers--Buses

Six percent (1,636) of the vehicles inspected were operated by carriers of passengers. Passenger carriers are classified into two types, "authorized" and "exempt." As in the case with property carrying vehicles, authorized passenger carriers are those who perform a for-hire service authorized by a certificate or permit issued by the ICC. Exempt or other carriers of passengers are those performing a for-hire service which is exempted from the requirements relating to operating authority by certain parts of the Interstate Commerce Act, and those whose classification was unknown at the time of the inspection.

Distribution of Out-of-Service Defects by Carrier Types

Table 3 displays the distribution of out-of-service actions by carrier type and the average number of defects per vehicle.

Table 3--Out-of-Service Actions by Carrier Type

Carrier Type	Total Vehicles Inspected	Total Out-of-Service	Average Number of Defects per Vehicle
Authorized	1,398	122	0.79
Exempt/Other	238	26	1.06

Table 2--Distribution of Defects by Type and Carrier Classification

Descriptions of Violations	All Carriers		Authorized		Private		Exempt		Other	
	Total	Out of Service	Total	Out of Service	Total	Out of Service	Total	Out of Service	Total	Out of Service
Coupling Devices	397	54	175	23	171	23	26	5	25	3
Emergency Equip.	8,539	---	2,859	---	3,814	---	760	---	1,106	---
Exhaust System	1,830	715	1,007	400	519	206	106	29	198	80
Frames	206	32	102	18	65	6	12	---	27	8
Fuel System	706	345	311	151	256	123	40	19	99	52
Glazing/Windows	1,916	---	844	---	630	---	214	---	228	---
Hazardous Matls.	3,691	315	1,473	143	1,831	135	46	10	341	27
Lighting/Elec.	36,285	2,191	14,670	889	14,901	837	2,891	187	3,823	278
Other Parts/ Accessories[1]	4,293	388	1,946	239	1,265	48	381	24	701	77
Parking Brake	488	237	185	71	220	126	31	16	52	24
Power Source	1,551	208[2]	827	105	417	61	107	14	200	28
Restraints[3]	318	135	111	40	152	75	19	6	36	14
Service Brakes	17,293	8,128	8,315	3,811	5,684	2,704	1,277	621	2,017	992
Steering System	671	231	322	114	194	62	47	22	108	33
Steering Axle Tires	284	57	115	24	93	14	29	6	47	13
Suspension System	1,967	635	864	285	674	206	192	75	237	69
Tires--Non-Steering Axle	4,476	1,474	2,136	710	1,356	440	413	123	571	201
Wheels	5,087	1,081	2,297	492	1,620	344	624	134	546	111
TOTAL	89,998	16,226	38,559	7,715	33,862	5,410	7,215	1,291	10,362	2,010

[1]Other Parts/Accessories include speedometer, horn, defrosters, mirrors, seat belt installation, windshield wipers.

[2]Engine of vehicle unable to start without external assistance.

[3]Restraints: Cargo restraints including front end structure, tie-down assemblies, and blocking/bracing.

The average number of defects discovered per vehicle inspection in calendar 1978 was 0.83. In calendar 1976 the average number of defects was 0.96, and in calendar year 1977 the average number of defects per vehicle was 1.08.

Table 4 shows the distribution of the out-of-service and less serious defects by defect type and carrier classification.

Table 4--Distribution of Defects by Type and Carrier Classification

Description of Violations	All Carriers Total	All Carriers Out of Service	Authorized Total	Authorized Out of Service	Exempt & Other Total	Exempt & Other Out of Service
Parking Brakes	4	3	3	2	1	1
Service Brakes	157	84	138	71	19	13
Bus Requirements[1]	249	1	178	1	71	--
Emergency Equip.	168	--	128	--	40	--
Exhaust System	13	11	11	9	2	2
Fuel System	3	1	2	--	1	1
Window Construction	119	--	107	--	12	--
Hazardous Matls.	1	--	1	--	--	--
Lighting/Electrical	414	7	354	6	60	1
Power Source	19	7[2]	14	6	5	1
Steering System	4	1	2	--	2	1
Suspension	5	4	5	4	--	--
Tires-Non-Steering Axle	46	4	36	4	10	--
Tires-Steering Axle	6	1	5	1	1	--
Wheels	80	14	61	14	19	--
Other Parts/ Accessories[3]	73	4	63	4	10	--
TOTAL	1,361	142	1,108	122	253	20

[1]Bus Requirements include standee line, notices to passengers, and light on emergency door.
[2]Engine of vehicle unable to start without external assistance.
[3]Other Parts/Accessories include speedometer, horn, defrosters, mirrors, seat belt installation for the driver, and windshield wipers.

Driver Violations

Inspection of the property and passenger carriers' drivers during calendar year 1978 disclosed 24,481 violations of the FMCSR. The largest percentage of the violations (52 percent) related to failure to comply with one or more of the requirements of Part 395 of the FMCSR, Hours of Service of Drivers. The largest single type of hours-of-service violation was the failure to maintain a driver's daily log (4,627 violations), followed by failure to have a current daily log (3,521 violations), and "other" log violations such as incomplete entries and failure to have the previous 30 days' logs in possession.

Although private carriers of property accounted for only 36 percent of the total number of vehicles inspected (FMCSR administered by BMCS are not applicable to private carriers of passengers), 47 percent of the total driver violations discovered are attributed to drivers of private property carrying vehicles. In addition, these drivers accounted for 56 percent of the most frequently found violation--failing to maintain a driver's daily log. Examination of drivers' daily logs for the 30 days prior to the time of the inspection disclosed 2,117 instances where the driver had violated Federal regulations which prescribe the maximum number of hours drivers may drive or remain on duty. Of the total number of hours of service violations, 1,522 instances had occurred on tours of duty prior to that during which the inspection was performed. Twenty-eight percent (595) of the drivers who had violated the Hours of Service Regulations were discovered to be operating vehicles in excess of the maximum number of hours permitted at the time of the inspection and were consequently placed out-of-service by the BMCS investigator at the time and place of the inspection.

Table 5 shows the distribution of driver violations by type and carrier classification.

Vehicle Defects and Accident Involvement

Motor carriers of passengers and property are required by the FMCSR to report to BMCS those accidents which resulted in (a) estimated property damage to the vehicles, cargo, and fixed objects of $2,000 or more, or (b) a personal injury, or (c) the death of any person. In calendar year 1978, 1,597 accidents were reported to BMCS as having occurred as a result of a mechanical defect or failure. Of the 1,597 accidents involving mechanical defects reported in 1978, those defects cited most often were defects in the brake system (496 or 31 percent), followed by tires (429 or 27 percent), steering (83 or five percent), wheels (82 or five percent), and coupling devices (80 or five percent). In 1978, those accidents attributed to mechanical defects resulted in 124 fatalities, 1,305 injuries, and $19,983,560 in property damage. Inspection of commercial vehicles in operation during calendar year 1978 showed that the five most common mechanical defects which caused accidents during that year (brakes, tires, steering, wheels, and coupling devices) accounted for 31 percent of the total number of defects discovered during BMCS roadside inspections, and resulted in 70 percent of the total number of out-of-service actions taken by BMCS staff members to remove defective vehicles from the highways.

Table 5--Driver Violations by Violation Type and Carrier Classification

Descriptions of Violations	All Drivers Total	All Drivers Out of Service	Authorized Total	Authorized Out of Service	Private Total	Private Out of Service	Exempt Total	Exempt Out of Service	Other Total	Other Out of Service
10 Hours	1,281	278	441	118	512	91	189	50	139	19
15 Hours	425	108	139	35	219	50	40	17	27	6
60 Hours	51	15	14	6	28	9	3	--	6	--
70 Hours	360	194	140	77	141	73	48	26	31	18
No Logs	4,627	--	924	--	2,626	--	516	--	561	--
Logs not Current	3,521	--	1,861	--	1,044	--	290	--	326	--
False Log Entries	328	--	170	--	88	--	33	--	37	--
Other Log Violations[1]	2,336	--	781	--	1,000	--	246	--	309	--
Medical Certificate Violations	5,761	--	923	--	3,546	--	540	--	752	--
Other Driver Violations[2]	5,791	--	2,122	--	2,513	--	518	--	638	--
TOTAL	24,481	595	7,515	236	11,717	223	2,423	93	2,826	43

[1]Other log violations include failure to prepare logs in the form and manner prescribed and failure to have previous 30 days' logs in possession at time of inspection.
[2]Other driver violations include failure to use seat belts, transportation of unauthorized passengers, and failure to comply with State and local driving laws.

Conclusion

Since 1975 the percentage of vehicles inspected by the BMCS and found to have imminently hazardous mechanical defects has increased approximately three percent per year. The majority of these serious defects can be readily detected by visual/audible inspection, and the elimination of these defects can have an appreciable impact on the Federal effort to reduce commercial vehicle accidents. In addition to a commitment to increase the Federal inspections of commercial vehicles in operation during 1979, BMCS recently instituted rulemaking to revise the Federal Safety Regulations to address commercial vehicle drivers' failure to prepare logs and inadequate inspection, repair, and maintenance procedures.

The largest single violation on the part of commercial vehicle drivers is the failure to maintain a driver's daily log. The failure to maintain this record of a driver's hours of service enables those drivers who are operating in excess of the maximum number of hours permitted to defeat attempts by Federal inspectors to determine the actual hours of service of the driver. Effective June 18, 1979, all drivers who are required to prepare logs will be placed out-of-service for a minimum of eight hours if a currently-maintained driver's log is not available for inspection during the course of Federal driver-vehicle roadside checks.

The present hours of service limitations contained in the Federal Regulations have been the subject of extended analysis and seven public hearings held in major cities throughout the United States. The next step in the rulemaking process, the publication of a Notice of Proposed Rulemaking, is planned for the Fall of 1980.

In addition to making the hours of service rules more effective, Part 396 of the Federal Motor Carrier Safety Regulations relating to the inspection, repair, and maintenance of interstate commercial vehicles has been extensively revised to insure more adequate inspection and maintenance procedures. This revision of the Regulations will become effective on April 1, 1980.

The Bureau of Motor Carrier Safety intends to continue its efforts in deterring poorly maintained vehicles and fatigued drivers from traveling the highways by expanded safety program monitoring, increased enforcement, and the training of State personnel in inspection procedures.

REGULATION, LEGISLATION AND THE TEAMSTERS

An Equation for Highway Safety

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Introduction:

"In 1978, 5,075 Americans were killed in accidents involving heavy trucks, an alarming increase of 53 percent over 1975. During the same period, the number of all highway fatalities increased 12.6 percent, in itself a significant climb.

"Likewise, fatalities to occupants of heavy trucks numbered 1,010, an increase of 41 percent from 1975. Fatalities to occupants of passenger cars rose only 7 percent during that period. This shocking mortality rate makes truck driving one of the more dangerous occupations in America."[1]

These words of Senator Charles Percy in introducing the "Truck Safety Act" last year echo the concern raised by all facets of the transportation community over the epidemic increase in heavy truck fatalities. The government, vehicle manufacturers, carriers and organized labor have all recognized the crying need for concrete methods of reducing this ominous trend.

Scope of Problem:

The problems created by heavy truck accidents are not limited to heavy trucks. Of the 50,000 highway fatalities recorded in 1978, 17,578, or nearly 35 percent, involved a truck or bus of any size or weight.[2] Heavy trucks represent a significant fraction of that total.

What are the reasons for this rapid increase in fatal accidents involving trucks? In complex transportation issues like this one, there are as many different theories as there are people available to give them. However, of particular interest to this group is the nature and extent of equipment failure as a factor in these highway accidents.

Whether an accident was caused by a vehicle defect is often difficult to determine, especially in the case of a driver fatality. Estimates of defect involvement range from a high of 30 percent down to one or two percent. A Department of Transportation study of large truck accidents in Texas found that in 1973, about 6.4 percent of the vehicles involved were reported by the police to have defects that were connected with their accident.[3] This is opposed to a two percent defect rate for passenger cars from the same data.

The study went on to point out, "the most numerous defects reported were brakes, tires and wheel failures-about 70 percent of all defect accident cases. Note that all of these are failures in relatively visible components of the vehicles."[4]

This high rate of defect involvement in accidents is alarming evidence of improper or inadequate maintenance procedures on the part of carriers. Even more startling is the recent experience of the Bureau of Motor Carrier Safety (BMCS) in conducting roadside inspections of heavy trucks.

The Bureau has the responsibility of enforcing the Federal Motor Carrier Safety Regulations. Among other things, these standards set out the minimum requirements for necessary parts and accessories and for inspection and maintenance of heavy commercial vehicles. Upon discovering a vehicle in a roadside inspection which does not meet specific criteria for safe operation (for example, a leaking air brake hose), the BMCS inspector will place the vehicle "out of service" until the defect is repaired. This means that the vehicle may not be moved (except to tow it away from the inspection site for repairs), until it is placed in safe operating condition.

In the latter half of 1978 through March 1979, BMCS inspected over 2,500 vehicles during special spot inspection throughout the country. Of this number, 46 percent were placed out of service.[5] It must be remembered that vehicles selected for inspection are usually first given a cursory visual check. If the vehicle has a noticeable defect, it is then moved to a safer location for a more thorough inspection. Therefore, the argument is made that only the "worst" vehicles are inspected.

But during that same period, a total of 307 vehicles were selected for inspection completely at random. Of that group, fully 34 percent were placed out of service. And in a recent roadcheck of over 1,700 vehicles traveling in the Mississippi River area, 44 percent of those inspected were deemed unfit for service.[6]

Defects in the brake system accounted for the greatest number of out of service violations, followed by tire and electrical system deficiencies. In the Mississippi inspections, faulty brakes made up 56 percent of the total out of service violations.[7]

What is the significance of these figures? They mean that for each truck passed on the highway, there is a better than one in three chance that it has a serious defect in a vital safety related system. If that truck is involved in an accident as a result of the defect (or for any other reason), it is likely that another vehicle will be involved, usually a passenger car.[8] And in accidents of that nature, cars come out on the short end of the stick.

In collisions between cars and tractor-trailers resulting in a fatality, one study has shown the car occupant death rate to be ten times higher than the truck rate.[9] Another review of several studies has shown that for each truck occupant fatality in crashes involving trucks, there are 30 to 40 other highway users killed.[10]

The bottom line is that trucking accidents are not only of grave concern to our organization as representatives of the truck driver, but also must be the concern of all parties interested in highway safety.

With the scope of the problem evident, the question now is, how to keep defective and unsafe vehicles off the road? The title of this paper refers to three distinct groups: the Executive Agency, the

Legislature, and the Union. All three play a vital role in arriving at solutions to the problems presented by mechanical failures. None by itself can adequately deal with all aspects of the problem.

To further explore this concept, it is helpful to examine a few specific examples in which the interaction of these groups is leading to concrete methods of preventing mechanical failures due to vehicle defects.

The Executive Agency:

A recent Department of Transportation(DOT) study on preventive maintenance isolated four factors necessary for the alleviation of commercial vehicle defects:[11]

1. Detection of Defect
2. Communication to Responsible Party
3. Repair
4. Reporting to the Detector that Repair has been Effected.

A post-trip inspection by the driver and his completion of a vehicle condition report were, at the time of the study (1975), the sole required general inspection duties of the carrier. Accordingly, the DOT study recommended both a pre and post-trip inspection, preferably conducted by the driver, as a method of detecting defects and communicating them to the carrier, (steps 1 and 2 detailed above).

The DOT study also suggested that the driver's post-trip inspection report, complete with notation of maintenance activity, be carried on the power unit for the review of the next driver.

In light of these recommendations, and in response to our experience as representatives of hundreds of thousands of truck drivers, the Teamsters Union formally petitioned BMCS for a revision of the Federal Motor Carrier Safety Regulations concerning maintenance, inspection, and repair.

Specifically, we asked BMCS to require a pre-trip visual inspection in addition to the already mandated post-trip inspection; to require that the post-trip inspection report be reviewed by maintenance personnel and that listed defects be corrected, with the action taken noted on the inspection report; and that the completed inspection report be then placed on the power unit for the review of the next driver.[12]

The chronology of subsequent events is a case study in the oftentimes frustrating regulatory process. BMCS responded to our June, 1976 petition by issuing an Advanced Notice of Proposed Rulemaking in April of 1977. This Notice explained the scope of the problem and asked interested parties for comments and suggestions.[13]

After lengthy and laborous study of the numerous comments received in response to the Notice, BMCS promulgated a Final Rule on July 2, 1979, to take effect on August 31, 1979.[14] The final rule incorporated the major recommendations of our petition. However, the industry was concerned that aspects of the new rule would raise operating costs without enhancing safety. To allow time for consideration of their protests, BMCS delayed the effective date four months to December 31, 1979.[15] A minor change was made in the wording of the regulation*, and BMCS again delayed the effective date to allow carriers to bring their operations into compliance, this time establishing April 1, 1980 as the new date.[16]

By this time, all parties appeared satisfied with the present state of the regulation. We felt that our petition was adequately addressed and most carriers found they could comply with only minimal changes in their recordkeeping and maintenance operations.

However, less than a month before the rule was to take effect, the United Parcel Service (UPS) went into Federal Court asking for both a review of the regulation and a stay of the effective date pending review.[17] UPS felt that the requirement that the post-trip inspection report be placed aboard the power unit would be prohibitively expensive and unnecessary for their type of operation, and that the driver's need for that information would be best served by allowing him daily access to the report at the dispatch site. The Department of Transportation responded and the Teamsters intervened in the suit, pointing out to the court both the logistical problems presented by the UPS proposal and the well documented advance in highway safety which the new regulations would bring about.

The case is still pending in the Circuit Court of Appeals, but the court did deny the UPS motion for a temporary stay. Therefore, the new regulations took effect the first of this month.

This rulemaking is an example of how the Teamsters Union and governmental agencies can work hand in hand to try to solve some of the problems presented by mechanical failures in trucks. We have also worked closely with various agencies on a wide range of issues, including steering, tire loading, air brake systems, and overall vehicle integrity. But regulation is only one entry in our "highway safety equation".

Role of Legislature:

Some problems are best addressed through legislation. An example of such a problem is the limited scope of the authority of BMCS. The authority of BMCS in enforcing the Federal Motor Carrier Safety Regulations is basically restricted to placing violating vehicles out of service, although they may levy civil penalties for drivers who fail to properly maintain a daily log as required.

This method of enforcement has proven to be inefficient and ineffective. Even using their resources as best possible, BMCS was able to inspect less than one percent of three million commercial vehicles under its purview in 1978.[18] This low inspection rate provides no credible deterrent for unscrupulous carriers who try to increase profits at the expense of safe vehicle maintenance. Faced with only the pro-

* The requirement that drivers "certify" that repairs had been made was deleted because "the certification of repairs must be done by a mechanic with skill and experience in the technical areas" and because the requirement would "place an unfair burden on drivers" (44 Fed. Reg. 76525).

spect of occasionally repairing a vehicle placed out of service by a BMCS inspector at a roadside station, some carriers adopt a "break-down" maintenance policy, repairing vehicles only when they actually cease to operate.

The authority of BMCS in enforcing their regulations is limited by their enabling legislation, passed in 1937. Therefore, in order for them to obtain improved enforcement authority, it was necessary to repair to the legislature.

Over two years ago, the Teamsters Union entered into discussion with Senator Charles Percy (R.-Ill.) concerning the feasibility of introducing legislation to allow BMCS to more effectively enforce their own regulations.

The fruit of those discussions was S. 1390, the Truck Safety Act of 1979. The bill in part allows the Secretary of Transportation (presumably through BMCS) to levy civil penalties of up to $10,000 for non-compliance with federal safety regulations. In addition, it authorizes the Secretary to prosecute persons who knowingly and willfully violate the standards, and provides a maximum jail term of one year and a maximum fine of $25,000 upon conviction of such violators.

The bill addresses several other important issues which are not of great interest here, such as the working conditions of drivers. But one other provision of the bill stands to substantially reduce the number of defective vehicles on the highway. It would protect trucking employees from discipline, discharge or discrimination if they refuse to drive an unsafe vehicle or if they report safety violations to state or federal authorities.[19]

There presently exists a similar provision in the National Master Freight Agreement, negotiated between the Teamsters and the carriers.[20] While over one-half million trucking employees are covered by that agreement or one with a similar provision, there are thousands of drivers who don't enjoy the benefits of a Teamster contract and need the protection provided by this piece of legislation.

The Truck Safety Act passed the Senate by an overwhelming margin in March, and is currently under consideration in the House. Its passage there will mark a great stride toward safer trucks.

A number of other issues are before the legislature which may have an impact on the vehicle defect problem, including the relaxation of economic regulation in the trucking industry and length and weight laws. Accordingly, the Teamsters Union attempts to put forth the concerns of our membership whenever legislation affecting their interests is contemplated.

The Union Role:

The hand of the government, either executive or legislative, is sometimes necessary to reach solutions to our problems. But often problems can be solved without the aid of the state, through collective bargaining.

Long before the introduction of legislation providing protection against discrimination for employees who refuse to drive unsafe vehicles, the National Master Freight Contract had such a provision. To take another example, just as a BMCS inspector can place a vehicle "out of service", many contracts allow a Teamster business agent to "red tag" an unsafe vehicle with much the same effect.

The collective bargaining process can deal with issues at a speed seldom seen in either the legislature or the executive branch. One instance of this ability centers around the controversy over the safety of the so-called "cab-under" tractor-trailor configuration.

When the Teamsters Union first learned of the development of a truck design featuring the driver's compartment under the cargo unit, the initial reaction among officers with truck driving experience was that such a vehicle would be unsafe for a variety of reasons, including lack of forward visibility and vehicle crashworthiness. However, those conclusions were based on a "truck driver's common sense" approach to the issue, with little real engineering basis.

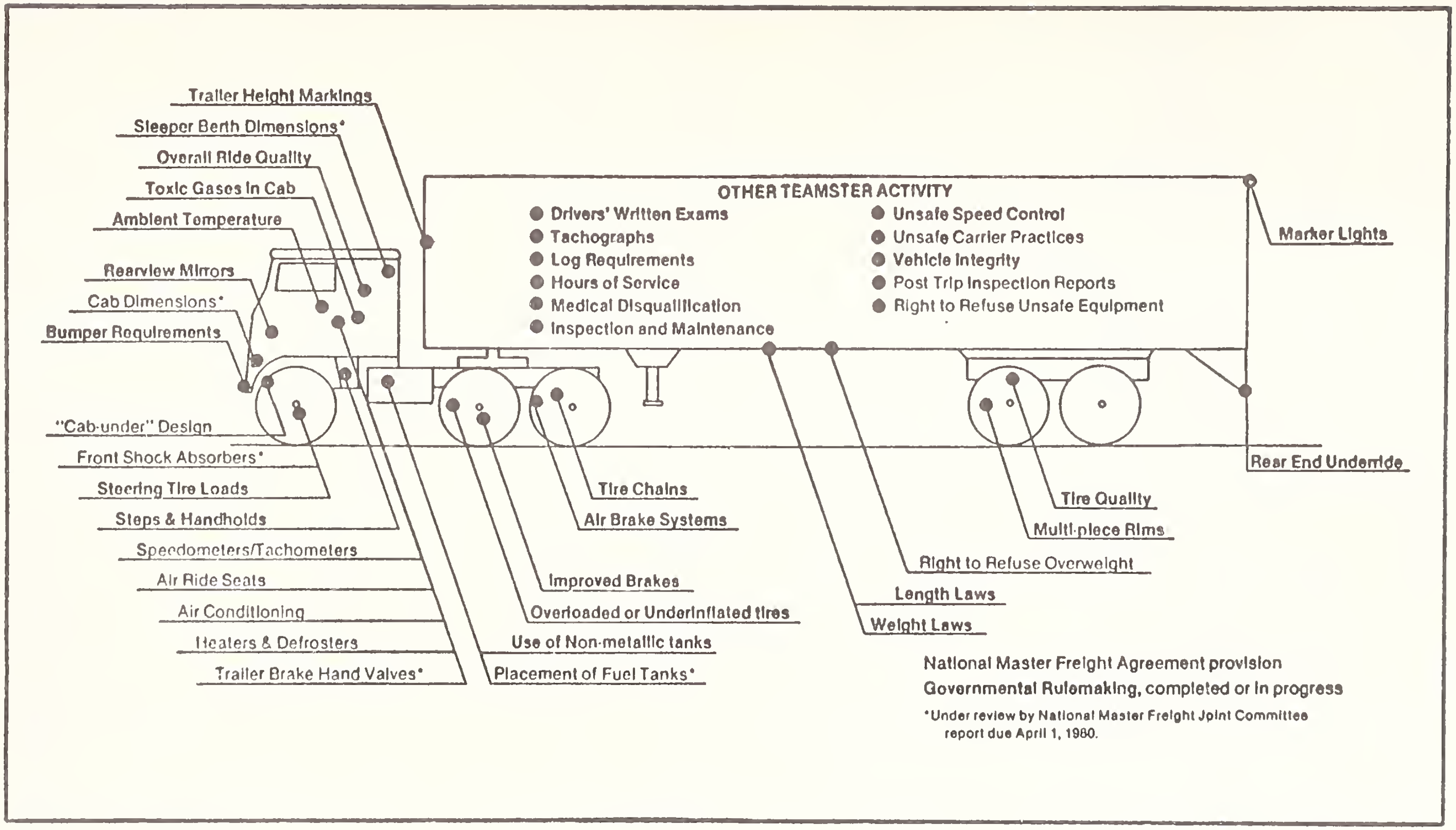
Accordingly, we enlisted the aid of the Highway Safety Research Institute of the University of Michigan, requesting an analysis of the safety and handling properties of the "cab-under" configuration. The resulting study[21] was the basis for contract negotiations for the latest National Master Freight Agreement. That contract now contains a clause reading: "No driver shall be required to drive a tractor designed with the cab under the trailer."[22]

While the Teamsters also petitioned the National Highway Traffic Safety Administration for a rule prohibiting operation of such a vehicle, we were able to adequately protect our membership from a hazardous vehicle without reliance on the government. Since Teamsters will not drive the "cab-under", its commercial application is extremely limited, and only a few units are now in operation.

The losses incurred through accidents resulting from vehicle defects, in terms of both human resources and property damage, are too great for society to ignore. This paper has sought to briefly explore the existing and future potential for a three-pronged attack on faulty and defective vehicles.

Through the cooperation and coordinated efforts of the Congress, the Department of Transportation, and Organized Labor, great strides have been made and will continue to be made toward the ultimate goal of safe highways.

APPENDIX "A"



This diagram represents a collection of the Safety and Health issues in the trucking industry which have received Teamster attention, governmental rulemaking or contractural negotiations.

FOOTNOTES

1. Congressional Record - Senate, June 21, 1979, p. S.8259.

2. National Highway Traffic Safety Administration, U. S. Department of Transportation, Fatal Accident Reporting System, 1978.

3. McDole, Thomas L., et al, Effect of Commercial Vehicle Systematic Preventive Maintenance on Specific Causes of Accidents. Ann Arbor, Michigan: The University of Michigan, Highway Safety Research Institute, July, 1975, p. 31.

4. Ibid., p. 6.

5. Bureau of Motor Carrier Safety, "Major and Special Emphasis Roadside Vehicle Inspections, August, 1978 - February, 1979," p. 1.

6. Bureau of Motor Carrier Safety, "Mississippi River Roadside Inspection, May 14-18, 1979," p. 4.

7. Ibid., p. 5.

8. Perchonok, Kenneth and Ranney, Thomas A., Analysis of Truck, Tractor/Trailer Accident Data, Buffalo, New York, Calspan Corp., June, 1976, p. 3.

9. Baker, Susan, et al, "Fatal Tractor-Trailer Crashes: Considerations in Setting Relevant Standards," Paper presented at the Fourth International Congress on Automotive Safety, July 14-17, 1975.

10. Perchonok, op. cit., p. 11.

11. McDole, op. cit., p. 11.

12. Durhame, R. V., letter to Dr. Robert Kaye, June 2, 1979.

13. 2 Fed. Reg. 18103.

14. 44 Fed. Reg. 38523.

15. 44 Fed. Reg. 50042.

16. 44 Fed. Reg. 76526.

17. United Parcel Service v. United States, 80-1268. (D.C. Cir., 1980).

18. Congressional Record - Senate, June 21, 1979, p. S.8259.

19. Senate Bill, S.1390, Section 11(b).

20. National Master Freight Agreement, Article 16, Section 1.

21. Bunch, Howard M., et al, "An Evaluation of the Safety and Handling Properties of the Cab-Under Truck Tractor Vehicle," Ann Arbor, Michigan: The University of Michigan, Highway Safety Research Institute, June, 1978.

22. National Master Freight Agreement, Article 16, Section 6(h).

STRESS SYSTEMS RELATED TO FRACTURE OF
DUCTILE AND BRITTLE MATERIALS

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In order to understand how various types of fractures are caused, one must understand the forces acting on the materials as well as the characteristics of the materials themselves. All fractures are caused by stresses, and the "weakest link" theory applies; that is, damage will occur when the stress on the weakest element exceeds its strength. Also, by understanding the ways in which single load, or monotonic, fractures are caused, one can then better understand fatigue fractures, which are the result of many thousands or millions of load applications.

To understand the forces, it is necessary to study the stress systems acting on the part. It is very useful to study the stress systems acting on a cylindrical member, such as a rod or shaft. A variety of stresses can be applied to a cylindrical member; also, the same principles can be applied to non-cylindrical parts. Shafts, and shaft-like parts, are very common parts and are widely used in construction of many assemblies and machines.

Stress systems are best studied by examining free-body diagrams which are simplified two-dimensional models of complex three-dimensional stress systems which are internal in the shaft. The figure shows the orientations of the normal stresses (tension and compression) and the shear stresses (sliding) which are 45° to the normal stresses. The free body diagrams of shafts in the pure types of loading--tension, torsion, and compression--are the simplest; they can then be related to more complex types of loading.

Tension Loading

When a shaft or similar shape is pulled in tension, it gets longer and narrower, just as does a rubber band when it is pulled. Similarly, the square in the free body diagram is elongated in the direction of the tensile stress and is contracted in the direction of the compressive stress, as is shown at the left in the sketch. Note that the shear stresses are at 45° angles to the axial tensile stress and the transverse compressive stress. Also, note that there are two sets of shear stresses, each perpendicular (90°) to the other, diagonally between the normal stress directions.

Ductile materials, by definition, are those that deform because the shear stress exceeds the shear strength before any other type of damage takes place. That is, the shear strength is the "weak link" in the system, and is the controlling factor. Therefore, under a tensile force, the internal crystal structure of the metal shears, or slips, microscopically on the millions of shear planes in the metal, resulting in lateral deformation, commonly called "necking down", prior to final fracture. This necking phenomenon occurs in the plastic regime, which means that the deformation is irreversible. Of course, plastic deformation is characteristic of any fracture of a ductile material. An interesting fact is that fractures of ductile metals in pure tension originate at or near the center of the shaft, toward the late stages of the fracture process. Many tiny internal cracks join by the process of "microvoid coalescence" to form a rough, jagged fracture surface. As the crack progresses outward, it eventually reaches a region near the surface that forms a 45° wall, or "shear lip", around the periphery of the fracture. This forms the familiar "cup and cone" fracture that is typical of tensile fractures of ductile materials. Incidentally, whenever a 45° shear lip is seen on any fracture, it locates the end of the fracture, or the last part to fracture.

Brittle materials, by definition, are those that fracture because the tensile stress exceeds the "cohesive" strength before any other type of damage can take place. Now, the cohesive strength is the "weak link" in the system, and is the controlling factor. Brittle materials always have a fracture direction that is perpendicular (90°) to the tensile stress, and have little or no deformation because fracture takes place before the metal can deform plastically, as in ductile materials. Thus, a tensile fracture of a brittle material has a direction that is essentially straight across the shaft. It has a bright, sparkling appearance when freshly fractured, characteristic of brittle materials. Since the magnitude of the controlling tensile stress is essentially uniform across the shaft, fracture can originate at any location in the cross section, in the absence of stress concentrations.

Torsion Loading

When a shaft is twisted in pure torsion, the same stress system--exemplified by the free body diagram--rotates 45° in one direction or the other, depending upon which way the shaft is twisted. When twisted as shown in the figure, the stress system rotates 45° counter clockwise. Note that the normal stresses (tension and compression components) are now at 45° to the shaft, while the shear stresses are now longitudinal and transverse. Each pair of stress components are mutually perpendicular to each other.

In a ductile material, the shear strength is again the "weak link" when the shear stress exceeds the shear strength. Again, plastic, or permanent, deformation occurs although it is not obvious, as is the necking down effect in tension. The shape of a cylindrical part, such as a shaft, is not changed by torsional deformation. An example will explain why:

Imagine that the shaft consists of an infinite number of infinitely thin disks. When the pack of disks is twisted, each disk slips a very small amount with respect to its neighbors. However, the diameter of each disk does not change, because of the slippage on the transverse shear plane. Eventually, fracture occurs on this transverse shear plane, which is essentially the sliding face between two adjacent disks. In pure torsion, the final rupture will be at the center of the shaft, although it will be offset toward one side if a bending stress is also present. Deformation also occurs on the longitudinal shear plane, but this does not normally cause fracture unless the material is weak in the longitudinal direction.

A brittle material in pure torsion will again fracture perpendicular to the tensile stress component, which is now at 45° to the axis of the shaft. This spiral type fracture is characteristic of brittle metals, including case hardened shafts, which are subjected to torsional loading. Other brittle materials, such as glass or chalk, will also fracture in a spiral manner if twisted.

The elastic stress distribution in pure torsion is maximum at the surface and zero at the center. Thus, fracture normally originates at the highest stressed region--the surface-- in pure torsion. This is true for all of the stress components: normal (tension and compression) as well as the shear stresses as is shown in the figure.

Compression Loading

When the shaft is loaded in axial compression (assuming a shaft that does not buckle) the stress component system again rotates 45° so that the compressive stress component is now axial, while the tensile stress component is transverse, as is shown in the figure. Now the shear stress components are again 45° to the shaft, as they were during tension loading.

A ductile material in compression does the reverse of what it did in tension: it becomes shorter and thicker under the action of the slippage on the shear planes. In short, it bulges when squeezed by the compressive force. This is characteristic of metals being cold headed, pancake forgings, and other ductile materials under axial compression.

A brittle material in pure compression will, as always, fracture perpendicular to the maximum tensile stress component. Since the tensile stress component is now transverse, the brittle fracture direction is now longitudinal, or parallel to the shaft. Brittle materials, such as extremely hard metals, glass, chalk, and rock will split or shatter longitudinally when loaded in compression. Indeed, this is the principle of rock crushing.

The elastic stress distribution in pure compression again is the reverse of that in tension: uniform across the section (assuming no stress concentration), except in the compression direction, as shown in the figure.

Bending Loading

When a part is loaded in bending, the convex surface has a free body diagram stress system similar to that shown here for tension. Conversely, the concave surface will be stressed in compression, and will have a stress system as shown for compression. Approximately midway between the two surfaces will be the neutral axis where all stresses are zero, while the surface stresses are at a maximum. Thus, failure can be expected to originate on the convex side of the bend where the tensile stress exists.

Fatigue

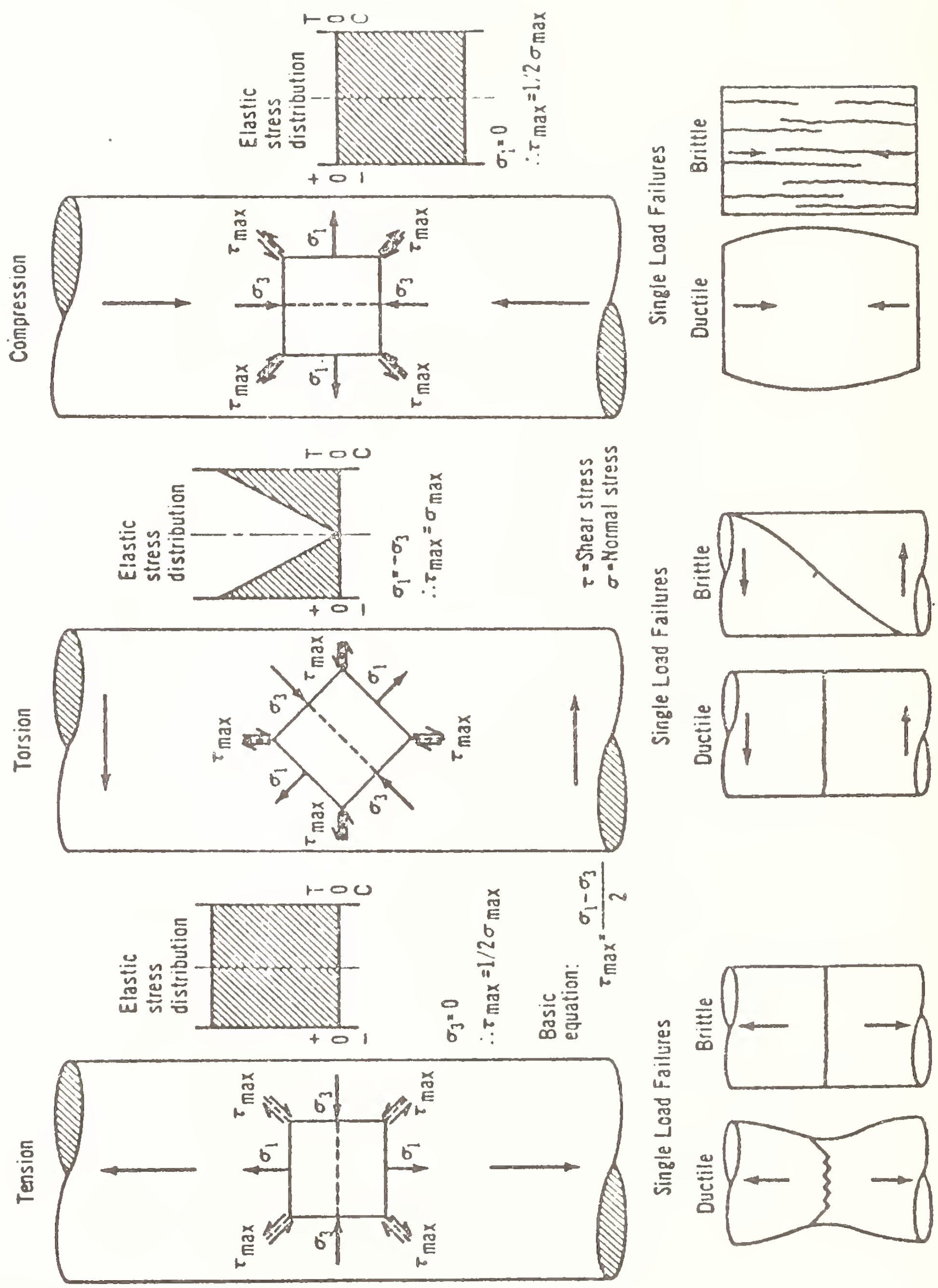
The preceding discussion was concerned with single load deformation and fracture. Fatigue is unique in that the magnitude of the multiple load applications are usually not high enough to cause plastic deformation. That is, the stresses are relatively low. However, upon many, many repetitions microscopic changes take place in the crystal structure that leads to formation of fatigue cracks.

The essential thing to remember is that the propagation, or growth, of fatigue cracks over a large period of time, is in exactly the same direction as the rapid crack growth of brittle materials under the same type of loading. That is because fatigue cracks propagate in a direction that is perpendicular to the principal tensile stress--the brittle fracture direction.

Thus it can be seen that fatigue cracks will propagate essentially straight across a shaft or other surface stressed in tension, such as the convex side of a shaft or other part in bending. Similarly, a shaft under repetitive torsional loads will form fatigue cracks 45° to the axis of the shaft. Reversed torsion--twisted alternately in both directions--will form two sets of fatigue cracks, each at approximately 45° to the shaft, or approximately 90° to each other. This is the action that forms the classic "starry" fatigue fractures of a splined shaft in which many cracks, each at 45° to the axis, occur during reversed torsional fatigue. If axial compressive forces cause fatigue cracks, they will be parallel to the axis of the shaft; however, this type of loading is rare.

Summary

By keeping in mind the principles outlined here, the analysis of single load and fatigue fractures can be better accomplished. The principles are always the same; however, confusion can be caused by misinterpretation and uncertainty as to the type of loading, particularly under combined loading conditions.



Tension
Torsion
Compression
Elastic stress distribution
+ 0 −
T 0 C
$\sigma_3 = 0$
$\therefore \tau_{max} = 1/2\,\sigma_{max}$
Basic equation: $\tau_{max} = \frac{\sigma_1 - \sigma_3}{2}$
$\sigma_1 = -\sigma_3$
$\therefore \tau_{max} = \sigma_{max}$
τ = Shear stress
σ = Normal stress
$\sigma_1 = 0$
$\therefore \tau_{max} = 1/2\,\sigma_{max}$
Single Load Failures
Ductile
Brittle

PERSPECTIVES ON DIAGNOSTIC SYSTEMS FOR THE TRUCKING INDUSTRY

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The paper is based on a survey which was conducted among the members of the American Trucking Associations Maintenance Council. The primary purpose of the survey was to obtain an indication of how state of the art technology could be best applied to resolving the trucking industry's maintenance problems. Additionally, the survey was designed to assess the industry's receptiveness to new and sophisticated maintenance aids. This paper, therefore, presents a summary of the results of the survey and the conclusions we have drawn. The key point of the paper is that the maintenance of several vehicle systems such as electrical including the charging and starting system, engines, and brakes constitute a major cost factor to the industry. This cost can most likely be reduced significantly by the use of state of the art diagnostic systems. While the trucking industry appears to need this capability, it will be up to the manufacturers to develop the concepts and demonstrate their value.

Key words: Automated test equipment; diagnostics; technology in truck maintenance; truck maintenance aids.

As many of you may be aware, the trucking industry is rather conservative when it comes to technological innovation. We believe, as do managers of other successful businesses, that the first objective of a company is to make a profit and since we operate on a very narrow profit margin there is a propensity to invest only in the tried and proven. There is very little individual research and development work related to vehicles sponsored by motor carriers. In fact, only a few trucking companies are large enough to have engineers assigned solely to such R & D functions. This then means that it is not only incumbent upon the manufacturers who supply us with our equipment to conceive and develop new concepts, but they must also demonstrate the concepts' value to us. In the area of diagnostics this is a major challenge. When it comes to offering the industry sophisticated computer controlled systems which instantaneously provide vehicle systems analyses, tell mechanics what to do to effect repairs, issue the parts, and generate the required record, I have found a level of skepticism among most truckers that only my education in the "Show me" state could have prepared me for. This situation does not, however, diminish our interest in finding better and more efficient ways to operate, but it does significantly affect our rate of progress and the manner in which we proceed.

To improve our maintenance capability, the industry has established working groups such as the ATA Maintenance Council and the Technical Advisory Group to affect coordination with government and the manufacturer; to aid in identifying our needs or, stated another way, our problems; and to continue this coordination in the development and testing of the solution. This process obviously leads to evolutionary, rather than revolutionary, changes in the industry. The development of diagnostic systems for commercial trucks has and, no doubt, will continue to adhere to this principle.

Our interest in the subject of diagnostics is not new. It arose during the early seventies when it became clear that motor vehicles would become much more complex as governmental rule-making began to dictate engineering designs. Antilock braking systems and emission controls confirmed this contention. To help deal with this situation, the position I currently serve in was created and a Task Force of representatives from trucking and the manufacturers was formed. The purpose of this Task Force was to look into the areas of diagnostics in general and antilock brakes in specific. While the antilock issue has been settled in the courts, there remains a nucleus of capable, forward thinking people who have been exposed to the state of the art in technology for diagnostics. These people have formed a Diagnostic Task Force for the purpose of continuing their efforts to determine the applicability of this technology to the trucking industry and to promote its acceptance where appropriate.

Following the premise that the onus for new product development is on the manufacturers, the Diagnostic Task Force has centered its efforts on <u>first</u> trying to determine where the industry can best benefit from diagnostics, <u>secondly</u>, on attempting to assess the industry's receptiveness of new and more sophisticated maintenance aids, and <u>thirdly</u>, to promote the awareness of a broad spectrum of the industry's maintenance managers and executives to what the state of the art in technology has to offer.

To accomplish our first two goals, we recently conducted a survey among the trucking firms which participate in the Maintenance Council. The survey questionnaire was quite comprehensive, consisting of sixty-three questions, but was not designed to derive exact analytical results. Realizing that fleet executives have very limited time for such matters, we asked that they provide the best data that was readily available and that where time consuming searches and computations were required, an estimate would suffice. Therefore, I must characterize the results of this survey as ballpark indications.

With this inherent limitation, we sent out approximately 410 surveys to which we received 50 responses. I was somewhat disappointed initially with the limited response, but after some discussions with others who have conducted surveys within the industry I found that a 12% response is considered good.

Again, I must stress that this low response is not a reflection of a lack of interest in improved maintenance capabilities within the industry, but rather a combination of factors. First is the strenuous workload of the maintenance managers who must give priority to their daily operations. Second, as I eluded to earlier, it must be recognized that many of our present maintenance directors started as helpers or mechanics and while they know their trucks to the nth degree, they may not have the scientific training which would allow them to understand and appreciate the benefits of complex advance technology systems. Last and certainly not the least important is the detailed twelve page survey which might tend to discourage all but the most zealous from responding.

Although there may be some bias, since prople most interested in the subject are most likely to respond, we do feel that the results are reasonably representative of the industry. At any rate, it's the best information we have. The size of the firms varied from one as small as twenty-five vehicles to a firm owning 85,000 vehicles. A total of approximately 140,000 vehicles are represented. I should note that for the purposes of the survey a vehicle could be a pickup & delivery vehicle, a straight truck, a tractor, or a trailer. The firms also varied in type of business and the manner in which the vehicles are utilized. Approximately 1/3 of the firms were common carriers, 1/3 were private carriers, and the remaining 1/3 consisted mostly of leasing companies and specialized haulers.

In structuring the survey we tried to anticipate the more profitable ways of using diagnostics. Among these were:

1. Identifying pending failures in the incipient stage so that repairs could be effected before costly damage occurred.

2. Simplifying maintenance by readily identifying the problem and thus reducing manhours expended.

3. Reducing lost revenues and operational delays, by precluding enroute breakdowns.

4. Enabling the trucking firms to meet ultimate EPA in-service standards by maintaining their vehicles within prescribed parametric limits.

To assess the correctness of these conjectures, we designed our questions to acquire an indication of the industry's maintenance and operating philosophy, identify their maintenance problems, and assess their preference for diagnostic systems.

In examining the industry's maintenance and operating philosophy we found that on the average all vehicles, with the exception of trailers, are retained for approximately five years. This equals about 160,000 miles on P & D vehicles, 190,000 miles on straight trucks, and 550,000 miles on tractors. Trailers are generally kept for 10 years and several firms retain them throughout their useful life.

This rate of turnover is a significant indicator of a fleet's need for an extensive maintenance investment and also indicates how quickly a firm might acquire newer vehicles which may be more readily amenable to the use of automated diagnostics. Let us take a look at tractors as an example. As I just mentioned, the average trade-in point for the original owners of tractors is 550,000 miles. The average engine life before a major overhaul is required is approximately 300,000 miles. Transmissions last for about 350,000 miles and differentials and axles require overhaul or replacement at about 315,000 miles. As you can surmise, the vehicle is essentially rebuilt at about the 300,000 mile point and traded before the second major overhaul is imminent.

With surging increases in the cost of money, the declining availability of capital, and the slowing of the economy, this vehicle turnover pattern could change. It may prove to be wiser or possibly necessary for many carriers to accomplish that second round of vehicle rebuild. This then would extend the average original ownership of a vehicle to approximately 900,000 miles or about nine years. An alternative approach would be to try to extend the overhaul interval through greater emphasis on preventive maintenance and the use of more sophisticated prognostics and diagnostics. Additionally, I would anticipate, and we are seeing to some degree, a more intense emphasis on "spec'ing" for greater overall reliability.

Actually, all carriers have a scheduled Preventive Maintenance Program. However, the schedules appear to be as diverse as there are trucking firms. According to our survey each tractor receives a PM inspection on the average every 11,400 miles. This compares well with a report by Burlington Fleet Services, a data processing service for trucking firms, which states that linehaul tractor PMs are conducted on an average of every 12,111 miles. Further, it should be noted that most fleets use a graduated inspection program where through a cycle of two to three inspections, the extent of the inspection increases as a function of the mileage interval increase. For example, some companies have A, B, and C inspections which are conducted every 4000, 12000, and 24000 miles respectively. The details of these inspections vary from 190 items on an A inspection to 254 on a C.

While each fleet may have its own distinct schedule and inspection procedure, one point is certain--that all PM checks are time consuming and their productivity is limited by the ability of the mechanic and capability of his test equipment. For this reason, we believe that the development of a system to be used to accomplish quick, revealing PM checks would be one of the most cost effective applications of diagnostic systems. In fact, a prototype of such a system for diesel engines has been developed by Hamilton Test Systems and is presently undergoing in-service testing with the New York Transit Authority. In this test program the bus engines are tested each day as the bus proceeds through the checklane. A computer printout is produced which lists the measured engine operating parameters and flags any that exceeds predetermined limits. This type of system inherently provides the capability to track the history of each parameter. Such a running historic record in turn provides the maintenance manager with the information necessary to predict failure and schedule maintenance to accomodate his operational schedule yet preclude catastrophic failures.

The possible cost benefits of such a system is indicated by the fact that each tractor, as an example, is out of service for maintenance 14 days a year. This is about a 5% out of service rate which must be reflected in lost profits. Since profits from each tractor average about $170 a day, when multiplied by 14 that comes to a $2380 loss annually. While this out of service rate could never be reduced to zero, a reduction of only two or three days could result in significant cost savings, especially for the large fleets. Even for a company with only 100 tractors, a two day reduction in the out of service rate could result in a $34,000 a year increase in profits.

Another area where we expect a payoff in the use of advanced diagnostics is in mechanic productivity. From our survey we found that when a problem occurred, the mechanics spend about 18% of their time locating the cause of the problem. Further, approximately 7.5% of the vehicles which have had maintenance performed return within 30 days for repair of the same or a related problem. The ultimate diagnostics systems which would be acceptable to the industry should allow the mechanic to use a quick and simple procedure to identify and isolate the cause or causes of a problem and make a lasting repair. As you will see when we discuss the preference portion of our survey, one of the maintenance managers' primary goals is to be able to "fix it right the first time."

This leads us to ask about the capability of the mechanics and in particular to attempt to assess their adaptability to new techniques. Presently, most mechanics perform all general maintenance functions. Only a few firms have specialists. Our survey indicates that most firms perform all their maintenance including major overhauls. The exceptions are primarily body and framework and precision machining. The average experience level of the mechanics is 10.65 years, although this varies from as low as two years with one company to as high as twenty-five years with another. About 55.6% of the mechanics have received factory or technical training and 80% of the companies provide periodic training for their mechanics. Although most companies have only relatively rudimentary test equipment, several companies have invested in more elaborate test gear, especially test equipment for the electrical starting and charging system. In fact, nine of the 30 responders own dynometers. Every company reported that they have not experienced any bias on the part of mechanics against new test gear. Actually, most mechanics are both willing and able to use the equipment.

These findings lead us to believe that the commercial trucking industry will readily adapt to more modern test equipment and techniques. In reaching this conclusion, we are not disregarding the difficulty some automotive manufacturers have experienced in trying to promote new approaches such as plug-in diagnostics. The fact is that the motivation of the auto service mechanic and his shop manager for maximum efficiency is probably not as great as it is in a company shop which maintains its own vehicles. Also, through "spec'ing," most fleets keep a reasonably standardized fleet. Thus, the fleet mechanics are not required to work on as large a variety of vehicles.

While attempting to identify the major maintenance problems, we found that by far the most frequently identified problems were electrical. Among these, the most prevalent problem was burned out light bulbs. Now, a burned out bulb is easy to identify and doesn't require any special diagnostic system. However, it can be difficult to replace, create a safety hazard, and result in enroute delays and fines. Since this does not require a diagnostic remedy, why even mention it? The reason is that it points out a consideration which we must keep in mind and one that I believe is the key to this conference. That is, that improvement in the reliability of an item which frequently fails may very well be more cost effective than providing a better way to maintain it. In the case of lights, we believe that with the reliability achieved in today's sealbeam headlamps that the technology must be available to provide long lasting lamps for the other lighting needs on commercial vehicles.

The other major electrical problems involve alternators, batteries, starters, wiring and connectors. Development of a quick and simple capability for a pretrip check of the starting and charging system would most likely result in a significant savings by reducing highway breakdowns. Better than half of our road service calls for all types of commercial vehicles are in response to electrical system problems. The average tractor requires 2.7 road service calls per year with an approximate cost of $90 per response. This equals $243 per year per tractor. Considering that in 1977 there were 1.2 million tractors registered in the U.S., it is clear that reducing electrical system problems will have high potential for cost benefits.

The second major problem area identified was the engine. The major causes of engine failures were worn rings, bushings and injectors. Certainly all metal parts are destined to wear and failure, but effective diagnosis of impending premature failures with immediate corrective maintenance will very likely cause an increase in the average life of these engines and the possible concomitant extension of the use of the vehicle. We are all familiar with the old cliche "But for a nail, etc." well, in the real world of trucking an appropriate expression could be "But for a $10 part a $[illegible]000 engine was lost." The maintenance managers are concerned with this problem. For this reason approximately 75% of those responding to our survey are using oil sampling analysis to attempt to anticipate failures.

The third most significant area of complaint was the brake systems. Brake system maintenance is needed on the average, every 164,000 miles. This requirement is attributed primarily to air line failure and lining wear. Here, as with the light bulbs, the tendency is to conclude that diagnostics may be of little benefit and emphasis should be placed on designed-in reliability. However, there may be some imaginative approach such as a plug-in system for checking valves, timing, and balance which might contribute to lining life. Here again it will be up to the manufacturer to show us that there is a better way--that we can move away from the hammer and screw driver type of inspections.

While survey results indicate that manual transmissions for diesel tractors are lasting 350,000 miles, we are not convinced that this is a true representation of the life of these components. Some of our fleets have been experiencing an unusually short operational life with some of the new transmissions which are designed to mate with the high torque rise engines. Many of the problems we have seen are typical new product failures which we believe will soon be eliminated. Thus it is possible that the average life of transmissions which have been debugged are a lot longer. As an indication of this, two of our responders reported an average transmission life of 700,000 miles. Ideally, we would want to see drive train component last approximately 600,000 miles, which exceeds the average trade-in point. I am not aware of any diagnostic test equipment that has been developed for the drive train. Here again, the emphasis might best be placed on reliability rather than repair.

One additional area of concern is tires. Although we did not gather sufficient data relative to tire failures to draw any conclusions, the trucking industry is interested in the benefits derived from properly inflated tires such as fuel savings, reduced tire wear, as well as safety. For this reason we support the efforts of NHTSA and the manufacturers to develop tire pressure monitors. We ask only that before these devices are mandated that their reliability be adequately proven. To aid in this process, the ATA Maintenance Council has been fleet testing the commercially available devices and will continue these tests into the foreseeable future. The results of the tests are available to NHTSA. It is in our best interest as well as the government rulemakers that we not repeat the billion dollar mistake that we encountered with anti-lock brakes.

We concluded our survey with a series of questions designed to determine the industry's preference for diagnostics.

From these questions we found that the most desired application of diagnostics is to detect problems in the charging and starting system. The second most preferred application is for the engine, followed by one for the brakes.

When asked to select their primary reason for purchasing diagnostic equipment, the answer selected by a sizable majority was: to reduce unnecessary repairs and parts replacements. The other choice in the order they were rated were:

o Improve mechanic productivity and reduce training.

o Reduce maintenance costs by reducing road calls.

- Improve fuel economy.

- Comply with Clean Air Standards.

The relative emphasis given to Clean Air Standards simply reflects the fact that the maintenance manager must view his more immediate problems with greater seriousness. At present except for some rudimentary smoke opacity limits, there are no Clean Air Standards for in-service diesel trucks. If emission standards do ever come into being it will most likely be more feasible and reasonable to enforce them by placing acceptable limits on the performance parameters of the engine. Monitoring these parameters will not only alert the vehicle owner when a limit is exceeded but should allow immediate diagnosis of the problem. We believe that this approach will more satisfactorily prevent air pollution from heavy duty vehicles as well as aid the fleet operator in maintaining his vehicles and in gaining maximum utilization from his fleet.

When asked whether they would want an on-board system, off-board system, or both, only one responder selected on-board. The remainder split evenly between off-board and both.

From some of the comments received both in the survey and in our discussions with members of the industry, we believe that the apparent bias against on-board electronics is a result of the antilock episode. Due to this stigma, any on-board application of electronic devices will require explicit proof of reliability. In this regard, we are working through the SAE to develop an Electical Equipment Environmental Recommended Practice for Trucks similar to the J1211 standard for automobiles. I believe as more electronic sensors and controls are added to commercial vehicles that use of a certain level of on-board sensing as well as diagnostics will be inevitable.

I can foresee a number of advantages to having on-board sensors. Not the least of these would be the time savings in test system hook-up and the possibility for instantaneous readout. A prototype 32 parameter system has been developed by Rockwell and has received extensive testing in the government and industry sponsored heavy-duty vehicle fuel economy test program. To the best of my knowledge there are no plans at this time to market this system, but a much simpler system is on the market. Again, I believe that proven reliability and cost will be the driving factors in gaining industry acceptance of such a system.

In assessing how the maintenance managers preferred to use the equipment, we found that three-fourths of those responding felt that the equipment should remain in the shop rather than be portable. They also preferred that the equipment be in a single unit, but be movable within the shop. Interestingly, only five wanted to use diagnostic systems in the check lane. I relate this lack of interest in a more innovative approach to my experience in the Air Force. As history has shown the horse soldiers were not overly enamoured with the tank, nor did the leaders of the Army and Navy rush forth to embrace Gen. Mitchell's concept of Air Power. Well, this sort of perception of change is still with us. Today's fighter pilot is not overly receptive to long range standoff weapons launched from big aircraft. The point is, as Alvin Toffler detailed in his book, Future Shock, that most people are more content with the status quo. The introduction of sophisticated diagnostic equipment will change the mode of operation of most companies but it will probably be for people not intimately associated with the daily operation of a fleet maintenance facility to develop and show the advantage of these changes. The New York Transit Check Lane Test Program which I previously mentioned is a good example.

The growth of the use of computers and data processing in the trucking industry will help facilitate this change. Although computers are extensively used in the accounting side of the trucking industry, their use is still budding within the maintenance area. Approximately half of the responders have computer-based maintenance data systems, but of those who do not, more than half are converting or plan to convert to computers in the near future. The remainder felt that their company was too small to make the use of a computer economical.

Not too astonishing then was the response of more than half of the companies that they wanted a diagnostics system which would feed information directly into a central maintenance data system.

Finally, the one area that surprised me the most was the response to the question concerning how much they would be willing to pay for a diagnostic system. They could choose between less than $5000, $5000 to $10000, $10000 to $20000, and over $20000. Most of the responders felt the system should cost between $5000 and $10000. However, two were willing to pay in excess of $20000. Of course the value of these figures is no better than the commitment of the responder. But it does indicate that contrary to my previous impression, maintenance managers are willing to invest in expensive equipment if the cost effectiveness is there.

I believe the industry is on the threshold of making dramatic changes in their maintenance concept. With the expanded use of diagnostics by the engine producers and the development of electronic feedback systems for pollution control and fuel economy, the field use of diagnostics will be a natural follow-on. The interest is there. The challenge is for the manufacturers to plan the development of these maintenance aids, now, as the changes in vehicle design are evolving so that they can provide us with systems that meet our needs when those needs arise.

INVESTIGATION OF DEFECTS IN MOTOR CARRIER ACTIVITIES

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Congress recognized in writing the National Traffic and Motor Vehicle Safety Act of 1966 that safety standards could not possibly cover everything that could be wrong with an automobile. It therefore created the Office of Defects Investigation, whose task it is to discover and investigate potential safety-related defects in motor vehicles and motor vehicle equipment.

The NHTSA receives some 5,000-6,000 owner complaints every month. In previous years, most of these complaints were in letter form, but as our nationwide toll-free Hotline becomes better known, the trend is shifting from letters to phone calls. The "Auto Safety Hotline" is accessible, on a toll-free basis, to all vehicle owners and to those desiring information (or wishing to comment) on vehicle safety defects. This toll-free number (800-424-9393) can be used by callers from the 48 contiguous States (Alaska, Hawaii, Puerto Rico, Guam and the Canal Zone cannot use this number).

The purpose of the "Auto Safety Hotline" is to solicit accurate and timely information from the public on vehicle safety problems and to help us gather potential safety defect data. Reports from the Auto Safety Hotline are used as an early warning to identify problems and trends in specific vehicles and items of equipment and to assist in defect investigations and recall actions. Copies of this data are also furnished to the appropriate manufacturers.

In addition to the Hotline we have some 2,500 repair shops across the Nation involved in a Parts Return Program where suspect parts are furnished for our engineering analysis to identify the existence of safety-related manufacturing and design defects. In the 7 years this program has been in operation, we have received some 7,000 suspect parts.

All such inputs are coded for computer storage when received. The type of information stored includes owner name and address, vehicle make, model, year, VIN, components involved, fault, and whether an accident or injury resulted. The letter is then microfilmed, primarily so that we have a permanent copy and secondarily for logistical purposes. At the present time, NHTSA has over 500,000 inputs in our computerized complaint file. The physical storage and retrieval problems of this many documents necessitates our microfilming and use of computer systems.

An initial screening of all incoming inputs separates out those with significant potential safety problems. These are then further screened and analyzed by the Engineering Analysis Division to define the defect and determine if any are candidates for a formal investigation. This analysis is performed by an engineer assigned to that particular item, and usually includes contacting the vehicle owner, checking our computer file for other similar complaints, determining if a service bulletin or previous recall campaign applies to the subject problem, contacting the manufacturer, and in some cases testing as necessary.

Once the defect has been defined and it appears to be significant, it is discussed at a defects review panel meeting. The group, which consists of engineers, legal counsel, and consumer services, decides if a formal investigation is warranted. If so, the matter is turned over to the Defects Evaluation Division for processing. At the time an investigation is opened, the manufacturer is immediately notified by telephone, and a press release is often prepared which outlines the basis of the case and describes the alleged defect.

The Defects Evaluation Division is charged with the management and completion of the formal case investigation. The case investigator builds the case, typically through owner interviews, information requests to the manufacturer, R. L. Polk surveys, additional testing by our Ohio Test Facility or other contractors and any additional information received from the public from consumer advisories and press releases. It is not unusual for the number of consumer complaints on a particular item to increase a hundred-fold after the initiation of a formal investigation is announced in the media.

If an initial determination is made, the manufacturer and other interested parties, whether consumer groups or private individuals, are given the opportunity to present their views in a well publicized public meeting. After the meeting, our Administrator makes a final determination based upon the investigation file and the comments from the public meeting. If the Administrator's decision is that a safety-related defect does exist, the manufacturer is ordered to notify vehicle owners and provide remedy in accordance with the Act. If a manufacturer disagrees with the determination, he can resort to the judicial process for relief.

EFFECTIVENESS OF NHTSA'S DEFECTS PROGRAM

IN THE MOTOR TRUCK AREA

In order for NHTSA to effectively exercise its authority in the area of safety-related defect recalls, the cooperation of the vehicle driver, vehicle owner or fleet, the Bureau of Motor Carrier Safety, Teamsters, and the vehicle and component manufacturers is needed. This cooperation involves communication at six levels:

1. Vehicle drivers, if they do not own the vehicle, should communicate problems they are experiencing on the road to the vehicle owner, the vehicle owner's maintenance personnel, or possibly to the NHTSA.

2. Vehicle owners or fleets should report suspect defects to the NHTSA or the vehicle or equipment manufacturer.

3. BMCS field inspectors should report thru channels, defects detected.

4. Vehicle and equipment manufacturers must report any known safety-related defect to the NHTSA.

5. Teamsters safety personnel should report suspect defects to the NHTSA.

6. Vehicle and equipment manufacturers must correct any safety-related defects in a timely manner after a recall campaign has been established.

APPENDIX

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31st Meeting

Failure Prevention
in
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2. Proceedings of Meeting Nos. 1-9 (set of 5) . . . . AD 721 359

Meeting Nos. 1-5	Papers and discussion on failure analysis and control
Meeting No. 6	"Detection, Diagnosis and Prognosis" December 6, 1968
Meeting No. 7	"Failure Mechanisms as Identified with Helicopter Transmissions" March 27, 1969
Meeting No. 8	"Critical Failure Problem Areas in the Aircraft Gas Turbine Engine" June 25-26, 1969
Meeting No. 9	"Potential for Reduction of Mechanical Failure through Design Methodology" November 5-6, 1969

3. Proceedings of Meeting No. 10 . . . . AD 721 912
   "Vibration Analysis Systems"
   January 21-22, 1970

4. Proceedings of Meeting No. 11 . . . . AD 724 475
   "Failure Mechanisms: Fatigue"
   April 7-8, 1970

5. Proceedings of Meeting No. 12 . . . . AD 721 913
   "Identification and Prevention of Mechanical Failures in Internal Combustion Engines"
   July 8-9, 1970

6. Proceedings of Meeting No. 13 . . . . AD 724 637
"Standards as a Design Tool in Surface Specification for Mechanical Components and Structures"
October 19-20, 1970

7. Proceedings of Meeting No. 14 . . . . AD 721 355
"Advances in Decision-Making Processes in Detection, Diagnosis and Prognosis"
January 25-26, 1971

8. Proceedings of Meeting No. 15 . . . . AD 725 200
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14. Proceedings of Meeting No. 21 (NBS SP 433) . . . SN003-003-01639-0
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Microfiche Copy: $2.25

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17. Proceedings of Meeting No. 24 (NBS SP 468) . . . SN003-003-01760-4
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18. Proceedings of Meeting No. 25 (NBS SP 487) . . . SN003-003-01829-5
"Engineering Design"
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19. Proceedings of Meeting No. 26 (NBS SP 494) . . . SN003-003-01844-9
"Detection, Diagnosis and Prognosis"
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20. Proceedings of Meeting No. 27 (NBS SP 514) . . . SN003-003-01935-6
"Product Durability and Life"
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Cost, Printed Copy: $3.75

21. Proceedings of Meeting No. 28 (NBS SP 547) . . . SN003-003-02083-4
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22. Proceedings of Meeting No. 29 (NBS SP 563) . . . SN003-003-02120-2
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10. SUPPLEMENTARY NOTES

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☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT (A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)

These proceedings consist of 18 submitted entries (16 papers and 2 abstracts) from the 31st meeting of the Mechanical Failures Prevention Group which was held at the National Bureau of Standards, Gaithersburg, Maryland, April 22-24, 1980. The theme of the symposium was failure prevention in ground transportation systems. Areas of interest included rail vehicles and structures, highway and road bridges, pipeline transportation systems, and motor carriers.

12. KEY WORDS (Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)

bridges; diagnostic systems; failure; failure detection systems; fracture; fracture control; ground transportation; motor carriers; pipelines; rail structures; rail vehicles; reliability; transportation systems.

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