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Calspan

RAIL HAZARDOUS MATERIAL TANK CAR DESIGN STUDY

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FOREWORD

This report was prepared by Calspan Corporation, Buffalo, New York, 14221 to cover work accomplished from November 1972 through April 1975 under Contract No. DOT-FR-20069. This program has been administered by Mr. Donald Levine and Mr. David Dancer of the Federal Railroad Administration, and Mr. William Hathaway of the Transportation Systems Center, Department of Transportation.

ABSTRACT

Tank cars containing flammable liquefied compressed gas ladings have been involved in many accidents involving death, injury, and large amounts of property damage. The losses have generally occurred following puncture of a tank in a derailment or from rough handling in terminal operations. Subsequent ignition of escaping lading frequently has resulted in the development of a general conflagration, often accompanied by the violent rupture and rocketing of other fire-exposed pressure tank cars. This report provides the basis for defining practical and economical safety improvements and identifies the safety research gaps which must be closed before a prototype tank car can be designed to optimal safety/economic considerations. Increased safety can be realized by decreasing the probability of an initial puncture, and/or decreasing the probability of subsequent tank ruptures of other cars from fire exposure. Six areas were given particular consideration because of their greater potential for success for 112A/114A series tank cars. These were: 1) operational changes, 2) head shields, 3) modified couplers, 4) thermal insulation, 5) tank material changes, and 6) safety relief system modifications.

Head shields and modified couplers were found likely to be "cost-beneficial." The expense/loss reduction break-even points as a function of coating life were determined for thermal shield materials. A safety relief valve, which is actuated by excessive lading temperature and then remains open until reduction to ambient pressure, was found to have high potential for being cost beneficial.

Other tank car research is reviewed and an improved thermal model for calculating the effects on a tank car exposed to fire is presented. It was found that full scale fire test results confirmed earlier analyses indicating substantial underestimation in existing design calculations of potential heat loadings to fire-exposed cars.

In general, economic impact of mandated changes in over-the-road and terminal operating procedures may be expected to be greater than that of a car design change to achieve the same safety objective. Neither change may be "cost-beneficial" when this term is interpreted to mean zero negative impact on car profitability or commodity transport cost. On the other hand, it was found that a modification increasing the cost of a tank car by 10 percent tends to increase the delivered cost of LPG, for example, only on the order of 0.5 percent. Therefore, substantial expenditures could be made for car improvements with relatively little -- but not zero -- overall economic impact.

Substantial improvements in car safety appear to be feasible without resort to the use of exotic materials or fabrication techniques which could not be accommodated by existing tank car manufacturing facilities.

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SUMMARY AND CONCLUSIONS

The broad objectives of this research effort were to:

1. Provide the basis for defining practical and economical safety improvements which can be either retrofitted to in-service tank cars or incorporated into the design and manufacture of new tank cars, and

2. Define the safety research gaps which must be remedied before a prototype tank car can be designed to optimal safety/economic considerations.

The greatest accident losses involving tank cars have occurred for class 112A/114A cars containing compressed liquefied gas lading. The losses have generally occurred following a puncture in a derailment or other abnormal operation. The lading flowing out of the puncture may be subsequently ignited and the resulting fire causes damage in the surrounding area. If the ignition occurs after considerable amount of lading has been dumped and vaporized, the damage area can be extensive. Often the fire heats one or more tank cars that have remained intact during the initial accident. After a time interval, the heated tanks rupture, greatly increasing the severity and area of damage.

This report examines possible methods of increasing safety of tank cars. Increasing the safety of tank cars can be accomplished by two means: decreasing the probability of puncture or decreasing the probability of subsequent tank ruptures of other cars after a puncture and fire. Six areas within these two broad categories were given particular consideration because of their greater potential for success. These are: 1) operational changes; 2) head shields; 3) modified couplers; 4) thermal insulation; 5) tank material changes; and 6) safety relief valve modifications.

Operational Changes

Relative to operational changes, it was concluded that:

Reductions in authorized speeds for trains carrying hazardous commodities by tank car would, in the overall picture, reduce the frequency of severe accidents.

Cost effectiveness of reduction in speed is highly variable with respect to route. Many cases would not show adoption to be cost beneficial. Individual consideration of cases would be required.

Adoption of a general rule requiring the use of "buffer cars" between hazardous material laden tank cars is not indicated.

Operational restrictions affecting standard practice in terminal operations will tend to be an expensive option when compared with car design changes to obtain the same level of safety.

Head Shields

Head shields are shields located at the ends of tank cars to reduce the probability of head puncture. Cost benefit analyses were developed for head shields and the other modifications considered in this report. The cost benefit analyses are based on published accident loss data, estimated effectiveness of the modification in reducing accident losses, estimated cost of the modification, expected life of the modification, and expected capital return. Head shields were found to be cost beneficial. In this and subsequent cost/benefit analyses, it was found to be more realistic to increase published loss data by 25 percent based on a thorough re-evaluation of five accidents chosen to be representative of a range of dollar losses per accident. The analysis shows the economic benefit of head shields to be $577 for new cars and $375 for existing cars at a cost of capital assumption of 10 percent. (The economic benefit is the amount that could be spent on a modification and be paid back, including interest, from the reduction in accident losses minus the cost of the modification.) If there were no capital recovery allowance, the reduction in losses minus the cost of the shield would be $2408 for new cars and $2134 for existing cars.

The results of the head shield cost/benefit analysis are different from other published results primarily because of a redistribution of accident dollar losses. Previously, accident dollar losses had been assigned to the tank element that failed. That is, if during an accident, a tank head was punctured with a resulting loss of lading and subsequent fire, the damage caused by the lost lading was assigned to the category of head puncture. Similarly, if the tank shell was punctured, the losses were assigned to the category of shell puncture. Using this rationale with the relatively small amount of historical data available, results in the conclusion that shell punctures which accounted for only 18 percent of the lading spills were responsible for 68 percent of the dollar losses. The historical data are too limited to provide the correct distribution of losses between head and shell punctures. If enough tank car accidents were investigated over a long period of time, the dollar loss distribution would be expected to match the puncture distribution, inasmuch as shell punctures do not inherently produce more costly losses than head punctures. Supporting evidence is presented in the main body of this report indicating that dollar losses are strongly related to puncture distribution for a more extensive set of data including all classes of tank cars.

Modified Couplers

Couplers are the puncture source for the majority of head punctures. Modified couplers have been proposed which prevent vertical disengagement during a derailment or other abnormal operation condition and thereby reduce probability of head punctures. The cost/benefit analysis of modified couplers also requires a redistribution of losses from previously published reports. The economic benefit of modified couplers depends on the basis of comparison. Both type E and type F couplers have been used on tank cars. Modified E and F couplers were found to be cost beneficial compared to either standard E or F couplers. For example, the economic benefit of modified type E couplers compared with standard type E couplers was found to be $491 for cost of capital assumed to be 10 percent. At zero interest rate, the net savings would be $833.

Thermal Insulation

Thermal insulation may be applied to the outside of a tank to prevent or delay overheating and overpressurization of a tank exposed to fire which could lead to tank rupture and greatly expanded damage area. Two types of thermal insulation have been considered: steel jacketed insulation as presently used on type 105A cars and proposed thermal shield coatings that are applied directly to the outside of the tank. Thermal shield coatings have been proposed because of their possible overall lower cost. Because thermal shield coatings are only in the proposed stage, their cost, effectiveness, and life expectancy are not known. Therefore, the cost/benefit analysis considered these factors parametrically. The best estimates of the initial cost of thermal shield coatings which would not cause any increase in lifetime operational costs are given in Table 1.

Table I

BEST ESTIMATES OF "JUSTIFIABLE" COST OF 100% EFFECTIVE THERMAL SHIELD COATINGS ON 112A/114A TANK CARS

LIFE OF THERMAL SHIELD, YRS.	"JUSTIFIABLE COST," $	
	10% INTEREST RATE	0% INTEREST RATE
1	429	451
5	1772	2255
10	2850	4510
15	3504	6765
30	4284	13,530

This "justifiable" cost is the amount that could be spent today on a thermal shield coating resulting in sufficient savings over the life of the coating to repay both application and interest costs on the assumption that the coating is completely effective in preventing fire damage due to the tank car. "Justifiable" cost only relates to the cost that is economically justifiable in comparison with present conditions. Justifiable cost is not a function of social consequences nor does it consider the possible alternative of the increased shipping costs. The analysis also applies for conventional jacketed insulation except that the savings due to the lack of additional corrosion protection would not be realized. Conventional jacketed insulation would then only be justified if it were less costly, or if the life of the shield were expected to be longer, or if the effectiveness were greater than for thermal shield coatings. Longer shield life for conventional insulation has been found to be obtained when compared with coatings tested to date. Because the practical applied thickness may be greater, jacketed insulation would probably provide greater thermal protection and, therefore, effectiveness more closely approaching 100 percent. For jacketed insulation, the best estimate of "justifiable" cost at 100 percent effectiveness would be $2907 for 30 years shield life at 10 percent interest rate. At zero interest, the corresponding value would be $9180.

Tank Material Changes

A change in tank material was examined as a possible solution on three bases. First, a higher strength tank material would resist puncture more effectively. Second, a change in material might prevent propagating failures. Third, a material with more high temperature strength would resist rupture to a higher temperature-pressure limit and, therefore, be more effective in preventing rupture. However, this modification was found to provide only minor benefit relative to cost compared to the other possible modifications.

Safety Relief Valve Modifications

To prevent tank car rupture during fire exposure, modified safety relief valves could be utilized to prevent excessive internal tank pressure. Increased area valves and modifications to insure vapor discharge were considered as possible methods of maintaining tank pressure at least at the specified design pressure but tanks could still fail at this pressure at high tank temperatures. The most viable valve modification was found to be a valve which is actuated by excessive lading temperature and then remains open to relieve the tank pressure to ambient, thereby preventing rupture. The justifiable cost of a valve system which would be 100 percent effective in preventing losses from fire exposure would be $2907. Additional studies are required for determining the effectiveness and cost of this system, but it appears to have a high potential as a cost beneficial modification.

Economic Sensitivity

In determining an upper bound consistent with a viable service, a sensitivity study was performed to determine the effect of increased tank car cost on the delivered price of the shipped commodity even if the car modification were not strictly cost beneficial. This study was limited to the shipment of LPG a distance of approximately 800 miles in 112A340W type cars. The conclusion is that for a tank car carrying only LPG, a 10 percent increase in the tank car cost would produce only an 0.5 percent increase in the delivered cost of LPG. Since LPG is probably the lowest priced commodity to be shipped in the noninsulated pressure cars, this 0.5 percent represents an upper bound on price increase. Similar conclusions would be obtained for trips of different lengths.

Supporting Studies

This report reviews the thermal research on tank cars that has been conducted at Calspan and other facilities. Improvements have been made in the Calspan tank car thermal model which was previously developed. The thermal model represents a tank car enveloped by fire either upright or rolled over at any angle. The tank car geometry is described by inputs for its length, diameter, shell thickness, number of relief valves, their position along the tank, their flow area, discharge coefficient, and the tilt or roll angle from the vertical. In addition, if external insulation is present, it is specified by its thickness, thermal conductivity (which may be varied with temperature) and the product of density and specific heat.

The heat input from the fire is described by inputs for its temperature, emissivity, and the heat transfer coefficient for convective heating. These quantities may be varied around the tank. Heat input to the lading is described for liquid and vapor separately by a heat transfer coefficient.

Liquid heat transfer coefficients are computed by equations that represent curve fits to experimental data, and are valid for propane only.

The model computes heat penetration to the lading, which results in a computed rise in temperature of the external insulation, if any, the tank shell, the vaporized lading, and the liquid lading. In computing the external heating, heat is reradiated to the fire at increasing rate, and convective heating decreases as the outer surface temperature rises, resulting in a reduced heat penetration to the lading. The steel of the shell is described by burst pressure tables that are based upon ultimate strength, and are prepared by calculating burst pressure from simple thin shell relations. The main body of this report fully describes the model and the Appendix includes a program listing.

One-fifth scale and full scale tests of tank cars exposed to fires have been conducted by the Naval Ordnance Laboratory and the Ballistic Research Laboratories. In all of these tests, a full size safety relief valve was used. The most significant observations made form the data of these tests are:

1. The heat flux to a full-scale tank from a JP-4 fire was of the order of 25,000-35,000 But/ft^2 hr to the wetted surface or roughly 4 times that assumed in the AAR Specifications for Tank Cars determined from $Q = 34{,}500A^{0.82}$.

2. The relief valve had insufficient vapor flow capacity to limit the tank pressure to 306 psig as required by the specifications or to below the tank test pressure (340 psig).

3. Vapor flow rates of the valve were about as predicted.

4. A tank failed at about the pressure-temperature conditions indicated by uniaxial strength data.

5. There was considerable temperature stratification within the tank before valve opening. This caused valve opening sooner than for uniform

temperature conditions. After valve opening, saturation conditions generally prevailed. The thermal model does not include provision for stratification and therefore does not calculate correctly before valve opening.

Areas Requiring Further Research

Technology presently exists for building a prototype tank car that is substantially safer than those presently in use and that is, or is nearly, cost beneficial. Head shields could be installed to reduce punctures and jacketed insulation similar to that on type 105A cars could be used to reduce rupture of cars exposed to fire. However, even more optimal designs could be formulated after additional research. The external and internal effects of a tank car have not been completely defined. In particular the effects of high heat fluxes over small areas (torching) have not been determined. Also, the effect of temperature stratification within the lading has not been fully assessed. Very little has been experimentally determined relative to safety relief valve operation while flowing LPG vapor or liquid in a situation similar to that experienced in a fire. The actual valve operational mode is not known. Testing of modified couplers and head shields under dynamic conditions and extensive analyses are required to fully assess their performance under derailment conditions. The main thrust of research on thermal shield coatings should be directed to evaluating the ability of low cost coatings to withstand ten years or more of railroad type service and still be able to provide significant insulation if involved in a fire.

I. INTRODUCTION

Tank cars have been under effective Federal regulation since July 1, 1927, when the Interstate Commerce Commission issued a set of specifications for "Tank Cars Handling Explosives, and Other Dangerous Commodities." The car, which on this date became the ICC105 class car, had been originally specified in 1918 by the Master Car Builders' Association (MCBA).

The tank had an especially heavy construction and was developed to transport volatile flammable products whose properties were such as to involve danger of loss of life in the event of rupture. The outstanding features of these cars, other than their rugged mechanical construction, was the requirement that they have at least 2 inches of insulation covered by a jacket of 1/8 inch thick steel.

In the early 1930's, the shipment of liquefied compressed hydrocarbon gases were confined to these specially designed tank cars. The shippers, however, began looking for a tank car designed to the characteristics of their products. As a result, a new class of cars was specified, ICC 105A200 thru ICC 105A600 cars, which allowed minimum plate thickness, safety relief valve start-to-discharge pressure, test pressure, etc. to be varied directly with tank design pressure. All of these cars , and in particular the 105A300, which was to transport liquefied petroleum gas (LPG), still required a minimum thickness of 2 inches of insulation and the 1/8 inch steel jacket.

The drive for economy led to still another change in these specifications about 1960. This car specification, for the 112A400W series car, was an outgrowth of the 105A400 specification except the removal of the requirement for insulation. Concurrently, changes in other governing specifications allowed the removal of expansion domes, side running boards, and an increase in the allowable weight on the rails. (Series 114A cars are similar to 112A cars except for valving and these two series of cars will be treated as one.) These changes, acting together, allowed car capacity in service to reach first

20,000 to 30,000 gallons and then on a prototype basis 50,000 to 60,000 gallons. The Department of Transportation has since set limits of 34,500 gallons and 263,000 pounds total rail weight.

Tank cars carrying flammable ladings have been involved in numerous accidents over the years. Particularly since the advent of class 112A/114A cars, the amount of dollar losses as a result of tank car involvement in accidents have been substantial. The Railway Progress Institute (RPI) and the Association of American Railroads (AAR) have undertaken a cooperative program titled Railroad Tank Car Safety Research and Test Project. The RPI/AAR has determined that there were 3853 tank cars damaged in 2321 accidents in the United States and Canada during the six year period 1965 through 1970.[1] It was determined that total losses due to mechanical damage of tank cars were more than $23,000,000 and total losses due to fires from tank car ladings were over $15,000,000. (These values are not necessarily additive because some of the fire losses were initiated by mechanical damage.) The largest accidents reported were at Laurel, Mississippi, January 25, 1969, $7,800,000, and Crescent City, Illinois, June 21, 1970, $1,900,000. Since the time of the RPI/AAR report, there have been several accidents each resulting in losses of millions of dollars.

The following sequence of events typifies an accident involving a tank car with compressed liquefied gas lading which results in large dollar losses. During a derailment or other abnormal occurrence, a tank car is punctured and the lading is subsequently ignited. The fire causes some damage in the surrounding area and heats one or more tank cars that have remained intact during the initial accident. The tank cars that are heated by the fire react as follows. As the lading increases in temperature, it expands and tends to fill the ullage space with liquid. After the ullage space is filled, the liquid continues to expand and forces open the safety relief valve with which each tank must be equipped. On further heating, the saturation pressure of the lading reaches the start-to-discharge pressure of the relief valve and the liquid level recedes as lading is released. While the lading is being heated, the tank shell is also increasing in temperature. Because of the low heat transfer coefficient from the tank shell to gaseous portions of the lading, the portions of the shell in contact with gaseous lading increase in temperature at a faster rate than portions of the shell in contact with liquid lading. If at any time during the heating, the stress in the shell due to internal pressure, and to a small degree thermal stress, exceeds the strength capability of the shell material at temperature, the tank will fail. Tank failures have often taken the form of large, rapidly propagating cracks with large, nearly instantaneous, release of burning lading. As the pressure is released, large amounts of lading are converted to the gaseous state. The result has been that portions of tanks weighing tons have rocketed hundreds of feet with resulting physical destruction and fire spread. Even without rocketing, the area of damage increases greatly when a tank ruptures.

The Federal Railroad Administration (FRA) of the U.S. Department of Transportation (DOT) has undertaken a research effort to improve the safety of moving hazardous liquefied compressed gases. Five phases are contemplated with the end result being the specification of a safe and economically practical tank car. The broad objectives of the first phase of the research effort were to:

1. Provide the basis for defining practical and economical safety improvements which can be either retrofitted to in-service tank cars or incorporated into the design and manufacture of new tank cars, and

2. Define the safety research gaps which must be remedied before a prototype tank car can be designed to optimal safety/economic considerations.

To carry on this effort, a multiphased program was developed which can be outlined in the following four categories.

1. Review tank car design specifications and codes along with operational procedures to determine the feasibility of changes to improve tank car safety.

2. Thoroughly review the ongoing research programs of the FRA and RPI/AAR.

3. Determine those design changes which can be applied either to existing cars, as a retrofit item, or incorporated into new car designs and which will improve safety on a cost beneficial basis.

4. Specify those technical areas in which further research should be accomplished before decisions can be made on the final configuration of the prototype tank car.

The program was later increased to include three specific additional tasks. These three tasks are listed below.

1. Perform a cost/benefit analysis for head shields applied to 112A/114A tank cars.

2. Review existing practice on design of tank cars and shipment of liquid ethylene. Also formulate and investigate the effectiveness of modifications to shipping regulations and tank car design specifications.

3. Perform a cost/benefit analysis of thermal shield coatings applied to 112A/114A tank cars. Three reports were issued covering these tasks (Ref. 2, 3, and 4).

Auxiliary tasks were completed simultaneously with the primary tasks listed above. These included: a literature review, a bibliography of which can be found in Appendix A, briefings and communications with several organizations.

II. DESIGN AND OPERATIONAL FACTORS

The purpose of this section is to establish, prior to recommending improvements, the effects on load factors of varying tank car design and operational parameters. The analysis has been divided into two general areas: 1) tank car design factors, and 2) operational aspects related to tank car safety.

A. Tank Car Design Factors

When considering design changes for safety, or other considerations, it is necessary to review governing factors which may limit the freedom or direction of change. Some of the important factors are:

- the physical and chemical properties of the lading;
- structural materials and fabrication technology available at reasonable cost;
- physical limitations--dimensional and weight--required to operate on the U.S. rail system;
- the operating environment.

The operating environment will be considered later, while for the moment, the discussion will be centered on the physical limitations.

There is a body of codes, specifications and design practices for tank cars. Title 49 of the Code of Federal Regulations, Part 179, contains the DOT specifications for tank cars handling hazardous commodities. These regulations are republished in tariff form (currently as part of R. M. Graziano's Tariff #29). Title 49 CFR, Part 179, incorporates by reference, portions of the AAR Specifications for Tank Cars. The latter document is also the controlling standard for tank cars handling non-regulated commodities. By virtue of the fact that portions of the AAR Specifications have the status of law, and also that many cars have been constructed to AAR Specifications alone under the special permit provisions of DOT regulations, the AAR Specifications have major design influence on cars handling hazardous commodities as well.

The AAR Tank Car Specifications incorporate by reference portions of the AAR Specifications for Design, Fabrication, and Construction of Freight Cars. The latter document (commonly called simply "the Design Manual") contains dimensional, weight, and stress allowables.

The Hazardous Materials Regulation Board of the Department of Transportation is the channel by which revisions to the Hazardous Materials Regulations (49 CFR 170-195), which includes tank car specifications, are promulgated.

The bulk of regulated commodities shipped by tank cars move in four classes of cars:

- 103 - "non-pressure" service--expansion dome, full underframe
- 111 - "non-pressure" service--domeless, majority stub-sill
- 105 - pressure service--insulated, full underframe
- 112 and 114 - pressure service--non-insulated, stub-sill

In terms of recent construction, the 111 and 112 series cars predominate. The 112 series cars have been involved in most of the very high cost accidents and have therefore received the majority of attention with respect to adequacy of design.

Of the potential safety improvements which have been considered for application to tank cars, thermal shields, head shields, and modified couplers have been of particular interest. Examining some of the design constraints that must be considered, we will first consider capacity limits, both volume and weight.

Figure 1 illustrates volume and weight limitations by car class and lading. In addition to general relationships, specific examples of individual cars selected from the Railway Equipment Register or the Car and Locomotive

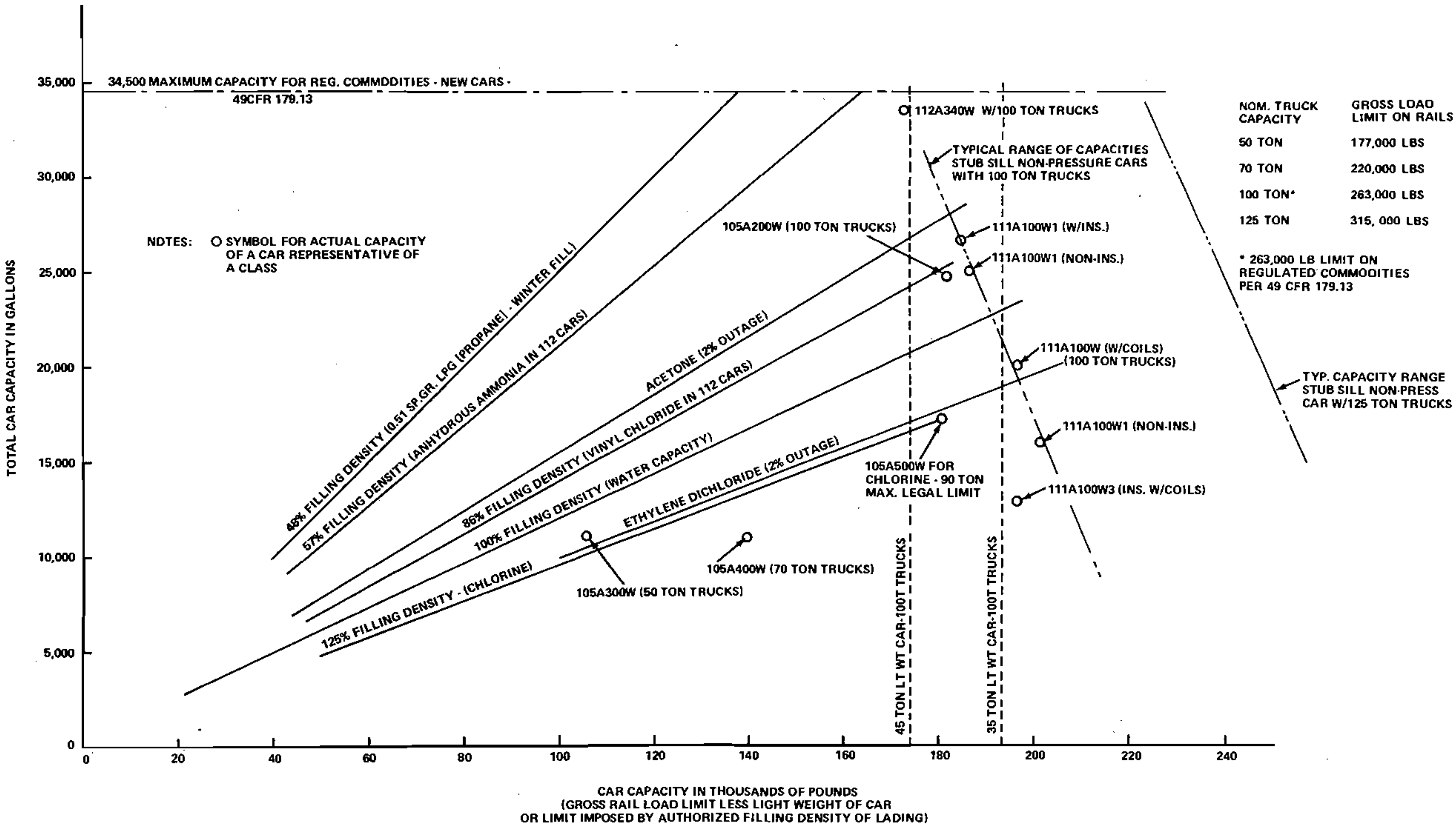


Figure 1 TANK CAR VOLUME AND WEIGHT LIMITATIONS BY CAR CLASS AND LADING

Cyclopedia, typical of a car class, are plotted. Filling density is expressed as the percent of the weight of a full car of water that may be loaded.

Principal bounding conditions for both weight and volume of cars built after November 1970 handling regulated commodities are the legal requirements imposed by 49 CFR 179.13. This paragraph states that a volume limitation of 34,500 gallons and a gross weight on rail of 263,000 lbs. apply. These boundaries appear in Figure 1. The effect of the gross weight restriction in terms of cargo carrying capacity for steel cars is approximated by the line identified as "typical range of capacities for stub-sill non-pressure cars with 100-ton trucks." (Nominal 100-ton trucks have a gross load limit of 263,000 lbs; see Figure 1 for other trucks.) Heavier walls for pressure cars would result in a slight shift in this line towards reduced capacity, with the slope of the line unchanged (for circular cross sections).

B. Capacity Limitations for Non-Pressure Cars

Referring again to Figure 1, observe the intercepts of the lines identified "Acetone" and "Ethylene Di-Chloride" with the previously described "Typical Range . . . 100 ton-trucks" line. Ethylene di-chloride is a material representative of the highest density liquids handled by tank cars. Acetone is typical in terms of density of a broad range of hydrocarbon liquids handled by tank car, and representative of one of the lower density materials hauled. Therefore, virtually all regulated commodities handled in 111 series cars are sharply weight limited rather than volume limited. Cars built will typically have a volumetric capacity yielding close to the gross rail load limit when filled. Therefore, retrofit programs for existing 111 series cars requiring substantial structures would likely require reductions in cargo volume to accommodate extra weight, an economic penalty which would have to be taken into account.

There are a number of 111 series cars in operation with 125-ton trucks (highest capacity 4-wheel truck), and some with 150-ton 6-wheel trucks, or 8-wheel span-bolster arrangements. These cars were either in service before

November 1970 and/or were designed for non-regulated commodities. They are products of the drive for economics associated with higher unit loadings.

Reductions in cargo volumes would also apply to new 111 series cars incorporating weight additions. This reduction could not be offset by fabricating the cars of stronger materials. These cars presently approach critical buckling limits; therefore, increased tensile properties would be of limited value. Note that we are not indicating that reductions in cargo volume would be, by definition, economically untenable. The effect must be considered, however.

Design changes involving moderate changes in physical dimensions would not in general prove a serious problem with non-pressure cars. This is discussed in further detail later.

C. Capacity Limitations for Pressure Cars

Vinyl chloride, anhydrous ammonia, and liquefied petroleum gas (LPG), e.g., propane, constitute the three principal hazardous commodities handled in 112A/114A series pressure cars. Examining Figure 1, we find that ammonia and propane are significantly volume limited. Therefore, weight addition for retrofit or new car design would not effect unit volume carried on these commodities. Vinyl chloride, on the other hand, is severely weight limited by the 100-ton truck restriction. A pre-1970 112A340W car with 125-ton trucks was limited to a gross load limit of 315,000 lbs, whereas new vinyl chloride cars are restricted to 263,000 lbs. That is, a new 112A340W car for vinyl chloride would be required to carry 52,000 lbs. less vinyl chloride than a pre-1970 112A340W car with 125-ton trucks. Therefore, the economics of carrying vinyl chloride in 112 cars has already been strongly shifted, even without considering design changes. In the case of vinyl chloride, a viable alternative car design is currently available in the older 105A200W configuration. In the 105A200W, a larger portion of the volume could be utilized without exceeding the 263,000 lb. gross load limit. No current proposals for design changes or retrofit programs would have anything like the effect of a 26-ton reduction in payload.

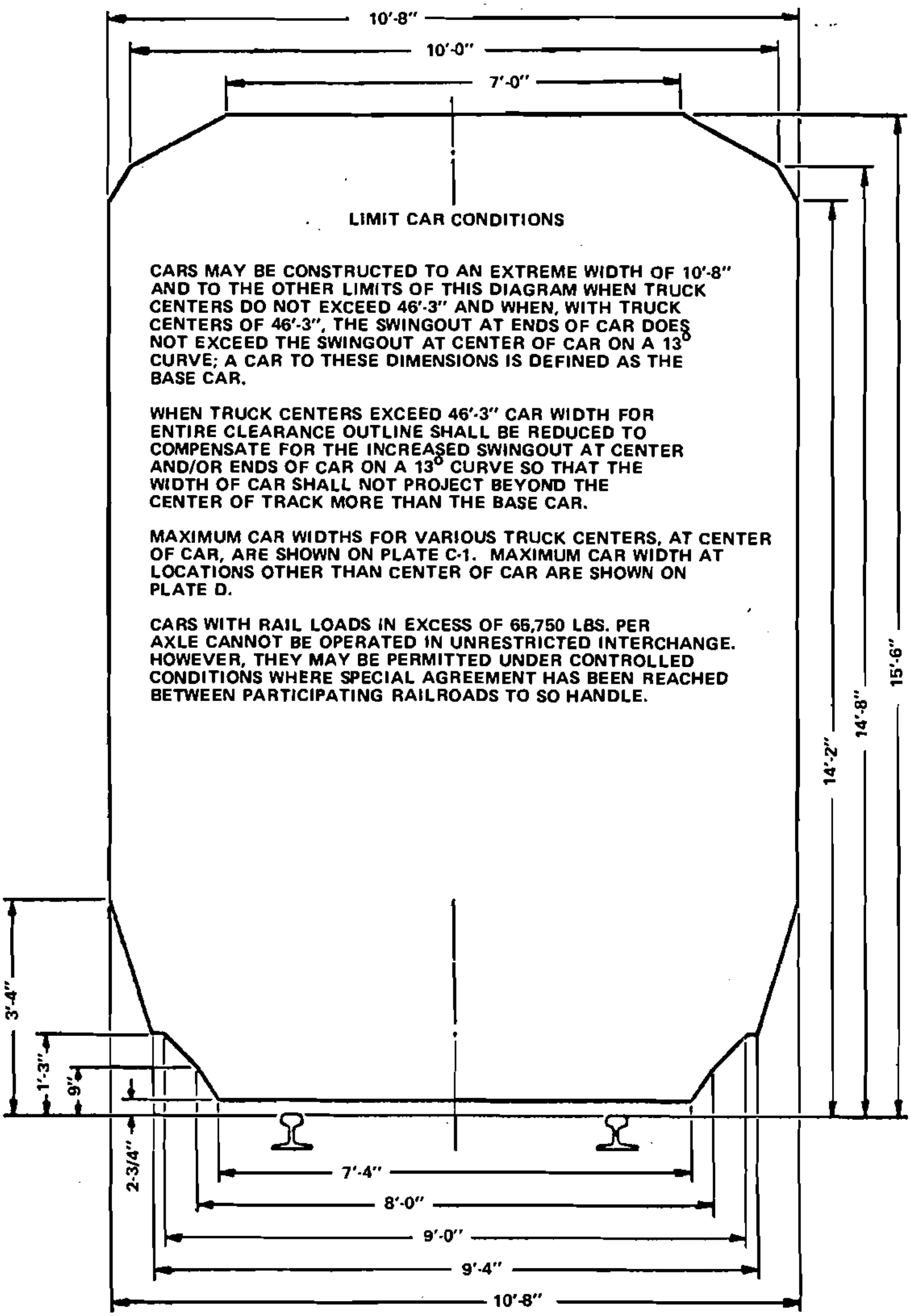
Although adding weight, for example in the form of head shields, to 112A/114A cars used in LPG and/or ammonia service does not present a serious problem, dimensional limitations could be a serious factor with regard to the addition of some form of thermal shielding over the shell area. Inspection of the basic dimensional constraints required to operate on the United States railroad system and comparing them with the diameter-length-volume relationships that apply to tank cars brings this into sharper focus.

Rail cars built conforming to Plate "B" limiting dimensions of the AAR Design Manual may operate in unrestricted interchange service. New tank cars are fabricated to fall within the Plate "B" limiting dimensions or the more generous limiting outline of Plate "C" for restricted interchange service. Figure 2 shows the basic Plate "C" limiting outline for rail cars. Cars fabricated to Plate "C" dimensions can operate on approximately 95 percent of U.S. rail mileage. Figure 3, identified as Plate C-1, defines reductions in maximum car width required for cars with truck centers longer than those of the "base" car. This is to accommodate swingout of the center of the car on curves, which otherwise could foul adjacent structures or track. Plate "D" of the design manual, not shown, is utilized to establish limits for car end overhang. The latter is not usually a controlling factor in tank car design. The maximum width reduction for increased truck centers (hence, longer cars) is very important, however.

Figure 4 plots the volume of a one-foot longitudinal segment of a tank car shell as a function of shell diameter. Intercepts are shown indicating the approximate maximum car length that can be built for a given diameter. We will discuss extreme width restrictions for long cars in more detail after reviewing 112A/114A series car designs currently in existence.

Initial 112A series cars were of what is commonly referred to as a "whalebelly" design, with the larger diameter center section eccentric to the end tubs. Subsequent revision of the AAR design manual permitting higher centers of gravity for loaded cars allowed construction of 112A series cars with a constant circular cross section. This design, sometimes referred to as the

FOR SPECIFIC RESTRICTED AREAS SEE "RAILWAY LINE CLEARANCES"



THE 2-3/4" ABOVE TOP OF RAIL IS ABSOLUTE MINIMUM UNDER ANY AND ALL CONDITIONS OF LADING, OPERATION, AND MAINTENANCE.

Figure 2 - EQUIPMENT DIAGRAM FOR LIMITED INTERCHANGE SERVICE (WILL CLEAR OVER 95% OF TOTAL MILEAGE) STANDARD PLATE C

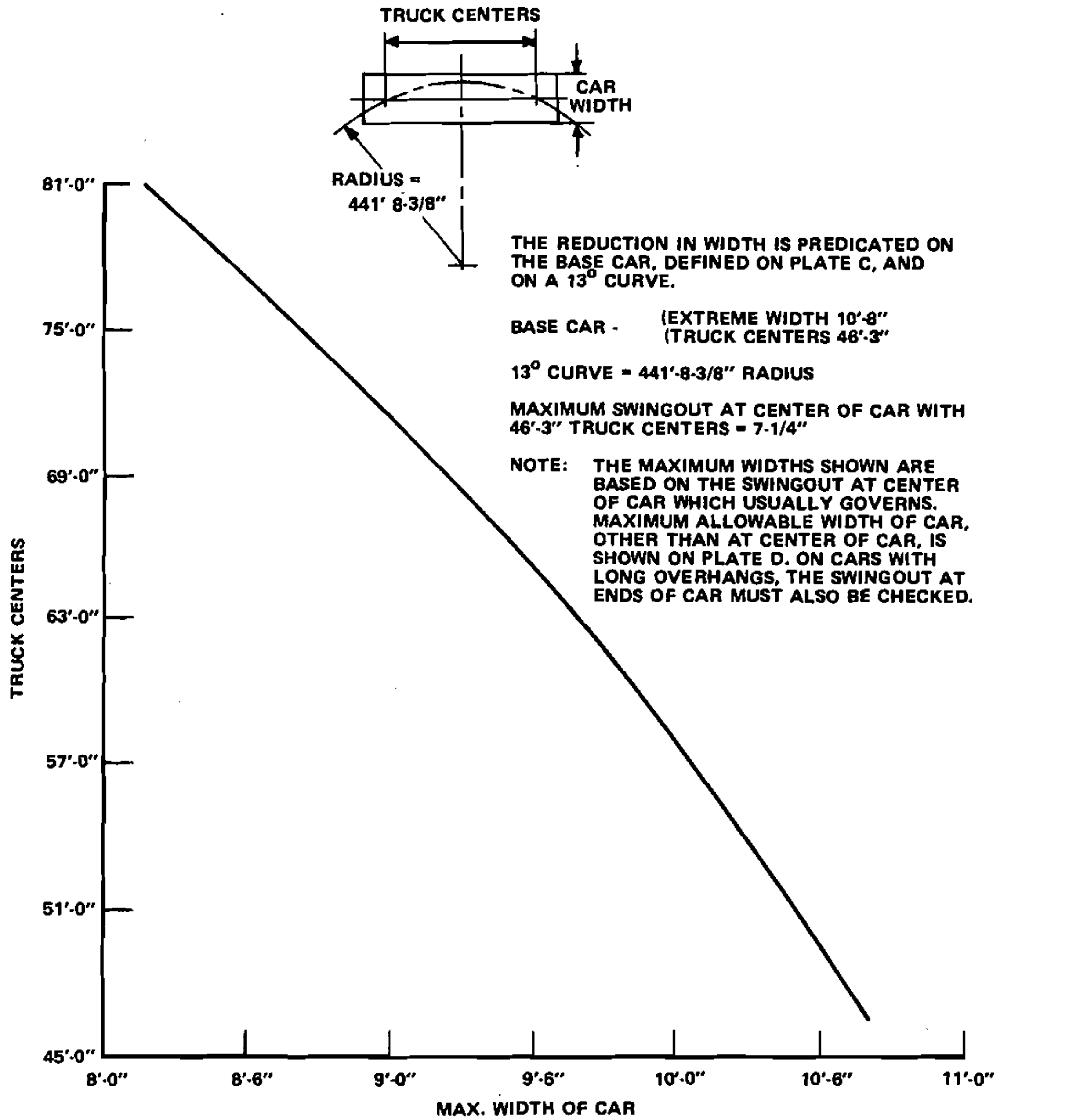


Figure 3 - NOTE: FOR USE WITH PLATE "C"– PLATE C-1
MAXIMUM WIDTH OF CARS WITH VARIOUS TRUCK CENTERS STANDARD ASSOCIATION OF AMERICAN RAILROADS MECHANICAL DIVISION

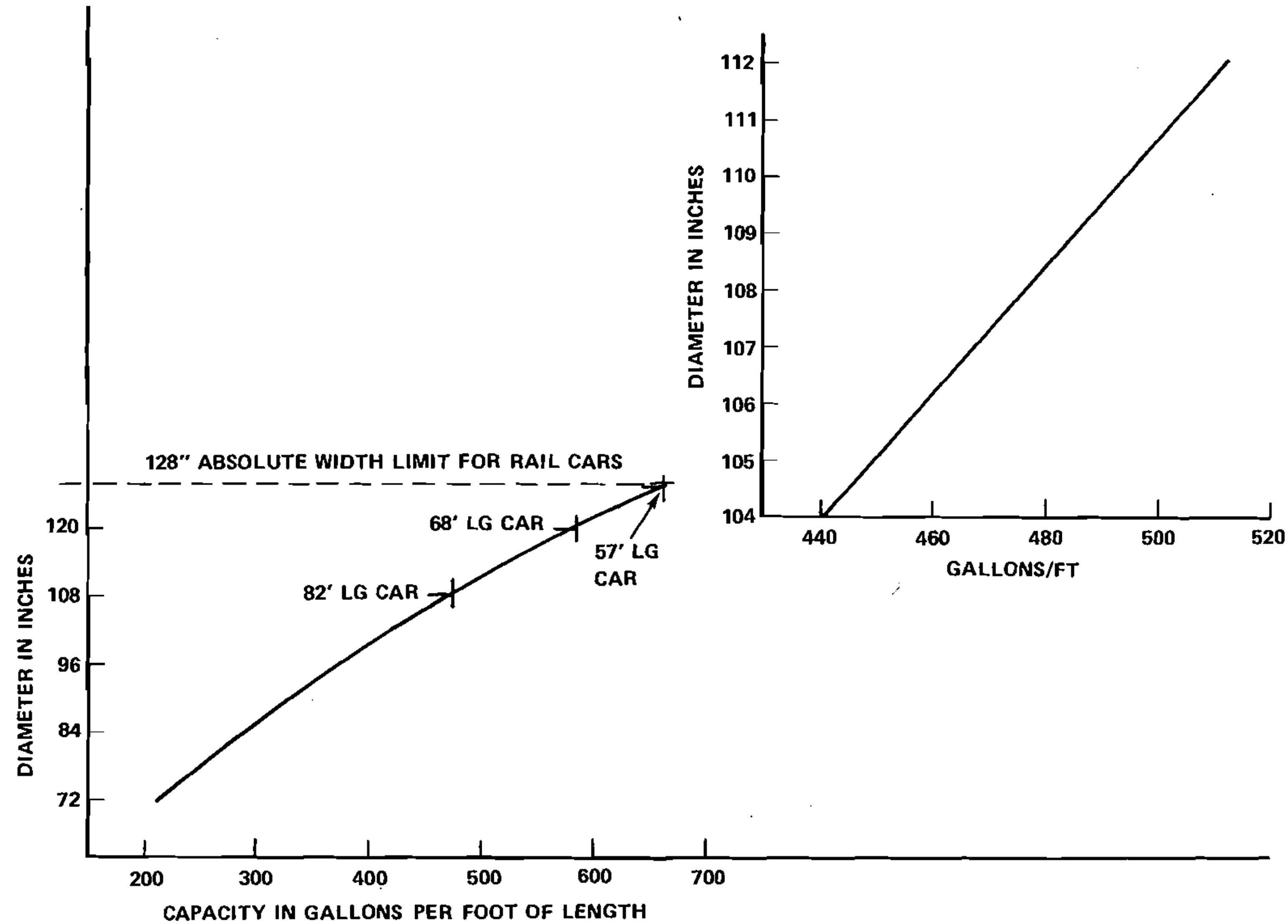


Figure 4 - VOLUME OF TANK CAR CIRCULAR SEGMENT PER FOOT OF LENGTH

"hot dog" configuration permits economies in fabrication. These two basic designs need to be considered in any retrofit program. Taking one particular car of each type, we can examine the problem of adding a thermal shield.

Example One: GATX 112A340W - "whalebelly"

Length over strikers:	64'8-1/2"
O.D. of end sections:	~106"
O.D. of mid section:	~120"
Truck centers:	53'9-1/2"

Taking our limiting dimension from Plate C-1 (Figure 3), we find the extreme width limit is 123". Therefore, only 1-1/2" maximum thickness is available at the horizontal mid-line of the car for applying a thermal shield. Referring to Figure 2, little problem exists with respect to the remainder of the car outline.

Example Two: GATX 112A340W - "hot dog" - 33,500 gallon

Length over strikers:	63'4"
Outside diameter:	~120-1/4"
Truck centers:	52'4-1/2"

From Figure 3, we determine the maximum width allowable to be 124". Therefore, approximately 1-7/8 inches thickness is available for a thermal shield at the horizontal mid-line for a car designed according to Plate C. To conform to the limiting dimensions of Plate B, less than 0.5 inch of thermal shield would be available for this car. It should be noted that there are variations from builder to builder, and in groups of cars from a given builder. Therefore, the dimensions given can be considered typical, but not absolute.

Considering new cars, one could postulate a reduction in shell diameter to accommodate a thermal shield, plus an increase in length in an attempt to hold the unit volume equal to current design. From Figure 4 we can determine that a reduction of one inch in diameter requires over a foot of additional length to maintain volume. Unfortunately, however, as we extend the length, we can see from Figure 3 that the allowable outside diameter of the car must be reduced of the order of 0.8 inch in diameter per foot of added length if truck position relative to the end of the car is considered constant. Therefore, the lengthening technique is not viable as a means of securing space for thermal shielding given a circular cross section.

Non-circular cross sections have been utilized in past experimental applications to secure very large capacities in pressure cars, but the fabrication technique is more expensive. However, a scheme of non-circular insulation while maintaining the same circular tank is discussed below which eliminates much of the added expense of non-circular cross-sections but provides significant insulation protection.

To maintain cars to approximately their present mid-plane dimension while providing significant insulation protection requires that the insulation be of non-circular geometry. A possible configuration is shown in Figure 5. Away from the mid-plane of the car, it would be possible to increase the insulation thickness at the sides because of the decreased swingout. For example, at the truck centers of the "hot dog" car described above, about 9 inches of thickness are available for insulation. However, it is assumed that the difficulty of forming the complex shape required to obtain maximum allowable insulation thickness eliminates a configuration with varying cross-sections from consideration.

For a car of uniform cross-section as shown in Figure 5, calculations indicate that 1 inch of insulation with a thermal conductivity of 0.1 Btu/hr ft^{o}F would be sufficient to reduce the heat input to the tank so that the present valve (as specified for an uninsulated car) would have sufficient flow capacity to prevent pressure buildup in a tank during a fire. The insulation would also have the effect of tending to prevent high tank wall temperatures in areas in contact with the vapor space.

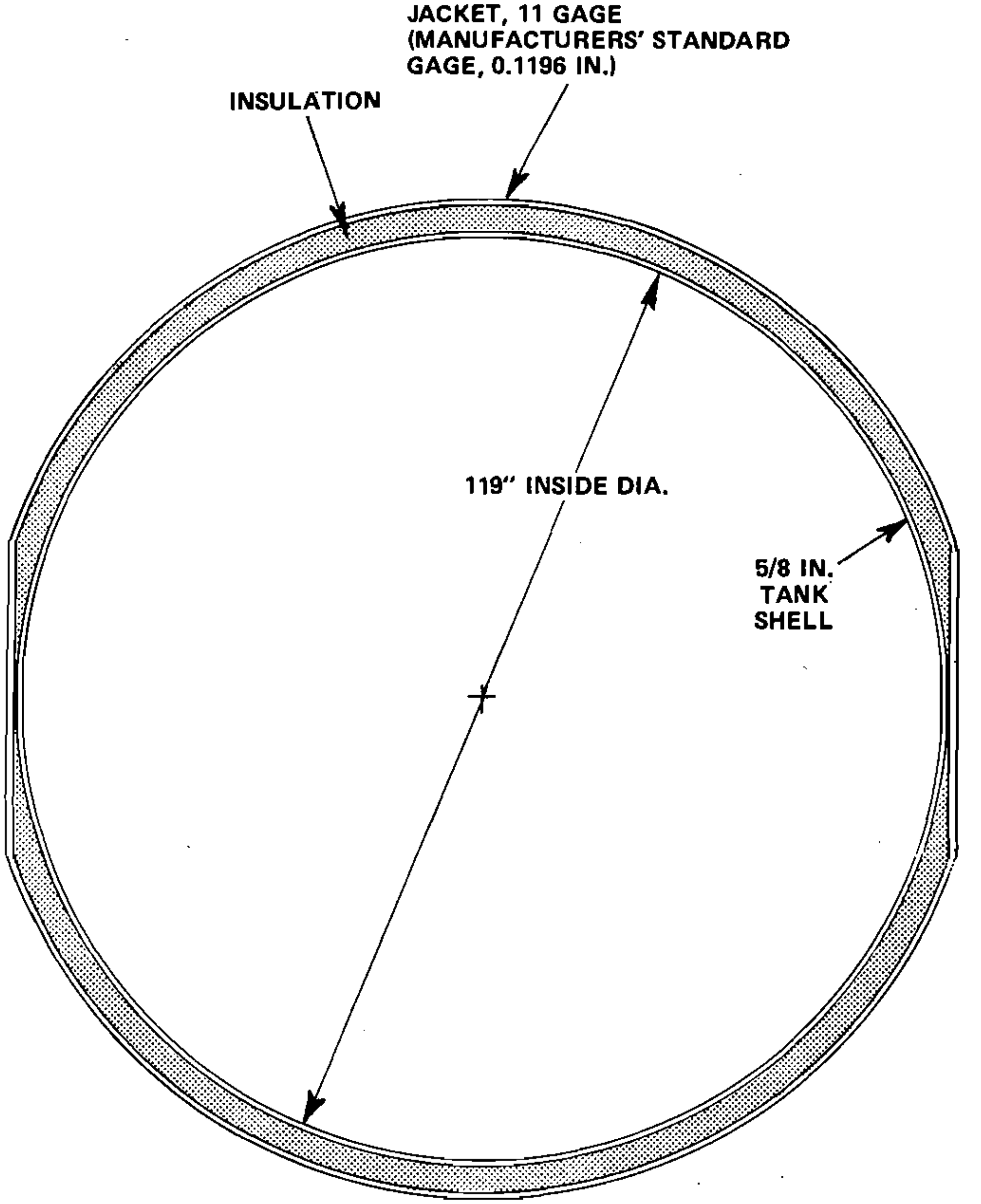


Figure 5 POSSIBLE INSULATION TECHNIQUE FOR 33,500 GALLON 112A TANK CARS WITHOUT INTERNAL VOLUME REDUCTION

D. Materials of Construction

Although materials of construction are discussed in more detail elsewhere in this report, it is appropriate to have a capsule discussion regarding some items of current practice with respect to design limitations.

The majority of new pressure tank cars are constructed of TC-128 steel, a carbon-manganese steel similar in specification to ASTM A612. A typical steel for 111 series cars would be the less costly (by approximately 15 percent) A515 Grade 70. The basic tensile and yield strength properties of these steels approach the maximum that can be effectively utilized in stub-sill design for the respective pressure levels involved. This is due to buckling being a controlling factor. Therefore, relatively little is to be gained from the strength standpoint in considering special alloys or other metals. In terms of high temperature properties, gains can be made with substitutions of other metals or alloys. Using insulation, however, is more attractive from the standpoint of performance and probably economics. A major benefit from insulation is limiting heat input into a car, hence limiting safety relief requirements for release of hazardous contents.

Low temperature properties can be improved, considering both currently used steels, or substitute alloys. Normalizing TC-128 to improve low temperature properties, now only required for liquid carbon dioxide service, would add approximately $600 to 112A340W car cost (shell and head).

A potential substitute steel, a low carbon-manganese, molybdenum, columbium alloy developed originally for arctic pipelines, has excellent low and high temperature properties, and has good weldability. This material would add approximately $3000 to the cost of a 112A340W car. Cost/benefit relationships are developed in Section IIIE.

E. Operational Aspects Related to Tank Car Safety

We have previously discussed a number of equipment related factors of importance to tank car safety. Turning now to some of the operational factors, we will first review briefly rail accident history.

In the decade of the 1960's division of accidents by causes, considering only negligence of employees, defects in equipment, and defects or improper maintenance of way and structures was approximately:*

Negligence:	38%
Equipment:	42%
Maintenance of Way:	20%

In recent years the maintenance of way related accidents have increased. Consider the 1972 derailment history of the state of Iowa as a microcosm of the national problem. There were 346 reportable derailments in 1972, an increase of 120 over 1971 (Reference 5). Equipment failures accounted for approximately 22 percent of the derailments and 30 percent of the damage. Failures related to maintenance of way and structure accounted for 52 percent of the derailments and 59 percent of the total damage. Four railroads account for the majority of ton-miles hauled in Iowa, the Rock Island (CRI&P), Northwestern (C&NW), Milwaukee (CMSt.P&P) and the Burlington Northern. Reported derailments for each in 1972, as reported in the Federal Annual Summary of Railroad Accidents Bulletin, are as follows:

CRI&P	-	136
C&NW	-	125
CMSt.P&P	-	39
BN	-	12

Inasmuch as the amount of ton-miles hauled per year is of the same order for these railroads, it appears that factors other than ton-miles governed the reportable derailments. It is also of interest to note that BN for which the fewest derailments are attributed is the most prosperous of the four.

*"Summary and Analysis of Accidents on Railroads in the United States," Interstate Commerce Commission, Calendar years 1960-1969.

The point of the discussion is that derailment must be considered a part of the tank car environment, and that there is little reason to expect immediate relief from the problem, as it is multi-faceted. Changes in tank car design to ameliorate the effects of an accident have been discussed. Such changes require substantial lengths of time to implement. Many operational changes, on the other hand, could be implemented rapidly. As with equipment alterations, the questions are how effective will the change be, and what will it cost.

Two obvious areas of operating practice to be examined are speed and spacing. The position of the Association of American Railroads (AAR) has been adamant against imposed speed restrictions for trains carrying hazardous commodities by tank car. Similarly, they have opposed regulations with regard to spacing of tank cars carrying hazardous commodities within a train. Their objections have been two-fold: (1) adoption would increase hazard rather than decrease it, and (2) adoption would be costly. Page five of Reference 6 develops some of the arguments the AAR has made in the past. A more recent communication to the NTSB from the AAR relative to rail safety is given in Reference 7. Examination of AAR objections leads to the conclusion that many arguments to the effect that hazard would be increased lack validity. On the other hand, there is considerable support for their assertions with respect to costs. Before delving further into these points, a review of existing regulations, both voluntary and mandatory, with regard to speed and spacing, is in order.

The Association of American Railroads, principally through the Bureau of Explosives (B of E), individual railroads, and the Federal Government have promulgated a great deal of regulations with respect to packaging, labeling, loading and unloading practices for hazardous commodities. Over the road operating practices have been principally the province of the individual railroads, however. Some Federal Regulations exist in this area, many derived from B of E recommendations. They are contained in 49 CFR 174. Excerpts from these regulations applicable to tank car transport are appended. Of particular interest is Paragraph 174.589, (i) and (j), covering position in trains of placarded loaded tank cars.

"(i) Position in train of loaded placarded tank car. In a freight train or a mixed train, except a train consisting entirely of placarded loaded tank cars and as provided in paragraph (j) of this section, a placarded loaded tank car shall when the length of the train permits, be not nearer than the sixth car from the engine, occupied caboose or passenger car.

(1) When the length of the freight train or mixed train will not permit it to be so placed, it shall be not nearer than the second car from the engine, occupied caboose or passenger car.

(2) When transported in a freight train engaged in "pickup" or "setoff" service, a placarded loaded tank car shall be not nearer than the second car from both engine or occupied caboose.

(j) Separating loaded tank cars placarded "Dangerous" from other cars in trains. In a freight train or mixed train either standing or during transportation thereof, a placarded loaded tank car must not be handled next to:

(1) Occupied passenger car, other than cars occupied by gas handlers and authorized personnel accompanying shipment.

(2) Occupied combination car, other than cars occupied by gas handlers and authorized personnel accompanying shipment.

(3) Any car placarded "Explosives."

(4) Engine or occupied caboose, (except when train consists only of placarded loaded tank cars).

(5) Any car placarded "Poison Gas" or "Flammable Poison Gas."

(6) Wooden under-frame car (except on narrow gauge railroads).

(7) Loaded flat car, other than specially equipped cars in trailer-on-flat-car service or flat cars loaded with automobiles, trucks, or trailer bodies which are secured by means of a device or devices designed and permanently installed on the flat car for that purpose and of a type generally accepted for handling in interchange between railroads. (Note: Flat cars equipped with permanently attached ends of rigid construction shall be considered as open-top cars. See subparagraph (8) of this paragraph.)

(8) Open-top car when any of the lading protrudes beyond the car ends or when any of the lading extending above the car ends is liable to shift so as to protrude beyond the car ends.

(9) Car, trailers or truck bodies on flat car with automatic refrigeration or heating apparatus in operation; car, trailers or truck bodies on flat car with open-flame apparatus in service of with internal combustion engines in operation.

(10) Car, trailers or truck bodies on flat car containing lighted heaters, stoves or lanterns except when car is occupied by gas handlers or authorized personnel accompanying shipment.

(11) Car loaded with live animals or fowl, occupied by an attendant."

The mechanism by which railroad employees are informed of these regulations is usually as part of the Operating Rules, or as Special Instructions issued in General Bulletins. Seldom is the statutory nature of the requirement stated.

There are no speed restrictions by the Federal Government directed specifically to tank car transport of hazardous commodities. Some individual railroads have adopted speed restrictions with regard to hazardous commodities. The Southern Railway, for example, has restricted the speed of convoy shipments of LPG. The Detroit, Toledo, and Ironton forbid movement of hazardous commodities in 112A/114A series cars without special autnority from senior management.

Let us examine some relationships between derailment and speed. The AAR has stated that "no causal relationship exists between higher authorized speeds and derailments." This is a moot point with respect to part of our discussion, since our primary interest is in the relationship between the severity of damage and speed given a derailment. Severe damage and loss of shell integrity is generally associated with those cars which have actually left the rails. Examination of actual derailments and computer simulations performed as part of the RPI/AAR Tank Car Study (Reference 8) show an approximately linear increase in number of cars derailed with increasing speed over typical operating ranges between 20 and 50 mph. Considering intensity of stress fields as a function of speed, it is clear that impacts of derailing cars with stationary objects will certainly be worse with increasing car speed. Considering damage potential as a function of kinetic energy, the potential would rise as the square of the velocity. The magnitude of forces involved with impacts between derailing cars moving at differential velocities would not be expected to be a strong function of initial speed of the train at the moment of derailment. These forces would appear primarily as the result of decelerations due to friction between the car and ground, with friction coefficients being relatively independent of velocity. On the other hand, the time over which these forces would act, hence time over which damaging events are likely to occur, is strongly dependent upon initial speed. The evidence that these forces are capable of violating the integrity of a tank is ample. It is interesting to note in Reference 6, page 5, the AAR states that ". . . speed reductions in open country introduce increased accident exposure through adjustment of slack, possibility of break-in-two, hot wheels and car fires from increased braking requirements." This statement is literally true; however, their apparent equating of "speed reductions" to "speed restrictions" is without foundation. The necessity of speed reductions is more common with relatively high speed operations to accommodate such things as local changes in profile, curvature, crossovers, restrictive signal indications, etc. while proceeding through a particular operating territory. On the other hand, the choice of a more moderate peak operating speed can permit near uniform speeds to be maintained, hence avoiding many speed reductions. Only if speed restrictions were in the 15 mph category and under would factors such as harmonic oscillations or harsh slack action with emergency application be expected to exert a strong influence.

The question of longer "exposure time" for slower moving trains must be considered. Accidents between meeting and passing trains will occur (e.g., Crete, Nebraska 2/18/69). The probability is quite low, but would be expected to increase if, for the purpose of discussion, we assume speed has no effect on accident frequency. Examination of accident history indicates, however, that single train accidents predominate by far. A slight increase in frequency of between train accidents could not be expected to offset reductions in the severity of single train accidents resulting from the adoption of speed restrictions. Considering increased "exposure time" of the single slower train, the question of operating time dependency on frequency of accidents comes into focus. Information available does not permit direct assessment of a possible relationship. We note that the RPI/AAR accident study (Reference 1) considered only mileage and calendar time. However, the kinetic energy, which must be dissipated during a derailment, increases with the square of velocity while the exposure time is only directly related to it. As already stated, the relations between accident severity and kinetic energy have yet to be established.

However, given the non-linear relation of velocity, a greater reduction in accident damage, and therefore cost, would be expected with reduced velocity and increased exposure time.

Reference 6, page 6, states that "critical analysis of train accidents has conclusively shown that no causal relationship exists between higher authorized speeds and derailments." No supporting evidence or references are cited. Considering the numbers of accidents due to equipment failure and failures in the track system cited at the beginning of this discussion, it is obvious that failure stress levels are being exceeded with regularity given current conditions of equipment and track in many areas of the rail system. It can be shown that stress levels in critical running gear, suspension systems and track structures increase with speed. Given these facts, it is difficult to avoid the conclusion that a causal relationship exists.

Therefore, while perhaps a defense can be made against imposition of blanket speed restrictions on trains handling hazardous materials by tank car, there is little question that lower speeds in many areas would reduce significantly the number of <u>severe</u> accidents.

Cost factors are far more variable with respect to operating changes than with equipment modification. With respect to speed, costs, hence cost/benefit factors, will be strongly dependent on the railroad, and even a particular route on a given railroad. Therefore, blanket statements of economic margin tied to a specific speed cannot be made. Unfortunately, those rail routes where speed restrictions from a safety standpoint would be most beneficial, are frequently those where the greatest increase in immediate operating costs would occur.

As part of the current everyday operating "traffic equation," trains moving at various speeds, with or without pick-up and set-off assignments, and non-uniformly distributed over a system, must be accommodated in a manner to maximize profit. Advance planning to optimize scheduling where possible, and flexible physical plant and communications to handle non-optimum traffic conditions which can occur are obviously required to approach that goal. Consider first a road with a transportation planning center, microwave data and voice links, radio nets, and reverse signalled double track with centralized traffic control. Such a system could accommodate, for example, occasional 30 mph trains interspersed with other higher speed traffic with minimal difficulty and expense. Given typical divisional runs and the nature of time/mileage calculations for crew wages, additional labor expense for added running time would be minimal.

A second example produces a different picture. Consider a single track automatic block signal protected line operated by train order. Authorized speeds of 60 mph for freight trains are permitted, and the line is subject to seasonal variations in traffic which tax capacity. Introduction of a mandated 30 mph operation into this inflexible arrangement could easily produce operational dysfunction, with soaring labor costs and non-economical utilization of equipment.

Many other conditions could be postulated with differing economic factors. The main point is that consideration of speed restrictions should not be rejected out ofhand--or introduced blindly.

We have previously noted that some statutory restrictions already exist with respect to placement of placarded tank cars in trains. No general requirements for "buffer cars" exist, however. Loading rack, switching, and way freight requirements virtually preclude any significant buffer arrangements, even if it were otherwise deemed desirable. This leaves through-freight considerations. We will examine first the probable effect of separated placement of tank car ladings whose primary hazard is toxicity, rather than flammability. The potential for at least one in a group of cars involved in a derailment to lose lading is roughly proportional to the number involved. On the other hand, by scattering toxic ladings throughout a train, the probability of one such car being involved in a derailment increases, whereas the probability of all cars being involved decreases. Current accident information is not sufficiently complete to fully assess these off-setting factors. However, cost considerations, as we will see later, in all probability remove a general requirement for separated placement of toxic ladings as the avenue of choice.

Flammable ladings, particularly those requiring pressure cars, present a somewhat different problem as presently configured. The sequence of events following a derailment with grouped cars typically is as follows: (1) a single car is punctured, frequently in the head; (2) a fire begins, enveloping adjacent tank cars; (3) violent ruptures of non-punctured cars occur causing severe and widespread damage; (4) violent ruptures will continue to propagate through the entire string of flammable ladings.

Elimination of violent rupture is a desirable goal, even if the frequency of fires without violent rupture were slightly increased. Section 5.3 of Reference 9 develops fire volume and duration information appropriate for a carload of propane. An example of fire magnitude, considering a liquid release of 100 lb/sec and emissivity of 0.1 gives an equivalent spherical diameter of the fire of 145 ft and fire duration approximately 20 minutes. At 100 ft from the fire center the average heat flux by radiation is 9700 Btu/ft^2-hr. Within the fire, the average radiation flux rate is 24,500 Btu/ft^2-hr. From observed accidents and computer simulations (Reference 8), it is apparent that car separations of the order of 10 cars for 40 mph operating speeds, or 5 cars for 20 mph would be required to secure substantial protection from propagation of fire resulting in violent rupture and rocketing for cars with no thermal shielding.

While there would be some similarity in the cost variations between various routings to those described for speed restrictions, the reasons are different. Terminal cost increases would become virtually unavoidable. In a highly efficient operation it is likely that the average terminal costs probably would still double. Using the example case of a hypothetical trip with 128,000 lbs of propane from Houston to St. Louis developed in Section III-F, Economic Sensitivity, we can judge the effect on the price of the lading of doubling terminal costs. Adjusting Tables XIX and XX to reflect doubling of the terminal variable costs, a 2% rise in the destination price of the lading is indicated--

not a great deal. On the other hand, this would be equivalent to the lading price increase resulting from a $7500 increase in tank car price--without consideration of the protection benefit. In other words, substantial sums of money can be spent on thermal protection for tank cars to obviate the need for spacing with flammable lading.

The potential for head punctures in terminal switching operations is significant. Some of the most costly accidents, such as those at East St. Louis and Decatur, Illinois, have occurred in this manner.

Proposed operational changes to reduce the probability of coupler override due to impact have included the prohibition of uncoupling or cutting off cars in motion. This eliminates the practice of "kicking" cars in flat yards, and prevents the normal gravity switching employed in "hump" classification yards.

Without performing a detailed cost analysis, it is reasonable to postulate that the effect on terminal switching costs could be easily of the order of magnitude assumed in the preceding case, i.e., double. Based on this assumption, up to $7500 could be spent on a head shield to provide the equivalent protection provided by the operating restrictions.

Note that any change in operational practice that results in a transportation cost increase must immediately be borne by the carrier. The carrier then has to try to recover this cost by rate increase through regulatory authorities. On the other hand, changes in tank car equipment, such as head shields, place the immediate burden upon the car manufacturer, who may elect to pass this cost on without the encumbrance of a regulatory procedure. The ability of the manufacturer to recover the increased cost may be limited due to competitive pressures, however.

In summary, Calspan's conclusions with respect to changes in operating practices to secure safer transport of hazardous materials by tank car are:

- Reductions in authorized speeds for trains carrying hazardous commodities by tank car would, in the overall picture, reduce the frequency of severe accidents.

- Cost effectiveness of reduction in speed is highly variable with respect to route. Many cases would not show adoption to be cost beneficial. Individual consideration of cases would be required.

- Adoption of a general rule requiring the use of "buffer cars" between hazardous material laden tank cars is not indicated.

- Operational restrictions affecting standard practice in terminal operations will tend to be an expensive option when compared with car design changes to secure the same level of safety.

III. EVALUATION OF DESIGN IMPROVEMENTS

In recent years, with the advance of technology, ever-increasing varieties and quantities of hazardous materials are being shipped throughout the country, many by railroad tank car. With the rising trend of railroad accidents, the potential threat to property and life posed by these hazardous materials has increasingly been converted to real danger. The objective of the design improvements discussed here is to minimize the chance of catastrophe in the transport of hazardous materials by tank car.

Each tank car design change has been considered on the basis of its cost/benefit. That is, proposed solutions to be effective from a cost/benefit standpoint must be capable of reducing losses sufficiently to pay for cost of incorporating the change.

There are several factors which are required to determine the cost benefit ratio of a proposed change. They are:

1. Accident loss data - used to compute the savings generated by a 100 percent effective modification,
2. Estimate of the probable improvement in safety (i.e., effectiveness of the proposed modification in reducing accident losses),
3. Cost of modification,
4. Expected life of modification,
5. Capital return expected by investor.

The RPI/AAR has reported that losses due to mechanical damage were determined from a comprehensive record of 3,853 tank cars damaged in 2,321 accidents during the six-year period 1965 through 1970. Of these, 625 cars lost lading due to mechanical damage resulting in an economic loss estimated by the RPI/AAR to be $23,300,000. During this same period, an additional 228 cars were damaged by exposure to fire, while loaded, resulting in the loss of

$15,432,000. It is to be noted that these two loss figures are not additive (i.e., some lost lading was originally due to mechanical damage which subsequently caught fire and caused additional damage. In such a case the total loss would be assigned to the mechanical damage category and the fire damage would also be assigned to the fire damage category). Care has to be taken in assigning losses to the correct category so that accurate loss figures are generated. Table II is a summary of the losses due to mechanical and thermal damage presented according to the class of car as taken from RPI/AAR data. (We shall reevaluate some of these estimates later in this report but the values given in Table II are satisfactory for present purposes.)

Because of the cost of making improvements to tank cars, it is highly unlikely that cost-effective modifications can be made to any class of car which has resulted in only small losses/car/year. As seen in Table II only 112A/114A tank cars have been responsible for significant losses. Therefore, in the work that follows, only modifications to this type of car will be considered as being potentially cost beneficial.

If one dollar is lost per car per year, the adoption of a 100 percent effective design modification to stop that loss will essentially save one dollar/car/year. If the modification has a life of 30 years, all other factors remaining constant, in 30 years, the modification is expected to save $30. Since it is expected that the modification will generate $30 over its lifetime, the expenditure of $30 today to install the modification would at first be expected to be justified, and the cost-benefit ratio would be unity. However, since funds must be committed for 30 years which could be used elsewhere to earn some return, an adjustment in the expected saving is needed. At an interest rate of 10 percent, this adjustment is accomplished by multiplying the loss factor by 9.502 rather than 30 with the result that the amount which can be spent on a 100 percent effective modification is not $30 but $9.502. Further, if it is determined that the actual effectiveness of the real modification is 50 percent not 100 percent, then the justifiable amount which can be spent on the modification is only $4.751. A real modification, 50 percent effective in stopping the loss of $1/car/year can cost no more than $4.751/car to

Table II

SUMMARY OF LOSSES DUE TO MECHANICAL DAMAGE*

	Riveted	Steel, Non-Press. Non-Ins.	Steel, Non-Press. Ins.	Press. Ins. 105A	Pressure, Non-Ins. 112A, 114A	Other
Total Loss, 1965-1970	$945,770	$2,662,261	$625,330	$961,681	$15,937,749	$195,502
Avg. No. Cars	52,500	33,604	27,126	33,000	12,000	6,500
Cost/Year/Car	3.00	13.20	3.84	4.85	221.30	5.01

SUMMARY OF LOSSES* DUE TO FIRE EXPOSURE

	Riveted	Steel, Non-Press. Non-Ins.	Steel, Non-Press. Ins.	Press. Ins. 105A	Pressure, Non-Ins. 112A, 114A	Other
Total Loss, 1965-1970	$108,000	$1,418,000	$296,000	$1,419,000	$11,879,000	$312,000
Avg. No. Cars	52,500	33,604	27,126	33,000	12,000	6,500
Cost/Year/Car	0.34	7.03	1.82	7.17	164.98	8.00

*Data taken from RPI/AAR Report RA-02-2-18, Reference 1.

install. If it costs more, it is not cost beneficial.

The RPI/AAR has estimated the benefit of making 100 percent effective modifications to 112A/114A tank cars. These benefits are in terms of the allowable initial cost of installing the modification. These costs are shown in Table III. Here again, we shall adjust the figures of Table III in subsequent sections of this report, particularly in regard to head and shell modifications, but the Table is sufficient to indicate that only head, shell, coupler, and thermal protection modifications need be considered as having real potential for being cost beneficial.

An extensive review of the ongoing tank car research has been completed in this work. The majority of the work has centered on study of the Railroad Tank Car Safety Research and Test Project of RPI/AAR, which has acted as the focal point of tank car research for the past five years although the Federal Railroad Administration has also carried out research, often in conjunction with RPI/AAR. Individual manufacturers have carried out some significant work and this has been reviewed in conjunction with the similar work performed by RPI/AAR. The majority of the research carried on by manufacturers is, however, directed at improvements in fabrication and maintenance of existing types of cars.

The RPI/AAR research has addressed itself to the specific task of reducing the losses associated with tank car accidents. A comprehensive collection and analysis of accident statistics was produced, followed by a review of the tank elements which were most often related to tank failure. Tank heads and shells, including the material from which they were manufactured, relief valves, couplers, and thermal insulations were the components which were investigated. Unfortunately, not all the final reports covering this research have been issued by RPI/AAR. In particular, RPI/AAR's work on tank steels, thermal effects, and valve functioning, have yet to be reported.

The following sections of this report evaluate design improvements

Table III

ALLOWABLE INITIAL COST FOR 100% EFFECTIVE MODIFICATIONS TO 112A/114A CARS*

COST OF CAPITAL: 10%
ASSUMED LIFE: 30 YEARS

Type of Modification	Maximum Allowable Cost
Head	$ 527.47
Shell	1136.15
Attachment	5.39
Bottom Fittings	4.32
Top Fittings	28.00
Coupler, 10-year life	346.53
Thermal Protection	
30-Year Life	2546-2717
10-Year Life	1694-1808

*From Reference 1

of tank cars related to safety. The general areas consist of accident statistics, head shields, modified couplers, thermal shields, tank construction materials, and modified safety relief valves.

A. Accident Statistics

The RPI/AAR, in its effort to determine the costs associated with tank car accidents, compiled a comprehensive record of 3,853 cars damaged in 2,321 accidents. Of the damaged cars, 625 lost lading which resulted in $23,000,000 of additional damage. This damage assessment was broken down into two categories: (1) cost of lost lading, and (2) losses caused by the lost lading.

1. Re-evaluation of Losses

Calspan has reviewed the RPI/AAR data and, in particular, looked in detail at the losses in five accidents chosen to be representative of a range of dollar losses per accident. For this reassessment of losses, a number of information sources were utilized. Individual accident files of the RPI/AAR group, the National Fire Protection Association, Railroad Transportation Insurers, eye witnesses, city officials, and an attorney were among the sources utilized. This review found that in some instances the RPI/AAR estimates omitted some of the actual losses. In other instances, more than direct damage were included by Calspan such as cost of evacuation, manhours expended by public safety personnel, and loss of earnings resulting from temporary evacuation of businesses. Additional information was available at the time of the Calspan re-evaluation as a result of actual litigation settlements rather than projected settlements. The results of the re-evaluation are presented in Table IV. A complete explanation of what was included in these data is given in Appendix B. In general, the RPI/AAR estimates are lower than the Calspan estimates, primarily because Calspan included more than just direct damage.

Table IV

RESULTS OF ACCIDENT RE-EVALUATION

	AAR	Calspan	% Increase
New Athens, Ill. - 4/9/70	$ 84,000	$ 128,000	52
Armitige, Ohio - 4/25/70	4,800	11,000	131
Crescent City, Ill. - 6/21/70	1,900,000	2,200,000	15
South Byron, N. Y. - 8/27/70	119,000	146,000	22
Crete, Nebraska - 2/18/69	2,000,000	2,000,000	0

The arithmetic average of the increased costs presented in Table IV, is 44 percent. However, Calspan is of the opinion that increasing all costs by 44 percent based on such a small sample would be unrealistic. Our own investigator was on the site at both Crescent City, Illinois, and South Byron, New York, and these data are believed to be the most accurate. We believe that a reasonable estimate of the increased costs is 25 percent. Thus the accidents' costs were adjusted accordingly when the cost/benefit analyses were recomputed.

2. Update of Losses

The RPI/AAR loss data was obtained by examining data on accidents for the years 1965 through 1970. This data is the most extensive available at the present time. As more recent data becomes available, it should be utilized in the analyses; however, obtaining the necessary data is beyond the scope of this work. RPI/AAR is planning to compile losses for more recent years. In the present report the losses for the period 1965-1970 will be updated where necessary to present dollars to account for changing values of damaged items. Two different time bases have been used in the present report for the value of money. If cost of a modification and loss data are available as of a given year, dollar values for that year will be used to give a consistent representation of cost/benefit. If losses are available only for a certain time period and the cost of a modification has not been determined, the losses will be updated to present dollars to provide information on economically justifiable cost of a modification in terms of today's dollars.

The dollar losses in the RPI/AAR report are composed of the cost of lading and other losses caused by the loss of this lading including fire damage to equipment, real property, and loss of life. Since the time period of the report, the values of many of the damaged items have increased substantially. For example, lost propane lading was priced at 6¢/gal. for the RPI/AAR report but present source prices are 7.9¢/gal. for the 2/3 of the supply produced from natural gas and therefore regulated and this is expected to go to 11-13¢/gal shortly.[10] The wholesale price index has increased by 67 percent since 1967 (U.S. Department of Labor). It it is necessary in the analyses to update the loss values, values of ladings will be adjusted to their present worth and other losses will be evaluated on the basis of the change in the wholesale price index.

In the time period of the RPI/AAR evaluation of losses (1965-1970), two accidents accounted for a majority of the total losses involving 112A/114A tank cars. Laurel, Miss., 1/25/69, resulted in $7,800,000 in losses and Crescent City, Ill., 6/21/70, resulted in $1,900,000 in losses. Are accidents of this size likely during a six-year time period? Since 1970 several large accidents of 112A/114A cars have occurred. On October 19, 1971, in Houston, Texas, two tank cars were punctured and the subsequent fire caused 112A/114A tank cars to rupture.[11] No detailed estimate of the amount of damage was presented in the National Transportation Safety (NTSB) report but one fireman was killed, 50 people were injured including 20 hospitalized, 2 cars were destroyed, 14 cars were extensively damaged and six others lightly damaged. Also destroyed were a residence, a fire truck, an automobile, and a railroad motor truck. Several buildings incurred such damage as paint blisters or broken windows.

On January 22, 1972, an accident involving 112A tank cars loaded with propylene in East St. Louis, Illinois, resulted in property damage of more than $7 1/2 million and 223 people were injured enough to require medical treatment including 19 who were hospitalized.[12] A report has also been issued for an accident at Oneonta, N.Y., on February 12, 1974, in which four tank cars ruptured after exposure to fire.[13] In this accident 54 people were injured and substantial damage to residences was sustained at distances up to one mile. No damage estimate was given but based on the number of injured the losses were substantial.

On July 15, 1973, at Kingman, Arizona, a car ruptured after a fire erupted on a siding. Thirteen people died as a result and 95 were injured, many very severely. Extensive property damage occurred in the surrounding area. Property damage has been estimated to be $1,000,000 (Fire Journal, January 1974). No value of total losses can be made because litigation is still is process but considering the number of deaths and injuries, the total is expected to be more than $10 million. This accident is not being investigated by the NTSB as it is not considered to be a transportation accident because the car was parked in a private siding. However, the losses were a result of fire exposure of a 112A/114A tank car and these losses must be assigned to the 112A/114A category. It is not known whether any non-transportation accidents were omitted from the RPI/AAR data.

Other recent large accidents involving 112A/114A cars include, Decatur, Illinois, July 19, 1974, Houston, Texas, September 21, 1974, and La Mirada, California, October 31, 1974. Detailed reports have not been issued on these accidents so that conclusions on the amounts of losses must be tentative but the accidents indicate that there continues to be large dollar losses. At Decatur, Illinois, damage estimates are $14 million and two people were killed and 6 of 140 injured were in critical condition (Decatur Review, July 21, 1974). (Four additional people later died.) In the second Houston accident there was one death, 190 railcars destroyed, 240 cars heavily damaged, and several residences and businesses damaged. Total damage was estimated by the railroad to be $12-$14 million of which $4 million was damage incurred by the railroad (Railway Age, October 14, 1974). At La Mirada, California, an LPG tank car ruptured after fire exposure. There were no injuries but a railroad spokesman estimated damage at $1 million (Los Angeles Times, November 2, 1974).

The accidents mentioned above are a sampling of large accidents since 1970. They indicate a continuing problem of accidents involving losses of millions of dollars. Hence, the years 1965-1970 do not represent an overly severe loss period. There have been accidents since 1970 in which losses were even greater than the Laurel, Mississippi, accident which was by far the largest in the years 1965-1970. We shall use the losses from the RPI/AAR report as representative of the expected losses in future years.

B. Head Shields

A number of tanks have been punctured during derailments or other accidents resulting in substantial dollar losses and casualties in large part due to fires of spilled lading. Tank punctures are caused by striking couplers, trucks, and other objects. Several methods have been proposed for decreasing the likelihood of tank punctures. Among these are shields covering a portion or all of the tank heads primarily to prevent couplers from preceding or following cars from puncturing the tank during an accident. This section deals with a cost/benefit analysis of head shields for new and existing 112A/114A series tank cars. A cost/benefit analysis is composed of three key factors, namely:

1. The magnitude of expected dollar losses.
2. The cost per car of implementing a proposed modification.
3. The effectiveness of the modification in reducing dollar losses.

The amount of expected losses can be estimated from statistical review of historical data on losses. The cost of implementing a proposed modification can be determined from engineering estimates of costs. The effectiveness of the modification can be determined from analysis combined with available experimental test data. The term effectiveness as expressed here is a dimensionless factor determined by dividing expected overall reduction in losses with modified cars by losses anticipated with unmodified cars. The reduction in losses by adoption of the modification results from a reduced frequency of occurrence of head puncture. Reduction in the magnitude of loss for a given accident for which a puncture occurs is not implied.

The amount of expected losses with unmodified cars multiplied by the effectiveness determines the reduction in losses, i.e., savings, that can be expected. These savings can be utilized to pay for the modification plus interest over a number of years. The amount at 100 percent effectiveness that could be paid back, including interest, from the expected savings is termed present value. Any reduction in effectiveness of the modification reduces the present value proportionately. The economic benefit is the present value minus the cost of the modification. If the economic benefit is positive, it is then economically justifiable to make the modification.

The Railway Progress Institute (RPI) and the Association of American Railroads (AAR) in a cooperative research program have already investigated head shields and the losses occurring in tank car accidents.[14,15] The RPI/AAR reports list all tank cars known to have lost lading due to mechanical damage incurred in accidents during the period from 1965 to 1970. Incidents of loss are sorted by class of tank car and cause of loss. Loss figures are composed of two parts: (1) cost of lost lading and (2) other losses caused by the loss of this lading, including fire damage to equipment, real property, and loss of life. The RPI/AAR has reported the accident loss data due to punctures of 112A/114A tank cars. A review is given in Table V.

Table V

LOSSES DUE TO PUNCTURES OF 112A/114A TANK CARS*

	Cause	
	Head Puncture	Shell Puncture
Losses, $	3,997,633	8,610,791
No. of Cases	40	9
No. of Years	6	6
Avg. No. of Cars in Service	12,000	12,000
Losses, $/Car/Year	55.52	119.59
Total Losses - $12,608,424		

*Data taken from Ref. 1, p.

4. Distribution of Losses

The RPI/AAR, in their review of tank car accidents, determined what the immediate cause of a fire was and then assigned accident dollar losses according to the tank element which failed. For instance, if during an accident, a tank head was punctured with a resulting loss of lading and/or fire, the damage caused by the lost lading was assigned to the category of head puncture. Similarly, if a tank shell was punctured, causing damage, these were assigned to the category of shell puncture. "Shell" is considered to include only the cylindrical portion of the tank and "head" only the ends of the tank. In this way, the losses were assigned to a particular tank element and an estimate was made of the potential savings which could be realized if the frequency of occurrence for that type of failure could be reduced. The RPI/AAR results for all head and shell puncture accidents involving tank cars have been summarized in Table V.

Historically, there were 40 head punctures and 9 shell punctures which caused damage during the six-year period of 1965-1970. Intuitively, this is the type of distribution which would be expected. The tank head is exposed to the coupler of the adjoining car during the early phase of a derailment when the cars are still relatively well in line. During this period, the high compressive forces existing between cars, in conjunction with the vertical motion between cars, allows the coupler of an adjoining car to contact the tank head.

Later in the derailment sequence, once the cars are no longer in line, contact between cars can occur, but there is a substantially smaller chance of a concentrated force being applied to the shell. Coupler-shell and track-shell contacts occur, but so do the more acceptable shell-shell contacts. As a result, the distribution of punctures presented in Table V is as expected. The distribution of dollar losses presented in Table V is not, however, consistent with the puncture data.

The historical data imply that only 18 percent of the punctures are responsible for 68 percent of the dollar losses, while the other 82 percent of the punctures are responsible for only 32 percent of the losses. Calspan is of the opinion that simply relying on the relatively small amount of historical data introduced a fallacy into the RPI/AAR cost/benefit analysis. If enough tank car accidents were investigated over a long period of time, the loss distribution should match the tank ~~shell~~ puncture distribution. Shell punctures do not inherently produce more costly losses than head punctures. Since head punctures occur five times as often as shell punctures, in the long run, dollar losses due to head punctures should approach five times the losses due to shell punctures. In fact, it might be expected that head punctures would be more costly than shell punctures because head punctures may tend to occur more frequently during yard accidents near heavily populated areas. Accident data since 1970 have tended to agree with the revised distribution, i.e., the East St. Louis, Illinois, accident on January 22, 1972, which was caused by a head puncture and resulted in $7 1/2 million property damage plus 19 people injured enough to be hospitalized.[12] This accident resulted in dollar losses of the same magnitude as the total of all the previous accidents involving either head punctures or shell punctures. With such large losses from a single accident, the data can be incorrectly distorted towards greater losses from either head or shell punctures as the result of a single accident.

In support of the argument that losses should be proportional to the frequency of puncture occurrence, the puncture data for all classes of cars are summarized in Table VI. There are two entries for the 112A type of car. The first includes the losses at Laurel, Mississippi, and Crescent City, Illinois. The second excludes those losses. The effect of deleting these two accidents is shown in Table VI to reduce the total losses by 60 percent, indicating that these two accidents have a large distorting effect. In particular, the losses due to shell punctures are decreased 85 percent by eliminating these two accidents. The nature of this distortion can be determined by the last two entries in Table VI. Using all the accidents, the losses are inversely proportional to puncture frequency. When the two accidents are not included, the opposite results occur and accident losses become more directly proportional to puncture frequency, as would be expected.

Table VI

LOSS DUE TO PUNCTURES OF TANK CARS*[1]

	Cause			
	Head Punctures		Shell Punctures	
Car Type	$ Loss	No.	$ Loss	No.
Riveted Steel Cars	293,000	20	425,860	22
103 Non-Insulated	50,000	15	78,700	11
103 Insulated	3,100	1	30,665	3
111A Full Frame Non-Ins.	680,530	15	66,300	4
111A Full Frame Ins.	9,000	3	3,400	1
111A Stub Sill Non-Ins.	841,650	29	158,500	10
111A Stub Sill Ins.	292,550	12	30,000	3
105A Insulated	403,000	8	110,840	4
112A	3,918,000	40	8,439,265	9
112A Minus Laurel, Miss. and Crescent City, Ill.	2,356,000	38	297,265	7
113	50,240	5	34,060	8
Total	6,541,070	148	9,674,855	75
Total Minus Laurel, Miss. and Crescent City, Ill. Data	4,979,070	146	1,532,855	73

*Values are slightly different from those in Table V because in Table V losses due to unknown causes were proportionately distributed among the known causes.

Calspan has recomputed the cost/benefit analysis for head shields, applied to 112A/114A cars, using the statistically correct losses. Hence, head punctures were assigned 82 percent of the losses due to punctures, while shell punctures were assigned 18 percent. The redistributed losses are shown in Table VII using the same format as in Table V.

Applying the correct loss distribution, the cost benefit of head shields is recomputed as shown in Table VIII. Note that the cost of capital,

Table VII

REDISTRIBUTED LOSSES DUE TO PUNCTURES OF 112A/114A TANK CARS

	Cause	
	Head Puncture	Shell Puncture
Losses, $	10,292,591	2,315,833
No. of Cases	40	9
No. of Years	6	6
Avg. No. of Cars in Service	12,000	12,000
Losses, $/Car/Year	142.95	32.16
Total Losses - $12,608,424		

Table VIII

HEAD SHIELD COST/BENEFIT ANALYSIS - REDISTRIBUTED LOSSES

Cost of Capital: 10%
Assumed Life: 30 yrs.
112A/114A Tank Cars

Present Value = Losses/Car/Year x Present Value Factor*	
= $142.95 x 9.5 =	$1358
Shield Effectiveness (RPI/AAR estimate)	50%
Present Value at Stated Effectiveness = Present Value x Effectiveness	
= $1358 x .50 =	$ 679
Cost of Shield (RPI/AAR estimate)	
New Cars	$ 272
Existing Car	$ 474
Economic Benefit = Present Value at Stated Effectiveness	
- Cost of Shield	
New Cars: $679 - $272 =	$ 407
Existing Cars: $679 - $474 =	$ 205

*Present value of a stream of payments of $1/year for 30 years discounted continuously at an annual rate of 10%.

head shield life, and the shield effectiveness values are unchanged from those used by RPI/AAR. Head shields are now found beneficial by +$407 on new cars and by +$205 on existing cars. Therefore, there is a net economic benefit to be derived from installing heat shields on both new and existing 112A/114A tank cars.

The results of this cost/benefit analysis are compared with those of RPI/AAR in Table IX. In the first RPI/AAR study, the effectiveness of the head shields in preventing head punctures was determined to be 77 percent. In the second study, the effectiveness was downgraded to 50 percent and the cost of a head shield installation and accident losses were updated. Other than the redistribution of losses, the Calspan data utilize the same data and analytical techniques as the second RPI/AAR report. Analysis similar to that in Table VIII would show that head shields would be cost beneficial at effectiveness as low as 20 percent on new cars and 35 percent on existing cars.

Table IX

HEAD SHIELD COST/BENEFIT STUDIES

STUDY	INSTALLATION	SHIELD EFFECTIVENESS	SHIELD COST	ECONOMIC BENEFIT
RPI/AAR-DOT HEAD SHIELD STUDY, AUG. 71, REF. 12	NEW 112A/114A's	77%	$280	+$105
	EXISTING 112A/114A's	77%	$335	+$50
RPI/AAR REPT. RA-00-1-22, OCT. 72, REF. 1	NEW 112A/114A's	50%	$272	-$8
	EXISTING 112A/114A's	50%	$474	-$210
CALSPAN DISTRIBUTION OF LOSSES	NEW 112A/114A's	50%	$272	+$407
	EXISTING 112A/114A's	50%	$474	+$205

2. Amount of Losses

From the above, it is shown that head shields are cost beneficial for new and existing 112A/114A cars considering only proper distribution of losses reported by RPI/AAR. As a secondary effect, in addition to redistribution of losses, Calspan has also re-evaluated the amount of losses. (See the section of this report: Accident Statistic Research, Re-evaluation of Losses.) It was found that the RPI/AAR loss figures should be increased by about 25 percent. An increase of 25 percent in the accident losses increases the present value of losses by the same percentage. Table X is a restating of Table VIII utilizing a 25 percent increase in losses. The economic benefit is found to increase to $577 for new cars and to $375 for existing cars.

Table X

HEAD SHIELD COST/BENEFIT ANALYSIS - REDISTRIBUTED LOSSES AND 25 PERCENT INCREASE IN LOSSES

Cost of Capital: 10%
Assumed Life: 30 yrs.
112A/114A Tank Cars

Present Value:	142.95 x 1.25 x 9.5	$1698
Shield Effectiveness		50%
Present Value At Stated Effectiveness		$ 849
Cost of Shield		
	New Cars	$ 272
	Existing Cars	$ 474
Economic Benefit		
	New Cars	$ 577
	Existing Cars	$ 375

All of the above dollar values are in terms of dollars at the time considered in the RPI/AAR reports, i.e., 1965-1970. In terms of present dollars, both present value and the cost of shield would be higher than listed. It is beyond the scope of this work to convert these amounts to present dollars but undoubtedly the economic benefit would be about the same or even greater if this was done.

3. Cost of Capital

In the calculations by RPI/AAR of economic benefit of potential design changes, such as a thermal shield, a stream of payments was converted to a present sum by means of conventional interest formulas. The interest factor used was 10 percent. The use of 10 percent for capital recovery and earnings can be considered conservative. The question is, however, whether capital recovery should be allowed at all for correction of a design defect affecting safety.

There is a very strong precedent for no capital recovery allowance with respect to safety defects. It should be noted that automobile manufacturers have absorbed the total cost of the vast majority of recall campaigns for the correction of safety-related items. Other examples of instances where strict cost/benefit analyses have not been adhered to can also be cited. Nursing homes are subject to strict fire prevention safety measures. A cost/benefit analysis would reveal that the cost of safety items exceeds the reduction of losses. Because of the age of the victims, considerations of such things as potential future earnings result in no change in the conclusion that improvements are not cost effective. The response to the nursing home fire problem, on the other hand, has been one of increasingly stringent design requirements. A principal driving force behind these requirements has been the desire to prevent injury and death, with consideration beyond simple dollar balancing. Similarly, in the transportation industry, e.g., airline and pipeline, both voluntary and mandatory standards have not been derived from equalized cost of design versus loss data. For one thing, historical loss data are frequently unavailable or, in the case of new design, not applicable.

Cost/benefit studies are a very useful tool. However, with regard to safety considerations, they should extend beyond derivation of a balance point between cost of improvement and loss reduction. As a minimum, assessment of the impact of adoption of an improvement on the viability of the service should be considered. It is to this point that the sensitivity of transportation cost to car initial cost applies. The RPI/AAR study did not address this point at all. In essence, they looked only at a lower bound of a "permissible" expense based on current economics and did not include a look at an upper bound, i.e., the best design consistent with the viability of the service. The National Transportation Safety Board (NTSB) commented on the problems of implementing design changes following the Crete, Nebraska, incident.[16] To paraphrase this NTSB report: Changes to existing cars required because of faulty initial design should be considered as corrections of an overlooked matter rather than being considered as costly and profit reducing and therefore as questionable improvements.

As an indication of the effect of considering a reduced cost of capital, one computes the total savings attributed to a 50 percent effective shield to be $142.95 x 1.25 x 30 x 0.5 = $2680, assuming 0 percent return and 30-year shield life. Hence, a shield costing up to $2680 could be installed on either new or existing cars with return sufficient to pay for the shield over a 30-year period. That is, the reduction in losses minus the cost of the shield would be $2408 for new cars and $2134 for existing cars. The possible amount of investment in shield construction might be expected to produce a head shield having greater than 50 percent effectiveness and, thus, might actually provide some return on investment. The maximum amount which can be invested in a head shield with 30-year life and no interest return is, however, limited to $5360 even if the shield were 100 percent effective in preventing head punctures.

4. Review of Modeling and Test Work Done in Support of Head Shield Designs

The head shield work completed by RPI/AAR was presented in two parts. The original tank car research program included a task for investigating tank head punctures. This work was funded by DOT and resulted in a report, Reference 14, which treats the design of a prototype head shield. Following the completion of the DOT head shield study, RPI/AAR continued work on a test program that was to include approximately 20 full-scale tests and 74 1/5-scale model tests. This later work was reported in Reference 17. However, Reference 17 should be considered an appendix to the original DOT report inasmuch as all fundamental decisions that affected the conclusions of the head shield were developed and reported in Reference 14.

To accomplish the objective of the head shield study, the RPI/AAR established six tasks which are listed here for reference.

Task 1 - Identification of Tank Head Failure Characteristics
Task 2 - Establish Design Criteria
Task 3 - Establish Design Specification
Task 4 - Cost Analysis
Task 5 - Performance of Test Specifications
Task 6 - Define Prototype Research Program

Calspan has the following comments on the execution of the testing program. In view of the great cost of the test program undertaken, it seems surprising that so little apparent use was made of analytical techniques which are available for the direction of test programs. For instance, if at the outset of the RPI/AAR's Task 2 analysis, the factors controlling head failures were assumed to be:

1. Head properties: thickness, geometry, material
2. Commodity: outage, internal pressure, commodity weight
3. Impact characteristics: force and duration, impact velocity, location, and orientation
4. Tank car design and attachment construction details

one could then assume the existence of a relation of form:

$$\phi\ (F,t,P,M_1,M_2,V_1) = 0$$

where: t = head thickness

D = tank diameter

P = tank pressure

M_1 = mass of striking car

M_2 = mass of struck car

V_1 = velocity of striking car

F = contact force

A dimensional analysis could then have been performed to obtain a set of dimensionless groups such that equation (1) could be written as:

$$F = \sqrt[3]{(M_1V_1)^2P}\;\Phi\left[\frac{M_2}{M_1},\; t\;\sqrt[3]{P/M_1V_1^2}\right] \tag{2}$$

It is important to note that the functional relation, Φ, is unknown. Experiments could be performed, guided by the dimensionless groups, to determine the unknown function, Φ. However, in spite of the 74 1/5-scale tests that were run, no attempt was reported of using them in this systematic fashion.

The analysis that was reported in References 14 and 17 was an attempt to match a single curve to all the experimental data. The final result of this effort as reported in Reference 17 is:

$$F = .00383\;(\lambda)^{3/32}\;(W_1V_1)^{1.5}\eta \tag{3}$$

where: F = coupler force, 1000's lb

$\lambda = W_2/W_1$

W_1 = weight of striking car, 1000's lb

W_2 = weight of struck car, 1000's lb

V_1 = velocity of striking car, mph

η = pressure parameter

The reason that momentum appears in Equation (3) rather than the naturally arising kinetic energy of Equation (2) is due to an a priori assumption whereby energy was ignored. This assumption was made in spite of the fact that most penetration studies, such as those reported in References 18 and 19, indicate that indentation is work-energy dependent. Regardless, the value of an analysis lies in its ability to predict realistic effects. The value of Equation (3) can be judged from Table XI, which gives a comparison between the measured and predicted coupler force for the full scale 112A340W car tests, Reference 17.

Table XI

TEST RESULTS - 112A340W CARS

Test No.	Measured Force	Force Predicted from Equation 3	% Error
1	504,000 lb	1,244,000	146
2	675,000 lb	1,986,000	194

One may observe poor agreement between measured and predicted forces. One finds closer correlation (±10%) when Equation (3) is used to predict coupler force for a second series of tests run with riveted cars. This is understandable since the vast majority of data points used in RPI/AAR analysis were obtained using riveted cars. Further, the second series of tests were run at approximately constant impact momentum while the impact momentum of 112A340W cars (Table XI) was approximately three times larger. In the opinion of Calspan, the analysis carried out under Task 2.3 of Reference 14 is unsatisfactory and any conclusions based on these results should be questioned.

C. Modified Couplers

The RPI/AAR study has found that couplers caused 26 out of the total of 40 head punctures during 1965-1970.[1] It was also determined that 4 punctures were caused by other than couplers and in 10 cases the puncture mechanism was not known. No shell punctures were found to have been caused by couplers. In view of the large dollar losses that have resulted from head punctures (see Table VII), modification of couplers to prevent head punctures is a subject worthy of study.

Coupler research directed specifically at the problem of reducing disengagement during derailment has been ongoing in the AAR's Committee on Couplers and Draft Gear. Both type E and type F couplers have been used on tank cars. E couplers do not have provision for prevention of vertical disengagements when coupled with E couplers. Presently type F couplers are required for all new cars (49 CFR 179.14). Mated type F couplers provide a measure of vertical disengagement protection. Type F couplers engaged with type E couplers (the most common type on general freight equipment) inhibit downward disengagement only. Modified E and F couplers have also been proposed. These modified couplers include shelves on the top and bottom of the standard couplers intended to inhibit all vertical disengagement.

The safety shelf is not new but it has, for the first time, received some analytical consideration. American Steel Foundries published an investigation (Reference 20) in which the modified E coupler was subjected to vertical loads to determine the strength of the shelves. In addition, the dimensional parameters were investigated to determine, for the F coupler with the top shelf and E coupler with top and bottom shelf, if modified couplers will remain coupled to standard couplers under all AAR allowable conditions. Their conclusion, as presented in Reference 20, is that there is no guarantee that disengagement can be prevented.

The AAR conducted laboratory tests on the modified E and F couplers and reported the results in Reference 21. However, no buff forces were simulated while the vertical and horizontal investigations were being conducted. A real understanding of the action of a modified coupler during derailment has not yet been accomplished.

A cost benefit analysis was performed by the RPI/AAR for the application of the modified E and F couplers to 112A cars equipped with standard E couplers.[15] (All tank cars built after January 1, 1971, have been required to have standard F couplers, but previous to that time tank cars were built with standard E couplers.) Their conclusion was that modified E couplers on 112A/ 11rA tank cars would be cost beneficial compared with standard E couplers and that modified F couplers would not be cost beneficial compared with standard E couplers. Comparison was not made with standard F couplers because these were judged by RPI/AAR not to have any increased effectiveness even though they cost more than standard E couplers. We shall reexamine the cost benefit analysis of couplers along the guidelines used for the analysis of head shields. We shall also compare shelf couplers with standard F couplers.

1. Distribution of Losses

As in the consideration of head shields, the RPI/AAR assigned dollar losses according to the tank element that failed, head or shell. The total losses due to punctures should have been apportioned between the number of head and shell punctures. See the section: Head Shields, Distribution of Losses for a further discussion of the Calspan distribution of losses.

Table VII presented the redistributed losses due to head punctures. Using the RPI/AAR data on the number of head punctures that were a result of coupler strikes (26) compared with punctures from other sources (4), the losses due to couplers can be apportioned. The results are shown in Table XII. Modified couplers may be compared either with standard E of F couplers. RPI/AAR chose to compare modified couplers with standard E couplers because of their contention that standard F couplers have no advantage over standard E couplers yet cost more. However, standard F couplers are now required in all new car construction for the purpose of reducing punctures and jackknifing. This would indicate that modified couplers should be compared against standard F couplers. We shall present comparisons of modified couplers against both standard E and F couplers. The cost benefits are computed in Table XIII. RPI/AAR estimates are used for all terms except the amount of losses due to redistribution of losses. All dollars, both losses and costs of modifications, are in terms of 1965-1970 dollars. There is an economic benefit for all of the comparisons.

Table XII

REDISTRIBUTED LOSSES DUE TO COUPLER PUNCTURES OF 112A/114A TANK CARS

Losses, $	8,920,245
No. of Cases	26
No. of Years	6
Avg. No. of Cars in Service	12,000
Losses, $/Car/Year	124

Table XIII

MODIFIED COUPLER COST/BENEFIT ANALYSIS - REDISTRIBUTED LOSSES

Cost of Capital: 10%
Assumed Life: 10 yrs.
112A/114A Tank Cars

Present Value = Losses/Car/Year x Present Value Factor	
= $124 x 6.32 =	$784
Modified Coupler Effectiveness (RPI/AAR Estimate)	60%
Present Value at Stated Effectiveness = Present Value x Efficiency	
= $784 x .60 =	$470
Differential Cost of Modified E Coupler Compared With Standard E Coupler (RPI/AAR Estimate)	$ 97
Economic Benefit = Present Value at Stated Effectiveness - Differential Cost of Modified E Coupler	
= $470 - $97 =	$373

Table XIII (Cont'd.)

Differential Cost of Modified E Coupler Compared with Standard F Coupler (RPI/AAR Estimate)	-$285
Economic Benefit = Present Value at Stated Effectiveness - Differential Cost of Modified E Coupler = $470 - (-$285) =	$755
Differential Cost of Modified F Coupler Compared with Standard E Coupler (RPI/AAR Estimate)	$424
Economic Benefit = Present Value at Stated Effectiveness - Differential Cost of Modified F Coupler = $470 - $424 =	$28
Differential Cost of Modified F Coupler Compared with Standard F Coupler (RPI/AAR Estimate)	$42
Economic Benefit - Present Value at Stated Effectiveness - Differential Cost of Modified F Coupler = $470 - $42 =	$428

The calculations are for a modified coupler being installed instead of a standard coupler on new cars at the time of regularly scheduled installation of a new coupler. If a car was taken out of service before regularly scheduled and a standard coupler replaced by a modified coupler, the cost would be more than shown. However, it is only considered here that as couplers are replaced according to their normal attrition they would be replaced by modified couplers.

2. Amount of Losses

In the preceding, the loss data were taken from RPI/AAR reports. However, as discussed in the section: Accident Statistics, Re-evaluation of Losses, Calspan has determined that the RPI/AAR loss data should be increased by about 25 percent. Table XIV is a restating of Table XIII utilizing a 25 percent increase in losses. The economic benefit is found to increase to +$491.

Table XIV

MODIFIED COUPLER COST/BENEFIT ANALYSIS - REDISTRIBUTED LOSSES AND 25 PERCENT INCREASE IN LOSSES

Cost of Capital: 10%
Assumed Life: 10 yrs.
112A/114A Tank Cars

Present Value: $124 x 1.25 x 6.32 =	$980
Modified Coupler Effectiveness	60%
Present Value at Stated Effectiveness	$588
Differential Cost of Modified E Coupler Compared with Standard E Coupler	$ 97
Economic Benefit	$491

Table XIV (Cont'd.)

Differential Cost of Modified E Coupler Compared with Standard F Coupler	$285
Economic Benefit	$863
Differential Cost of Modified F Coupler Compared with Standard E Coupler	$424
Economic Benefit	$164
Differential Cost of Modified F Coupler Compared with Standard F Coupler	$ 42
Economic Benefit	$546

3. Cost of Capital

In the section Head Shields, Cost of Capital, the possibility of not including an interest factor in the calculation was discussed. At zero interest rate the net savings of a 60% effective modified E coupler compared to a standard E coupler would be ($124 x 1.25 x 10 x 0.6) - $97 = $833 under the same assumptions as Table XIV except for interest rate. Following the same order of comparisons given in Table XIV, the net savings would be $1215, $506, and $888, respectively.

Calspan would conclude that converting to modified couplers would be cost beneficial. However, the present knowledge regarding the mechanics of coupler interaction is still inadequate. More work is required, particularly well documented full scale tests.

D. Thermal Shields

The RPI/AAR have investigated thermal shields and the losses occurring in tank car accidents due to fires.[1,22,23] The RPI/AAR reports list all loaded tank cars known to have been exposed to fire during the years 1965-1970. Some data were also published for fires outside of this time period but primary emphasis was on these years. Incidents of loss are sorted by class of tank car. Loss figures are composed of two parts: (1) cost of lost lading and (2) other losses caused by the loss of this lading, including fire damage to equipment, real property, and loss of life. A review of the RPI/AAR loss data for 112A/114A tank cars exposed to fires is given in Table XV.

Table XV

112A/114A TANK CARS EXPOSED TO FIRE - 1965-1970*

Losses, $	11,879,000
No. of Cases	65
Lost All of Lading Due to Fire	56
Ruptured	50
Avg. No. of Cars in Service	12,000
No. of Years	6
Losses, $/Car/Year	165

*Data taken from Ref. 23 p. 7 and 8.

The RPI/AAR has also developed an estimate of the maximum value of a 100 percent effective thermal shield applied to 112A/114A tank cars. The analysis includes the effect of the reduction in costs normally incurred in applying a corrosion protection coating on uninsulated tanks. Because the thermal shield has not been specifically defined, the cost/benefit analysis must be conducted on a somewhat different basis than the cost benefit analyses of head shields and couplers. That is, because the cost and life of the thermal shield are unknown, the analysis can only determine the maximum justifiable cost that could be expended in installation of a thermal shield as a function of the expected life. In this report, we shall discuss the possible effectiveness that can be expected and present an update of losses in terms of present dollars, a re-evaluation of losses, and the effects of cost of capital.

1. Thermal Shield Effectiveness

The effectiveness of a thermal shield is a dimensionless factor determined by dividing expected overall savings with modified cars by losses anticipated with unmodified cars. Effectiveness as defined here is an index of the expected effectiveness of the shields in the aggregate. It is not a measure of the expected effectiveness of an individual shield in a given accident. Higher percentage effectiveness implies higher levels of overall protection. Because the thermal shield is not fully defined, it is not possible to determine an effectiveness. Appendix C presents historical data on 105A insulated tank cars which indicates that they have an insulation effectiveness approaching 100 percent. Following sections of this report include analyses which show the value of typical coatings in reducing heat input to a tank car. These considerations indicate that thermal shield coatings in conjunction with present relief valves sized for uninsulated 112A/114A tank cars can have an efficiency of nearly 100 percent if the coating remains attached to the tank shell during a fire. An efficiency of 100 percent has been used for all calculations in this report.

2. Update of Losses

The RPI/AAR cooperative research program has evaluated losses due to exposure of loaded tank cars to fire by examining data on accidents for the years 1965 through 1970. This data is the most extensive available at the present time. As more recent data becomes available it should be utilized in the analysis; however, obtaining the necessary data is beyond the scope of this work. RPI/AAR is planning to compile losses for more recent years. As this data becomes available, the analysis should be modified. In this report the losses for the period, 1965-1970 will be updated to present dollars to account for changing values of damaged items. A re-evaluation of the losses also will be made based on a more extensive investigation of losses for a few accidents. Some discussion will also be presented of losses since the time period of the RPI/AAR report.

In the section Accident Statistics,Update of Losses, the increase in the value of the ladings since the RPI/AAR reports was discussed along with the increase in other loss factors. To account for these changes in loss values, values of ladings have been adjusted to their present worth and other losses have been evaluated on the basis of the change in wholesale price index. This results in losses in terms of present costs of $19,800,000 compared with the $11,879,000 of Table XV for the period 1965-1970.

In addition, in the section, Accident Statistics, Re-evaluation of Losses, it was found that the RPI/AAR loss data should be increased by about 25 percent based on a re-evaluation of five accidents. The updated losses are given in Table XVI along with the 1965-1970 RPI/AAR data. Presently there are about 20,000 cars in service rather than the average 12,000 cars in 1965-1970. The precise number of cars is not important because per car costs are actually required for the cost benefit analysis.

Table XVI

LOSSES FOR 112A/114A TANK CARS EXPOSED TO FIRE

	RPI/AAR 1965-1970	Updated To Present Dollars	Losses Increased By 25 Percent	Increased No. of Cars
Losses, $	11,879,000	19,800,000	24,750,000	41,250,000
Avg. No. of Cars	12,000	12,000	12,000	20,000
No. of Yrs.	6	6	6	6
Losses, $/Car/Year	165	275	344	344

3. Cost of Capital

In the Section, Head Shields, Cost of Capital, the possibility of not including any interest factor in the calculation was discussed. We shall compute the justifiable costs of thermal shields on the basis of cost of capital of both 10 percent and zero percent.

4. Justifiable Cost of Thermal Shield Coatings

The RPI/AAR[1,23] determined the maximum justifiable cost of applying 100 percent effective thermal shield coatings to entire tank cars by estimating the cost of corrosion protection which the coating would replace and the accident losses that the coating would prevent. These savings can be utilized to pay for the modification plus interest over a number of years. The amount at 100 percent effectiveness that could be paid back, including interest, from expected savings is termed present value. Any reduction in effectiveness of the modification reduces the present value proportionately. The present value represents the economically justifiable cost of using a thermal shield.

RPI/AAR determined that the value of the corrosion protection of a thermal shield was \$121/car/year. (Note: This saving would not be realized for conventional jacketed insulation construction. Otherwise the savings would be similar.) This was determined in 1972. We shall increase this by 20 percent to \$145/car/year to update the savings to present dollars. An upper and a lower bound were put on the accident losses. The lower bound assumes that damage to the car itself (including trucks, brakes, etc.) would not be prevented by a thermal shield. The upper bound assumes that the thermal shield would have prevented all car damage. (Accident loss data have not delineated whether car damage was due to fire or the initial accident which necessitates the upper and lower bounds on losses.) The upper bound was \$165/car/year (Table XV) and the lower bound was \$147/car/year. We shall use these same values updated to present dollars and including an increment to account for the re-evaluation of losses. RPI/AAR used an interest rate of 10 percent in their calculations. We shall use this value and also a zero percent interest rate as discussed in the preceding section.

The results of the calculation of justifiable cost of applying a 100 percent effective thermal shield are shown in Table XVII. All of the updated values in this Table are based on current dollars. No projection has been made in terms of future dollars. Also, Table XVII is based on the assumption that the years 1965-1970 were a normal period for tank car accidents.

Table XVII

JUSTIFIABLE COST OF 100% EFFECTIVE THERMAL SHIELD COATING ON 112A/114A TANK CARS

LIFE OF THERMAL SHIELD, YRS.	RPI/AAR VALUES, 10% INTEREST RATE		UPDATED TO PRESENT DOLLARS				LOSSES INCREASED BY 25%			
			10% INTEREST RATE		0% INTEREST RATE		10% INTEREST RATE		0% INTEREST RATE	
	LOWER BOUND[1]	UPPER BOUND[2]	LOWER BOUND[3]	UPPER BOUND[4]	LOWER BOUND[3]	UPPER BOUND[4]	LOWER BOUND[5]	UPPER BOUND[6]	LOWER BOUND[5]	UPPER BOUND[6]
1	$ 255	$ 272	$ 371	$ 400	$ 390	$ 420	$ 429	$ 466	$ 451	$ 489
5	1053	1124	1533	1651	1950	2100	1772	1922	2255	2445
10	1694	1808	2465	2654	3900	4200	2850	3090	4510	4890
15	2082	2222	3030	3263	5850	6300	3504	3800	6765	7335
30	2546	2717	3705	3990	11,700	12,600	4284	4646	13,530	14,670

	ACCIDENT LOSS SAVINGS	+	CORROSION PROTECTION SAVINGS	=	TOTAL SAVINGS ($/CAR/YR)
1	147	+	121	=	268
2	165	+	121	=	286
3	245	+	145	=	390
4	275	+	145	=	420
5	306	+	145	=	451
6	344	+	145	=	489

LOWER BOUND ASSUMES THAT THERMAL SHIELD DOES NOT PREVENT ANY DAMAGE TO THE TANK CAR

UPPER BOUND ASSUMES THAT THERMAL SHIELD PREVENTS ALL DAMAGE TO CAR AND SUFFERS NONE ITSELF.

Based on the previous sections of this report, the columns headed "Losses Increased by 25 Percent" are believed to more closely represent the actual justifiable cost. Also, the lower bound probably is closest to being correct because it is believed that a thermal shield will not prevent much damage to a car, at least the car will often have to be taken out of service and shopped, which involves considerable expense. In any event, the lower bound provides a conservative estimate of the justifiable cost of a thermal shield coating. Based on the above comments, the justifiable cost of a thermal shield coating has been defined dependent only on the expected life of the shield and the chosen interest rate for cost of capital. For example, a coating with a life of 10 years which might be a desired goal, can be justified if its installed cost were $2850 at an interest rate of 10 percent or $4510 at zero interest rate.

Development of costs of coatings is not within the scope of this work but some discussion of the justifiable costs in terms of per square foot or per gallon of coating is possible. A 33,000 gallon 112A/114A tank car has very nearly 2000 ft^2 of outside surface area. Therefore, the justifiable cost is $\$1.40/ft^2$ to $\$2.30/ft^2$ for a coating with a 10 year life. Also, for this same coating a total of 370 gallons of coating would be required for a 0.3 in. thick coat. This is gallons actually remaining on the tank after cure. Depending on the type of application procedure and evaporation percentage, the actual amount of coating used could be much more. For 370 gallons the justifiable applied cost is $7.70/gal to $12.20/gal.

Conventional jacketed insulation such as found on 105A cars might also be considered for thermal shields. The analysis presented in this section would also be applicable to this type of construction except that the savings due to the lack of additional corrosion protection would not be realized. This type of construction would then only be justified if it were less costly or if the life of the shield were expected to be longer or if the effectiveness were greater. Shield life has been found to be longer compared with the coatings tested to date but final comparisons await further testing. Because the thickness is greater, the jacketed insulation would probably provide greater thermal protection and therefore effectiveness more closely approaching 100 percent. For this configuration, the best estimate of justifiable cost at 100 percent effectiveness would be $2907 for 30 years shield life at 10 percent interest rate. At zero interest the corresponding value would be $9180. Loss of carrying capacity is not considered in this estimate. Note that jacketed insulation need not be applied strictly to 105 car specifications to be effective. Thinner insulations not meeting existing conductance specifications at ambient temperature but maintaining integrity under fire exposure conditions may serve the needs of safety. Hence, a smaller penalty in reduced cargo volume would be incurred if this option were adopted for ladings which are volume limited. Additional information on this point is given in Section II-C, Capacity Limitations for Pressure Cars.

E. Other Modifications

Calspan investigated the use of mechanical shielding devices and metallurgical improvements to determine their effectiveness in increasing the tank's resistance to contact forces. The prevention of rocketing, caused by propagating tank failures, was investigated by increasing the tank wall thickness and changing the shell material properties. To this end, high-grade steels were substituted for TC-128B. These steels have greater elevated temperature tensile properties and better fracture toughness at the lower temperatures. The use of a filament-wound tank concept was considered as an alternative to steel construction. As an aid in stopping propagating type failure, the concept of a discontinuous tank structure was analyzed. Each of the proposed design modifications was evaluated in terms of applications to both new cars and as retrofits to existing cars. Further, for each design concept presented, an estimate of both the cost of implementation and the probable improvement in safety (benefit) derived from its application was determined. Using these figures, a cost-benefit analysis will determine whether the design change can be justified.

An investigation of the effects on payload of increased car weight caused by implementation of possible improved designs has been included in Section II, Design and Operational Factors.

The specific modifications which were given consideration in this section are:

1. Change specification of plate thickness and/or material to resist puncture forces and propagating failures.
2. Discontinuous tank structure.
3. Prevent coupler separation due to failure.

1. Tank Material Specifications

The primary mode of head failure is shear at the peripheral positions of the striking coupler. To be sure, a considerable amount of elastic and plastic buckling occurs prior to ultimate failure, but the amount of resisting force generated by these deformations can be considered small for the purpose of these approximate calculations. Based on the assumption that the shear forces are dominant, the governing equation for computing puncture forces is given by:

$$F = At\tau \qquad (4)$$

where:
F = puncture force
A = perimeter of striking coupler
t = plate thickness, and
τ = ultimate shear stress at failure.

Equation 4 predicts that a doubling of the resisting force can be accomplished by doubling either the plate thickness or the allowable shear stress. Armco Steel Corporation has quoted a price of 11.3¢/lb for the present TC-128 material, while an improved steel, which has twice the allowable shear stress, has been quoted at 16.8¢/lb. On this basis, the present cost of material used in constructing the 112A/114A car (TC-128, 5/8 in. thick, and 2000 ft^2 area) is approximately \$6000. Doubling the thickness would double the cost to \$12,000. On the other hand, doubling the allowable shear stress would increase the material cost to \$9000. Clearly, a change in material specification is preferable to changing plate thickness. However, what benefits can be expected from doubling allowable shear stress by any means. The limiting force for the present tank is approximately 400,000 lb. Doubling this force to 800,000 lb, is still lower than the forces likely to be produced during actual derailments.* From this standpoint, the effectiveness of increased shear strength is not expected to be large.

There are several other possible benefits which occur by adopting a higher grade steel. The RPI/AAR has proposed a number of metallurgical changes to reduce the number of tanks experiencing propagating type failures. Their recommendations include changing of rolling schedules, improved grain size, improved transition temperature limits, etc. Such minor modifications to the TC-128 material will produce only marginal changes in the frequency of propagating type fractures and/or increased resistance to puncture. Upgrading the material specification will allow the use of a steel with better fracture toughness at low ambient temperature ($0^{o}F$) and greater elevated temperature strength. Some benefit can be expected from increasing the low temperature fracture toughness, but the major benefit will be the increase in the elevated temperature strength. This latter benefit will be discussed in the section, Thermal Protection.

A possible alternative to be considered is increasing the strength of the heads only. However, the head shield is indicated to be a more effective solution. This design modification has already been determined to be cost effective as reported in a preceding section.

One concludes that modifications to the tank material specification and increase of plate thickness are not justified in view of their relatively high cost and questionable effectiveness. Further, if applied to the head alone,

*See, for example, selected results of the derailment simulation (Reference 8) where draft forces reached as high as a million pounds.

they are not as effective as a head shield. In addition, these modifications are not applicable as retrofits to existing cars, a consideration of importance since 20,000 112A/114A type cars are currently in service.

2. Filament-Wound Tanks

The possibility of using filament-wound tanks has been investigated in a preliminary fashion. The concept may immediately solve two of the most pressing problems related to the 112A/114A car in LPG service. These problems are catastrophic failure initiated by mechanical damage and retention of strength at high temperature.

The positive attributes of filament-wound tanks have been demonstrated experimentally on small scale tanks. Structural Composites Industries of Azusa, California, has performed a fire test in which a composite tank was subjected to a fire ($Q = 20,000\ Btu/ft^2$-hr) for approximately 1 hour without failure*. The external surfaces charred but retained their integrity. This same company has demonstrated the resistance of filament-wound tanks to impact damage.

The costs of these tanks are prohibitive at approximately 5 to 10 times the cost of a similar steel tank. Further investigation into this concept has not been carried out by Calspan since it is not at present cost beneficial. Further research, at a later time, to reduce the concept to economical practice should not be precluded.

3. Discontinuous Tank Structures

The traditional means of dealing with a propagating fracture has been to introduce a discontinuity into the structure. Some aircraft and missile structures incorporate riveted or bolted joints periodically so that propagating fractures cannot occur throughout the structure. The comparative cost of riveted construction is approximately two times the standard welding construction now in use, primarily because special consideration must be given to insuring that the riveted seam will remain leak free.

*Personal communication.

The primary function of the discontinuity is to arrest a propagating fracture. However, in tank cars, the arrested propagating fracture can lead to the formation of tubs, which, because of the thermodynamic state of the lading, can result in rocketing. Because of this very real possibility, the effectiveness of this solution has to be assigned a low value. This low effectiveness combined with the higher construction costs has resulted in this design modification being judged not cost beneficial.

4. Plastic Coupler Hinge

In a number of accidents, couplers have broken off at their shanks. This failure is often accompanied by a twisting action. In other cases, the coupler does not actually fail, but the sills are spread open by lateral forces. Since modified couplers are designed to resist separation, it is expected that larger loads can be brought to bear on the coupler components during derailment. In the studies of modified couplers, no modifications to accommodate these increased loads have been developed. As a result, it is expected that the number of coupler failures at the shank would increase upon adoption of modified coupler design.

A coupler which has separated due to failure of the shank may cause as much damage to the adjacent car as a coupler which has simply separated. To prevent this type of failure from occurring, the coupler, yoke, draft gear, and draft pocket must be designed as a unit. The incorporation of a weak link into this system will insure that failure can occur in a controlled manner.

An illustration of a weak link incorporated into the draft system is given by Figure 6. It consists of a plastic hinge which is capable of sustaining the 800,000 lb. static crush load and the 1,250,000 lb. dynamic load imposed as per the design manual. At substantially higher load, but still below the loads at which the coupler shank will fail, the hinge will yield plastically, allowing the constrained coupler forces to be relieved without causing a complete separation. The relieving of these forces will reduce even

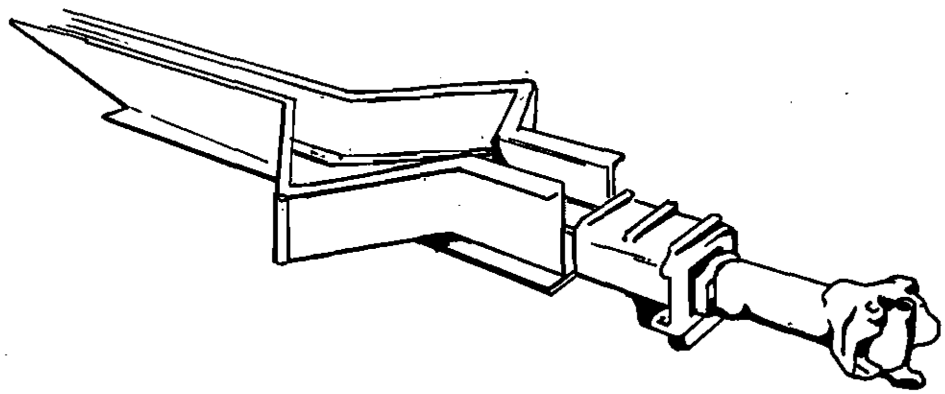


Figure 6 CONTROLLED FAILURE DRAFT SYSTEM

further the probability of head punctures and help keep cars aligned during derailment. A further advantage may be gained by preventing separation of the cars. This is the reduction of shell punctures. In the past if cars separated, lateral restraining force ceased to be applied and the cars tended to "jack knife" thus exposing the shell of one car to the trucks of another. Since shell-truck contacts are one of the most frequent causes of shell intrusion, preventing the separation of cars and allowing constraining forces to resist the turning motions of cars during derailments should reduce the tendency of cars to "jack-knife."

Any modification which can aid in keeping cars in line during derailments will have a strong influence on reducing the costs of tank car accidents. Placing a protective shield around the shell is too expensive to be cost beneficial. Prevention of shell-truck contact by keeping cars in line may be a practical alternative. However, a considerable amount of experimental work will be required to determine the feasibility of this modification and to establish an estimate of the expected benefit. Preliminary estimates of the cost of incorporating this change into new cars are approximately $3000/car.

5. Thermal Protection

As discussed in the section, Thermal Shields, substantial amounts of losses have been incurred due to fire exposure of tank cars. It was shown that losses are $344/car/year and the justifiable cost of thermal shield coatings were determined. Several modifications could be made other than thermal shields to reduce losses due to fires. Four other solutions appear practical enough to warrant serious study. They are:

1. Change tank material specification to resist thermally initiated propagating fractures.
2. Modify valve area to allow more efficient tank venting.
3. Modify valve to be actuated by lading temperature.
4. Insure vapor discharge.

A discussion of each of these modifications is presented in the following.

a. Tank Material Changes

There are two principal ways in which the frequency of thermally initiated propagating type fractures can be reduced. The first is to require a tank material with improved elevated temperature properties and the second is to add insulation to the car. A calculation was made using the Calspan Tank Car Thermal Model in which a standard, uninsulated car was compared with an insulated car and a car made of increased strength steel. The results of this calculation are shown in Figure 7. The conclusion based on this calculation is that the probable increase in safety is greater for the addition of small amounts of insulation than for changing the material specification. The underlying cause for this conclusion is that even an improved steel will not retain a significant amount of strength above 1200°F. This fact is well illustrated by Figure 8 in which the burst pressure of the tank has been plotted against wall temperature. The data beyond 1200°F has been entered as a dashed line to indicate that there is a considerable variation in properties at these high temperatures. In addition, no allowances have been made for the creep properties of the steel when it is exposed to high temperatures for periods of time greater than one hour. As a result, the estimated improvement in safety is small. In addition, the $3000 increase in the car cost (see Tank Material Specifications) is higher than several other, more effective modifications, and therefore, this design modification is not considered further.

b. Increased Valve Area

An alternative to adding insulation for thermal protection is to increase the safety relief valve capacity. The purpose of the increased valve capacity would be to maintain a lower tank pressure and thereby prevent rupture. Calculations have been made utilizing the Calspan Tank Car Thermal Model which compare the standard, uninsulated tank with a 0.055 ft^2 valve area to a similar car with four times the valve area. The results are shown in Figures 9 and 10. Also shown are results for an insulated tank with a standard valve. For both of the heat fluxes considered (which are typical of

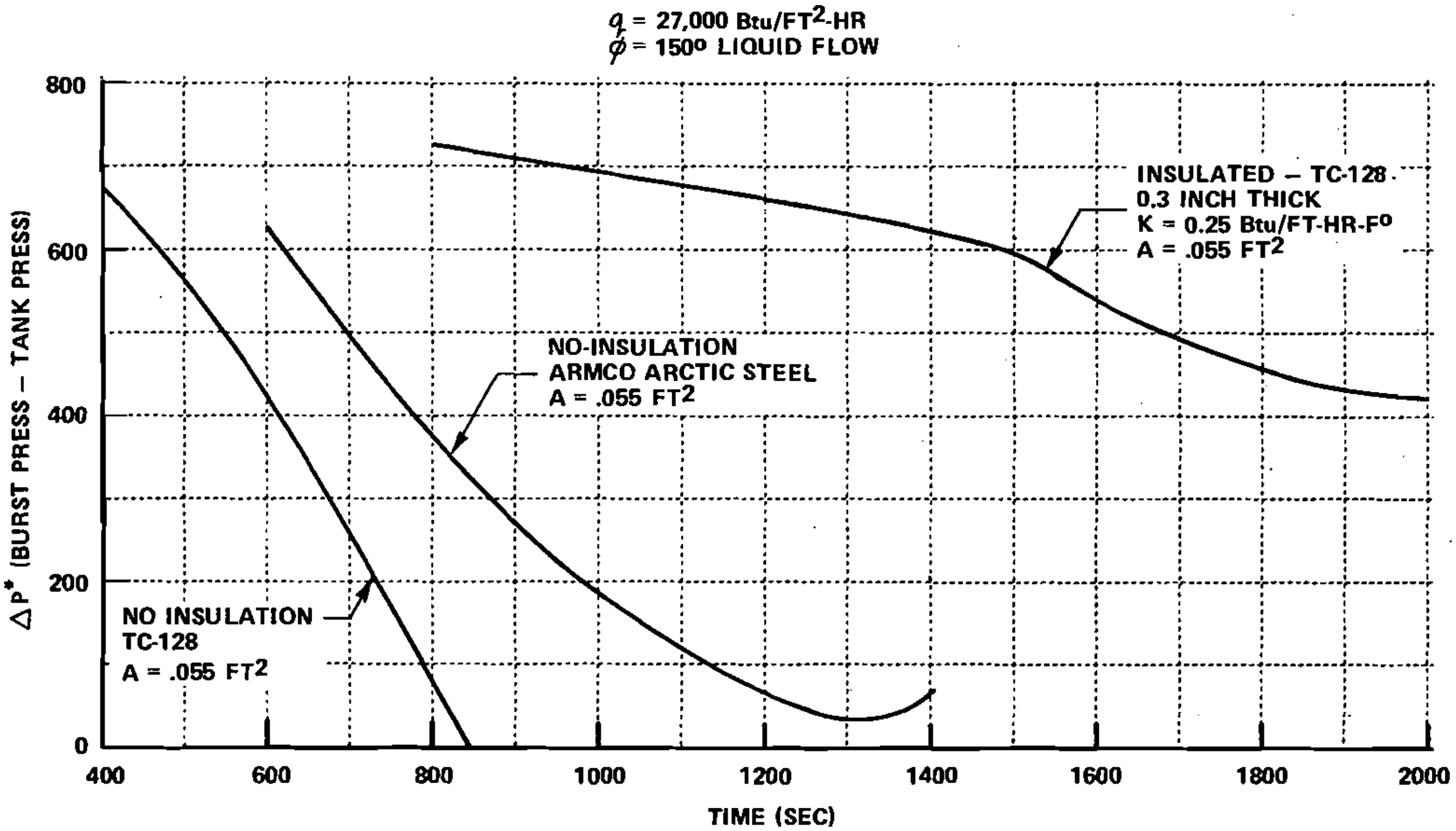


Figure 7 COMPARISON OF THE EFFECT OF INSULATION AND IMPROVED STEEL SPECIFICATION ON TANK PRESSURE

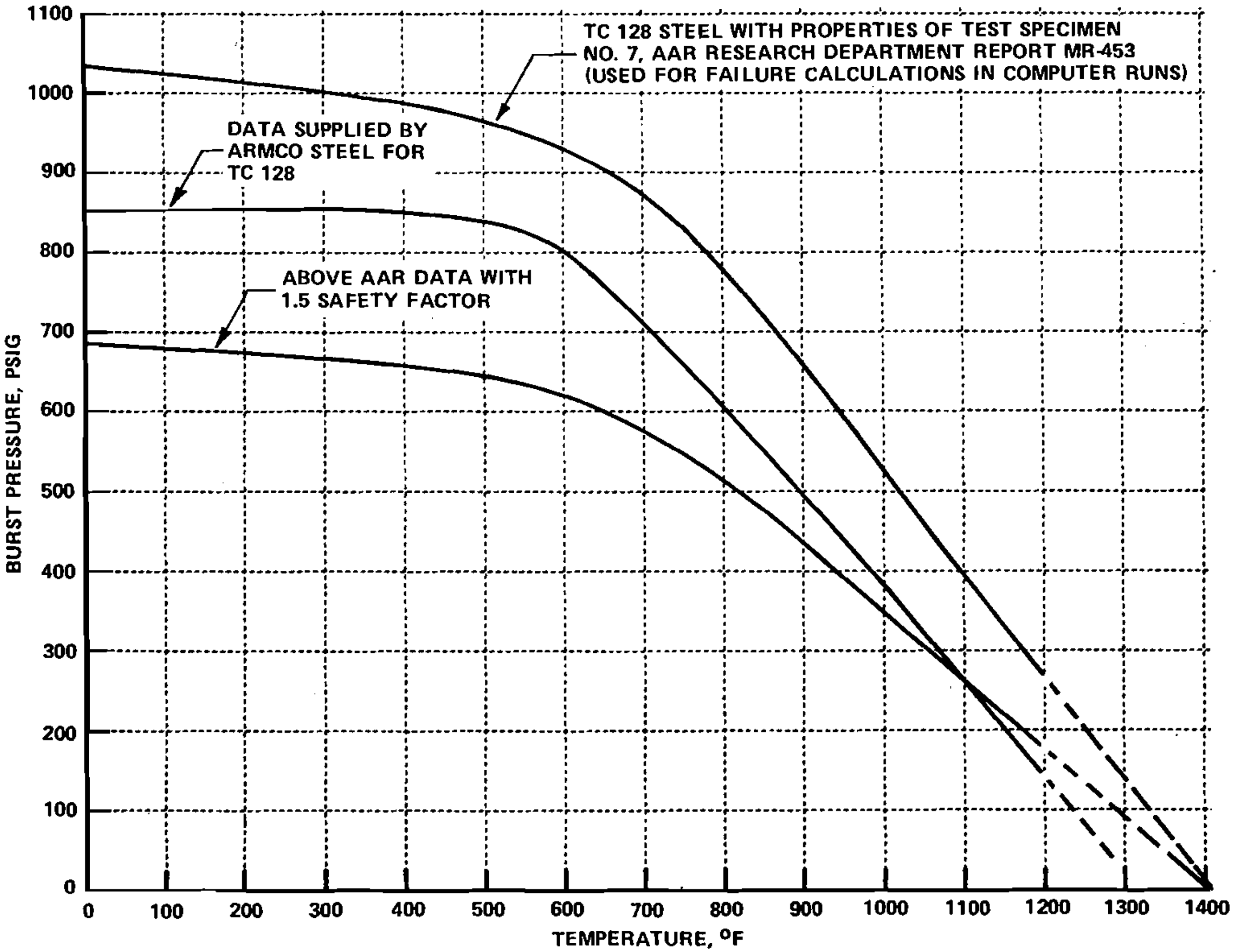


Figure 8 BURST PRESSURE VARIATION WITH TEMPERATURE FOR TC-128 STEEL, TANK DIAMETER 10 FT, TANK THICKNESS 0.625 IN.

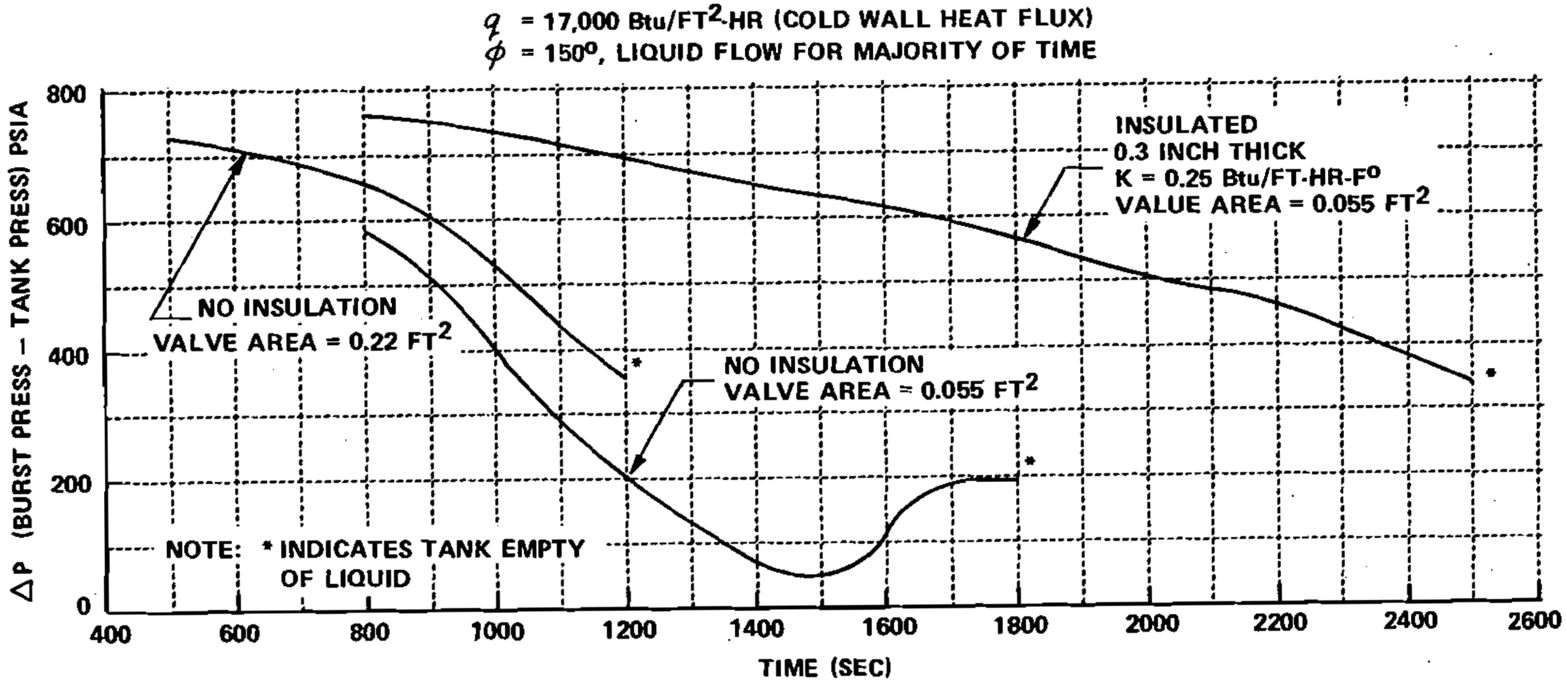


Figure 9 EFFECT OF INSULATION AND VALVE SIZE ON TANK BURST CONDITION

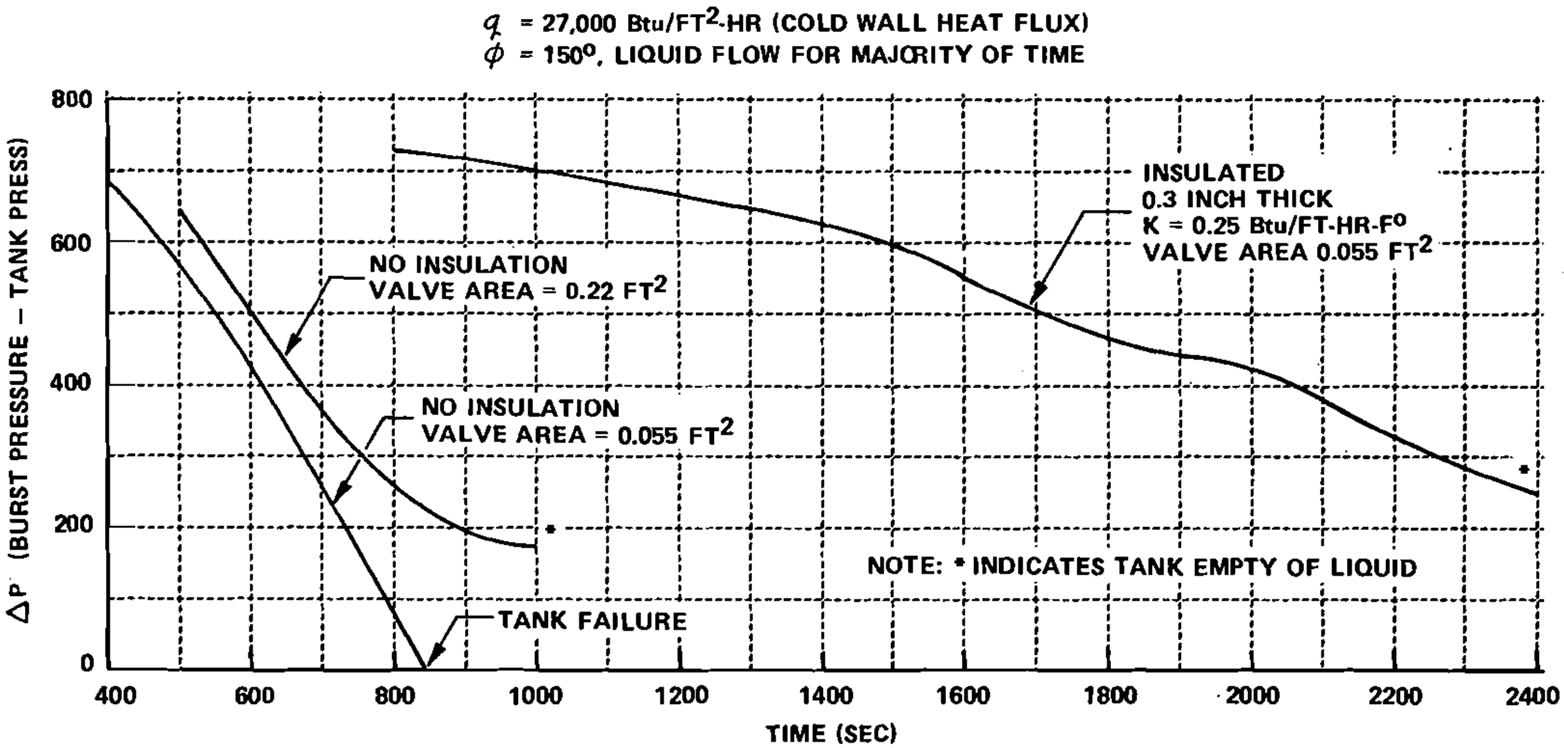


Figure 10 EFFECT OF INSULATION AND VALVE SIZE ON TANK BURST CONDITION

fires), increasing the valve area by a factor of four would allow complete venting of liquid without rupture. The insulation would provide a greater margin of safety but the lower cost of the valve justifies further consideration on a cost/beneficial basis. Modification of the existing valve arrangements may be required to allow for increased valve area particularly within the present manway. Either a single larger valve or multiple valves could be used.

Valves of four times the area of the present valve will cost more than the present valve but not by a factor of four. Extra cost has been estimated to be $1000. This cost is significantly less than for a thermal shield coating. The effectiveness of this modification in preventing losses resulting from fire cannot be fully assessed because the historical data on accidents is very incomplete relative to valve operation. It is not known in most cases whether or not the valve operated. However, in the 20 fire exposure cases in which it is known whether or not the valve operated, there were 11 instances in which the valves did not open and only one of these resulted in a major rupture.[23] The indication from this limited amount of data is that the tanks are not rupturing before valve operation but that even though the valves operate, ruptures are not prevented. As shown in Table XVII, the losses for tank cars exposed to fire are $306/car/year. For a 30 year life and 10 percent interest rate, the amount that could be invested in a 100 percent effective valve which would prevent the losses from fire exposure would be $305 x 9.5 = $2907.

There is one serious defect with a change in valve area as a safety improvement. According to the computer calculations, the top of the tank reaches a temperature such that the top is very close to failure even though the valve is cycling open and closed and maintaining pressure within the tank at the valve setpoint. The fire input parameters and tank burst strength are not known with sufficient accuracy (see Figure 8) to be certain tank failure could not occur at the valve setpoint. The strength of the steel is also known to decrease when held at elevated temperatures; a 25 percent decrease after 1 hour at 1000°F.[24] The uncertainty in these values is such that failure may actually occur in some instances. An alternative valve solution which lowers tank pressure below the valve setpoint is discussed below. The valve arrangement discussed in this section, however, appears to be a cost beneficial solution if it can be shown through further testing that shell temperatures will not be excessive at the valve set point.

c. Lading-Temperature-Actuated Valve

Another safety relief valve modification worthy of consideration consists of a primary valve which is actuated by excessive lading temperature. Such a valve would open at a set value of lading temperature which could only occur if the tank were exposed to fire. The valve would then remain open to assure that the tank pressure decreased to near ambient pressure, thus, eliminating the danger of tank rupture.

Present valves are made to open and close near a set point which keeps the tank at an elevated pressure (about 300 psig for 112A/114A cars). If the tank wall temperature becomes high enough, the tank will fail even at the set point. The required wall temperature is about 1200°F (Figure 8). This temperature includes no safety factor or effect of reduced strength of steel held at elevated temperature for long periods. From the accident data it is evident that tanks generally fail because the pressure is too high for existing tank strength at the wall temperature. It is not known whether the tank pressure was at the valve set point but it is assumed that there must be instances where this has occurred. Appendix C includes data which shows that of 55 cars exposed to fires of sufficient size to actuate the valve, 50 eventually ruptured. In some of these accidents the valve must have had sufficient capacity to maintain tank pressure at the set point and yet they still ruptured. If the tanks had a valve of sufficient capacity, actuated by lading temperature, and remaining open after first being actuated, then the tanks would probably not have ruptured and accident losses would have been much less.

To implement such a valve system, a secondary valve of conventional type would have to be included to relieve the tank in the event of overfills or if the tank goes shell full from ambient heating. To prevent the possibility that the secondary valve would have sufficient capacity to keep the pressure at its set point, the secondary valve should have a set point pressure above the saturation pressure corresponding to the temperature set point of the primary valve. For example, the primary valve could be set to open at a lading temperature of 140°F (corresponding to a propane saturation pressure of 290 psig) and the secondary valve could be set to open at 300 psig. In operation the secondary valve would open any time the pressure reached 300 psig such as during an overfill or if the tank heated sufficiently from the sun to overfill (115°F at summer loading conditions) but the only condition that would open the primary valve would be a lading temperature of 140°F. This would only happen during a fire. The primary valve would then remain open and the pressure in the tank would drop to safe levels. The accident data make it evident that pressure reduction is necessary to prevent tank rupture. If the lading reaches 140°F, the tank must be involved in a fire. With the present valve system, a fire results in a rupture 91 percent of the time (see Appendix C). The consequences of venting all of the lading are much less than for a rupture so that complete venting is the preferable alternative. The valve setpoints listed above are satisfactory only for propane. Other ladings such as liquefied ammonia would require a valve change.

As described in the preceding section, the amount that could be invested in a valve system which would be 100 percent effective in preventing losses from fire exposure would be $2907. The net cost of the lading-temperature-actuated valve system will be about the same as the increased size valves described in the preceding section, i.e. $1000. Therefore, an effectiveness of only 34 percent would result in a cost beneficial modification. As discussed above, the indication from the limited amount of accident data is that tanks have not been rupturing before valve operation and therefore a valve actuated by lading temperature is expected to be very effective. Calspan believes this technique warrants further study as a potentially cost/beneficial modification.

d. Positive Vapor Discharge

Reference 9 shows that vapor discharge from the safety relief valve poses a much less stringent requirement on valve size than liquid discharge. During an accident, a car may overturn and the valve may be required to flow liquid, potentially the most dangerous situation. A possible modification would consist of an arrangement that would insure that the valve was always communicating with the vapor for any tilt angle of the car. This might be accomplished by a flexible tubing connection from the valve to the vapor space; the end in the vapor space being attached to a float.

Besides the possible complexity of this type of modification in practical use, it suffers from the same deficiency as increased valve size. That is, even though the valve has sufficient capacity, it still operates to maintain the pressure near a set point and the tank could fail at this set point pressure if the tank walls reach a high enough temperature. Therefore, this type of modification is not expected to have a high effectiveness and the lading-temperature-actuated valve is believed to be the preferred alternative.

F. Economic Sensitivity

In determining an upper bound consistent with a viable service, Calspan performed a sensitivity study to determine the effect of increased tank car cost on the delivered price of the shipped commodity even if the car modification were not strictly cost beneficial. This study was limited to the shipment of LPG a distance of approximately 800 miles in 112A340W type cars. The general conclusion of this study is that for a tank car carrying only LPG, a 10 percent increase in the tank car cost would produce only a 0.5 percent increase in the delivered cost of the LPG. Since LPG is probably the lowest priced commodity to be shipped in the noninsulated pressure car, this 0.5 percent represents an upper bound on price increase. Similar conclusions would be obtained for trips of different lengths.

The assumptions used in this study are presented in Table XVIII along with the references from which these data were taken. Transportation costs were computed using the data and procedures presented in "Rail Carload Cost Scales by Territories," Reference 25. A hypothetical trip of a 112A car from Houston to St. Louis carrying 1280 cwt of LPG was used for the basic computation. Tank car initial costs were increased by $500, $1000, and $2000. These increased costs were spread over the 30-year life of the tank car so that, on the average, the car would return the increased cost.

In this country, the majority of the pressure cars are privately owned with approximately 60 percent owned by leasing companies. For cars owned by leasing companies, loading and car movement reports are furnished by the shipper to the lessor. Mileage earnings are then paid to the lessor by the railroad and credited to the lessee to be applied against rental charges accrued under respective leases. As a result, two sets of payments are made to the car lessor: (1) the mileage charge, and (2) an unknown lump sum specified by the lease agreement. Since only the mileage charge is visible, the increased tank car cost was incorporated into it. This procedure inflates the shipping rate to the benefit of the shipper's commodity price at the point of origin. The cost of the commodity at its destination, however, properly reflects the increased car cost.

Table XIX presents the method used to incorporate the increased car cost into the variable cost portion of the transportation rate. Table XX presents the computation of the transportation rate as a function of the increased car costs, and Table XXI presents the computation of the increased commodity costs. Figure 11 presents the last result graphically.

If the car modification is installed to reduce losses, there is a benefit derived from increasing the tank car cost. An attempt has been made to determine the magnitude of this benefit and incorporate it into the transportation rate computations. As an example, the modification was assumed to reduce the increased transportation rates by 25 percent. The result is plotted on Figure 11 as the 75 percent cost line.

Table XVIII

INCREASED CAR COST SPREAD OVER THE 112 FLEET

Average No. of cars, p. 4, Ref. 1	12,000
Average No. of years	30
Average annual tank car usage . . . . (car * miles/year - loaded) (p. 24, Ref. 1)	6.7×10^7
Increased car cost.	$500 $1000 $2000
General overhead rate . . . . Region IV, Table 7, Ref. 25	0.17473
Empty return ratio . . . . Region IV, Table 3, Ref. 25	1.08

Table XIX
CHANGE IN CAR RENTAL RATE TO INCLUDE INCREASED CAR COST

$$\Delta RC = \frac{\text{cost/car x car}}{\text{Tmile/yr} \quad (1 + E/R) * (1 + OH) \text{ x yr}} \qquad (¢/\text{mile})$$

Region IV

	ΔRC	75% ΔRC
\$ 500	0.122	0.091
\$1000	0.244	0.182
\$2000	0.488	0.364

INCREASED VARIABLE COST TO REFLECT INCREASED CAR COST

Region IV

$$CCM' = CCM + \Delta RC * (1 + OH)(1 + E/R) \qquad (¢/\text{mile})$$

Way Train	CCM	CCM' 100%	CCM' 75%
\$ 500	47.76796	48.0659	47.9914
\$1000		48.3640	48.2150
\$2000		48.9602	48.6621

Through Train			
\$ 500	42.44149	42.7394	42.6649
\$1000		43.0375	42.8888
\$2000		43.6337	43.3335

ΔRC	Change in rental cost
E/R	Empty return ratio
OH	Overhead factor
Tmile	Total miles = car x miles/car/yr. = miles/yr.
CCM	Cost per car mile

Table XX

TRANSPORTATION RATE AS FUNCTION OF INCREASED CAR COST

HOUSTON TO ST. LOUIS — 794 MILES
REGION IV — CIRCUITRY 1.16
100% — 1280 CWT

Increased Car Cost	$0	$500	$1000	$2000
Variable Cost				
Terminal				
Per carload	4132.544	4132.544	4132.544	4132.544
Per cwt	0.018	0.018	0.018	0.018
Total (per cwt)	3.2465	3.2465	3.2465	3.2465
Way Train				
Per car-mile	47.76796	48.0659	48.3640	48.9602
Per cwt-mile	0.01294	0.01294	0.01294	0.01294
Total cwt-mile				
Through Train				
Per car-mile	42.44149	42.7394	43.0375	43.6337
Per cwt-mile	0.01017	0.01017	0.01017	0.01017
Total cwt-mile				
Constant Expense				
Terminal per cwt	1.179	1.179	1.179	1.179
Line-Haul per cwt-mile	0.0082	0.0082	0.0082	0.0082
Fully Allocated Cost				
Way train cwt-mile	0.06781	0.06808	0.06835	0.06889
Through train cwt-mile	0.05977	0.06004	0.06031	0.06085
Total terminal cost	4.4255	4.4255	4.4255	4.4255
Total way train cost	5.6962	5.7187	5.7415	5.7869
Total through train cost	42.4379	42.6296	42.8214	43.2050
Total cost/cwt	52.5596	52.7738	52.9884	53.4174

Table XX (Cont.)

HOUSTON TO ST. LOUIS	794 MILES
REGION IV	CIRCUITRY 1.16
75%	1280 CWT

Increased Car Cost	$0	$500	$1000	$2000
Variable Cost				
Terminal				
Per carload	4132.544	4132.544	4132.544	4132.544
Per cwt	0.018	0.018	0.018	0.018
Total (per cwt)	3.2465	3.2465	3.2465	3.2465
Way Train				
Per car-mile	47.76796	47.9914	48.2150	48.6621
Per cwt-mile	0.01294	0.01294	0.01294	0.01294
Total cwt-mile				
Through Train				
Per car-mile	42.44149	42.6649	42.8885	43.3335
Per cwt-mile	0.01017	0.01017	0.01017	0.01017
Total cwt-mile				
Constant Expense				
Terminal per cwt	1.179	1.179	1.179	1.179
Line-Haul per cwt-mile	0.0082	0.0082	0.0082	0.0082
Fully Allocated Cost				
Way train cwt-mile	0.06791	0.06801	0.06821	0.06862
Through train cwt-mile	0.05977	0.05997	0.06017	0.06058
Total terminal cost	4.4255	4.4255	4.4255	4.4255
Total way train cost	5.6962	5.7132	5.7296	5.7642
Total through train cost	42.4379	42.5817	42.7256	43.0119
Total cost/cwt	52.5596	52.7204	52.8807	53.2016

Table XXI
PRICE INCREASE AT DESTINATION VS. INCREASED CAR COST

		$500	$1000	$2000
Works Price ¢/cwt	110.00	110.00	110.00	110.00
Transport Cost ¢/cwt	52.55	52.77 52.72	52.99 52.88	53.42 53.20
Price at Destination ¢/cwt	162.55	162.77 162.72	162.99 162.88	163.42 163.20
% Increase 100% 75%		0.13 0.09	0.27 0.20	0.53 0.39

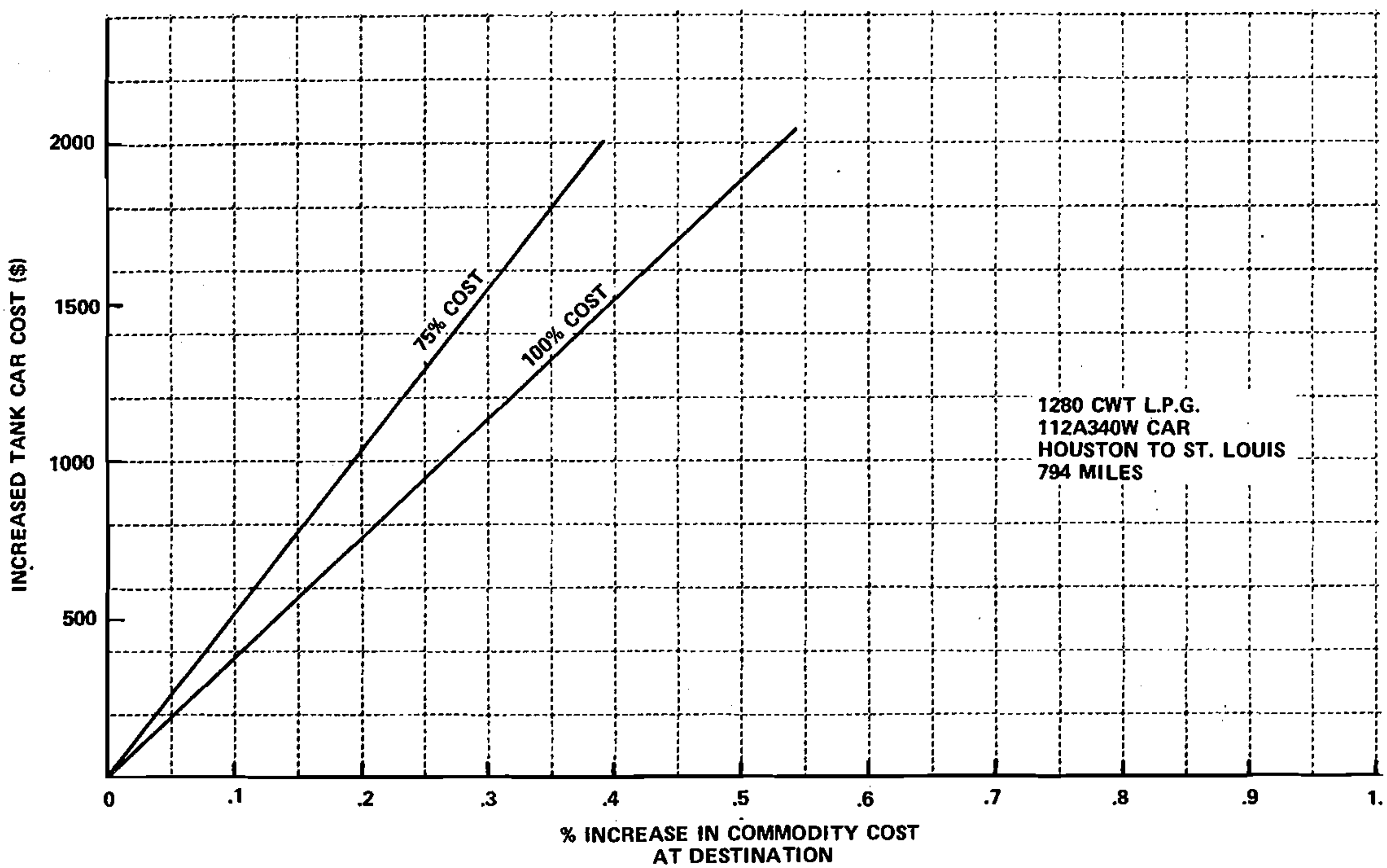


Figure 11 PRICE INCREASE AT DESTINATION vs INCREASED CAR COST

The scope of this sensitivity analysis is too limited to draw broad general conclusions. However, it does indicate that for modifications to tank cars of the order of $2000 to $4000 (including adjustments for economic life and cost of capital factors), the price increase of the delivered LPG will probably be 0.5 to 1 percent, a not impractical increase considering the protection afforded especially in light of the recently much greater increase in delivered price of LPG due to increased source prices.

IV. SUPPORTING STUDIES

Several studies are reported in this section in support of the evaluation of design improvements for tank cars. One of these supporting studies concerns the extension of the tank-fire computer simulation developed by Calspan. Use of this updated computer simulation in the analysis of the 1/5 scale and full scale fire tests which have been conducted at White Sands Missile Range is also presented. Also reported is a review of existing thermal research which is being conducted by FRA, RPI/AAR and others. Finally, several individual topics related to thermal research are reviewed.

A. Description of the Calspan Tank Car Thermal Model

The Calspan computer program for the mathematical model of the tank car, or thermal model, represents a tank car either upright or rolled over at any angle, that is enveloped by fire. The complete program is given in Appendix D. The tank car geometry is described by inputs for its length, diameter, shell thickness, number of relief valves, their position along the tank, their flow area, discharge coefficient, and the tilt or roll angle from the vertical.

In addition, if external insulation is present, it is specified by its thickness, thermal conductivity (which may be varied with temperature), and the product of density and specific heat.

The tank is divided into elements for computation by specifying the number of divisions around half the periphery and the number of divisions of length. The steel of the shell is described by burst pressure tables that are based upon ultimate strength, and are prepared by calculating burst pressure from a simple thin shell relation. That is, thermal stresses due to circumferential or longitudinal temperature gradients are assumed to be negligible. This is in agreement with analyses of failures of tanks in fires which indicate that the predominant failure mode is thinning of the shell over the vapor space followed by the initiation of a crack along a longitudinal line. This indicates a pressure induced failure rather than a thermal stress failure.

Valve operating pressure limits must be specified for opening and closing. The lading is described by an array of thermal properties for the saturated conditions, i.e., enthalpy, temperature, pressure, and specific volume of both liquid and vapor. In addition, its ratio of specific heat, gas constant and the total weight of lading per foot of tank must be given.

The heat input from the fire is described by inputs for its temperature, emissivity, and the heat transfer coefficient for convective heating. An emissivity for the tank shell must be given. Heat input to the lading is described for liquid and vapor separately by a heat transfer coefficient. Liquid heat transfer coefficients are computed by equations that represent curve fits to experimental data, and are valid for propane only.

Fundamental assumptions were made in the theoretical development of the model which contributed to rendering the problem tractable while keeping the model practical. These are listed as follows:

1. Temperature of the bulk of liquid is uniform.

2. The heat transfer coefficient for shell to lading heat transfer is uniformly distributed over the surface although a distribution is made between liquid and vapor heat transfer coefficients. The coefficient for vapor is constant but that for liquid is variable with pressure and temperature differences.

3. Conduction of heat in the tank shell in a direction parallel to the axis of the tank is negligible.

4. Thermal properties of the shell do not change with temperature but thermal conductivity of the insulation may vary.

5. The location of the liquid surface is identified only by the angle to the centroid of the particular element of the

tank car shell that it contacts. In all other respects, the surface is assumed to be confined to a horizontal plane.

The model computes heat penetration to the lading, which results in a computed rise in temperature of the external insulation, if any, the tank shell, the vaporized lading, and the liquid lading. In computing the external heating, heat is reradiated to the fire at increasing rate, and convective heating decreases as the outer surface temperature rises, resulting in a reduced heat penetration to the lading.

The initial effect of heat input to the lading is to cause pressure to rise because of the increase in vapor pressure as lading temperature rises. The amount of liquid also increases as it is heated. If the tank contains sufficient lading, a point is reached where the tank becomes shell full, and no vapor is present.

Such effects as these are duly represented as a result of computations using heat and mass balances on the lading. Given the specific volume for both vapor and liquid from either a previous computation cycle or the initial values, the masses of liquid and vapor per foot of length are computed. If the pressure at this previous time is sufficiently high to open the valve, the mass lost through the valve is computed for the computing interval and subtracted from the total mass of lading. Of course, a distinction is made between liquid and vapor flow depending upon the roll position of the valve relative to the instantaneous depth of liquid.

The solution for the conditions in the tank at the end of a computing interval is obtained by an iterative method because numerous simultaneous equations must be solved, which involve unknown variables that must be evaluated using tables or arrays of input data, such as the thermal properties of saturated lading. These equations include the heat balances mentioned earlier and mass balances that update the masses of liquid and vapor to maintain consistent sets of conditions.

The iteration procedure starts by estimating a new pressure and determining the saturated-lading thermal properties that correspond to it. Separate schemes are used for the open valve case. Then an equation for the heat balance to the liquid is used to solve for the new mass of liquid at the end of the computing interval. If an open valve lies below the surface of liquid, the mass of liquid is adjusted to reflect the liquid discharge. If no valves are open, the change in mass of liquid is restricted to either vaporization or condensation.

To continue, the iteration proceeds by computing the enthalpy of the vapor from a second heat balance that is taken on the vapor space. This enthalpy is not allowed to exceed the enthalpy of vapor at the temperature of the liquid by an amount corresponding to 0.7 times the difference between temperature of the liquid and temperature of the top element of the shell. Then new values for specific heat, temperature, and specific volume are computed from curve fits of the superheated propane vapor data which are described in Reference 26. These are used in a computation for QIN, the heat absorbed by the lading, which is obtained from an equation for the overall heat balance on the lading. QIN is compared to the summation of the heat transferred to the lading from the shell, PREV, which equals the total heat input to the liquid, QLSUM, plus the total heat input to the vapor, QGSUM. If agreement between PREV and QIN is unsatisfactory, a new pressure is estimated on the basis of the departure of QIN from PREV and the iteration procedure is repeated.

When agreement is achieved, a second test must be passed to insure that the tank pressure does not drop significantly below valve closing pressure in the event the valve has been open, or above valve opening pressure if it has been closed. When these results occur they simply indicate that the computing interval is too long to represent events accurately. Consequently, a feature is provided that alternately reduces the computing interval, recomputes all heat inputs, temperatures, etc., and reiterates for pressure. When the computing interval is reduced sufficiently to produce the correct tank pressure, relative to valve operation, the computation proceeds to the next phase. Note that this procedure does not restrict the tank pressure from rising to values above the valve opening pressure.

After the pressure has been determined, a computing loop is entered which checks for tank rupture due to excessive pressure at the shell element of highest temperature. If tank pressure exceeds the allowed pressure for burst, the computing stops, giving a burst condition printout.

Next, stresses are computed for the shell, which are printed out for the uppermost element. Time is updated, the original computing interval is restored in the event it had been changed during iteration, and a check is made for mass of liquid. If it has been reduced to zero, the computation stops. If not, it proceeds to the next cycle.

B. Improvements to the Tank Car Thermal Model for Use in Engineering Studies

The computer program for the tank car thermal model was devised in fulfillment of previous contracts (Reference 26). It has been further developed and refined during this present contract in order to make it more suitable to the needs of the work that is now underway. All changes made to the computer program were confined to the MAIN routine so it is the only one discussed in subsequent paragraphs. A FORTRAN listing is presented in Appendix D.

The first revisions to the program provide a variable temperature with distance above the liquid level in the vapor space. Previously, this was treated as having uniform temperature throughout. A single value for vapor temperature is required for the pressure iteration scheme which deals with the heat balance on the bulk of vapor. Consequently, an average value for vapor temperature is determined. The upper limit of vapor temperature is found at the extreme top of the tank and is the inside shell temperature there. The lower limit is the temperature of the liquid lading. The average value is higher than the lower limit by 0.7 times the difference between limits. This factor (0.7) is subject to change depending upon results of studies of fire test data and can even be made a variable to depend upon such characteristics as vapor space geometry, which changes with depth of liquid. The average temperature is introduced in the computation of average enthalpy of the vapor, TSTHG.

A scheme was devised for computing the temperature of the vapor adjacent to each shell element so that it would be available to the computation for QG, the heat transferred from shell to vapor. However, this complicates the program further and the advantages of using it are minor, inasmuch as the precision required of this computation is only moderate (which is fortunate because there is little information extant concerning heat transfer to propane vapor).

A second revision, and a major one, was to refine the iteration procedure for pressure in the tank car. First, iteration is necessary to solve all the equations (some are represented by tables) that define the changed conditions of the lading after a time interval while heat is added to the lading. This is explained in more detail in Reference 26. However, the procedure presented in that report is oversimplified. The chief relations were mass and heat balances on the overall lading and the mass balance was not complete because too many unknowns existed, although prior values for some were used which provided partial compensation for this deficiency. This situation has been remedied by providing separate heat balances for liquid and vapor in addition to the overall heat balance. Incorporating these equations required changing the logic of the iteration scheme in many respects although the basic concept was retained, i.e., one of estimating pressure, then evaluating the thermal properties of the lading for that pressure to permit computation for the heat absorbed by the lading during the computing interval and finally, comparing the heat absorbed by the lading with the heat transferred to it from the shell and revising the estimated pressure to repeat the iteration if the heat absorbed did not agree with the heat transferred.

Since this revision, the computer program has been exercised numerous times and it has been observed that the iteration procedure has been convergent for all conditions imposed upon it. These range from the case of high heating rates experienced by a bare tank in a fire to one of greatly moderated heat penetration to the shell resulting from the use of very effective insulation and include computations for full size tank cars as well as 1/5-scale tanks.

A third revision is an improved model or technique for representing insulation placed around the outside of the tank shell. The technique provided with the previous model, Reference 26, neglected the heat storage term in the heat conduction equation. It included a representation of the degradation of effective thickness of insulation that decomposes when heated to excessive temperatures. This model has been replaced with one that is somewhat more complicated, in which the heat storage effect is included, but the degradation scheme has been dropped. (It is possible to incorporate degradation although this would require more effort because changes in logic are required to adapt it to the new procedure.)

Moreover, the improved model is two-dimensional in its treatment of insulation. The computations are similar to the determination of tank shell temperature, whereby a grid system is used that provides for radial heat conduction through the shell and heat conduction in the peripheral direction.

Calculations have been made of the response of a tank car to fires using the improved computer programs. The calculation conditions are listed in Table XXII from the best available information. The fire temperature and effective emissivity were taken from the one-fifth scale test results as described in the next section. The effective emissivity used is an experimentally determined value required for the heat transfer calculations and is not necessarily the same as the optical emissivity. A body within a fire cools immediately adjacent gases which then shield it from the bulk of the fire. This reduces the effective emissivity to which the body is subjected.

Several of the other calculation conditions relate to a typical 112A340W tank car of about 33,000-gallon capacity containing propane at the summer loading density. The calculations have considered membrane failure of the tank, with steel strength as given by test specimen No. 7 of AAR Research Department Report MR-453, Figure 8. However, there are several other failure criteria which may also be used. The test pressure of 112A340W tank cars is 340 psig and the AAR's Specification for Tank Cars limits the allowable tank pressure to 306 psig for any shell temperature. This failure criterion is conservative

Table XXII

CALCULATION CONDITIONS

Fire temperature: 1700°F

Effective fire emissivity: 0.4

Convection coefficient adjusted to give a cold wall heat flux equal to that given on curves

The whole tank car is subjected to the same cold wall heat flux

Initial tank temperature: 70°F

Summer loading of propane, i.e., shell full conditions are reached at lading temperature of 115°F

Tank diameter: 10 ft.

Tank length: 60 ft.

Tank wall thickness: 0.625 in.

Tank material: TC 128 with strength properties of test specimen No. 7, AAR Research Department Report MR-453

Amount of insulation: None

Valve discharge coefficient: 0.65

Valve fully opens at 295 psia and fully closes at 280 psia in that it includes a safety factor for most values of shell temperature and it is the only criterion that is based on actual tank test pressures.

Another failure criterion that we have considered is applying a safety factor to the burst pressures. We have used a safety factor of 1.5. Data supplied by Armco Steel Corporation for TC 128 steel indicate that a 1.5 safety factor applied to AAR strength data is not overly conservative. In fact, at temperatures above 1100°F, the Armco data indicate less strength than the AAR data with a 1.5 safety factor. To account for these differences at high temperature, we have reduced the AAR strength data by either a 1.5 safety factor or 200 psi, whichever is greater.

The calculation conditions relative to valve operation are not well known due to the lack of actual test data, but the values shown in Table XXII represent the best available information. Data from the valve tests being conducted at Edwards AFB and full-scale fire tests should provide more definitive information on valve performance.

Figures 12 and 13 show a 33,000 gal. 112A/114A tank response to a fire of 17,000 Btu/hr ft^2 cold wall heat flux. Figure 12 is with the valve vertically up (vapor discharge) and Figure 13 is with the valve at 150° from the vertical (liquid discharge for the majority of the time). For the vapor discharge condition, the valve appears to have sufficient capacity to prevent pressure buildup above 305 psia. However, the temperature of the shell reaches a magnitude such that the tank is close to failure at the valve opening pressure of 295 psia. The borderline nature of this case is also a consequence of the fire conditions used in the calculations which are not precisely known. If the 200 psi safety factor is utilized for this circumstance, the tank would have been considered to have failed at 1720 sec.

Figure 13 shows conditions in a tank with the valve at 150°, i.e., for liquid flow for the majority of the time. This figure gives an indication of the increased severity of liquid flow circumstances. The conditions are the same as those calculated for Figure 12 other than the valve location. For the

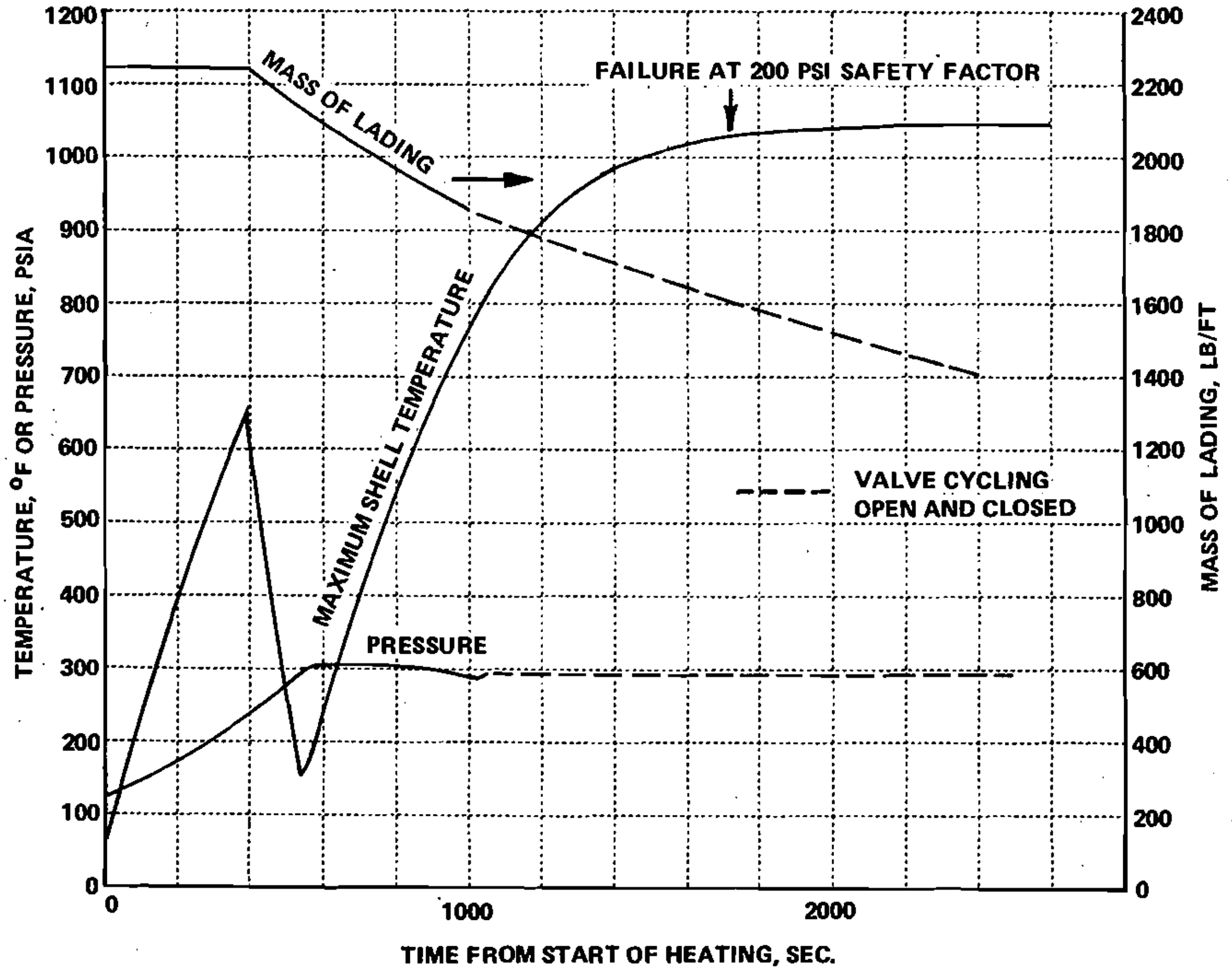


Figure 12 PRESSURE, MAXIMUM SHELL TEMPERATURE, AND MASS OF LADING IN TANK WITH SUMMER PROPANE LOADING SUBJECTED TO 17,000 BTU/HR FT^2 COLD WALL HEAT FLUX, 0.055 FT^2 VALVE AREA, VALVE AT 0°, SEE TABLE XXII FOR OTHER CALCULATION CONDITIONS

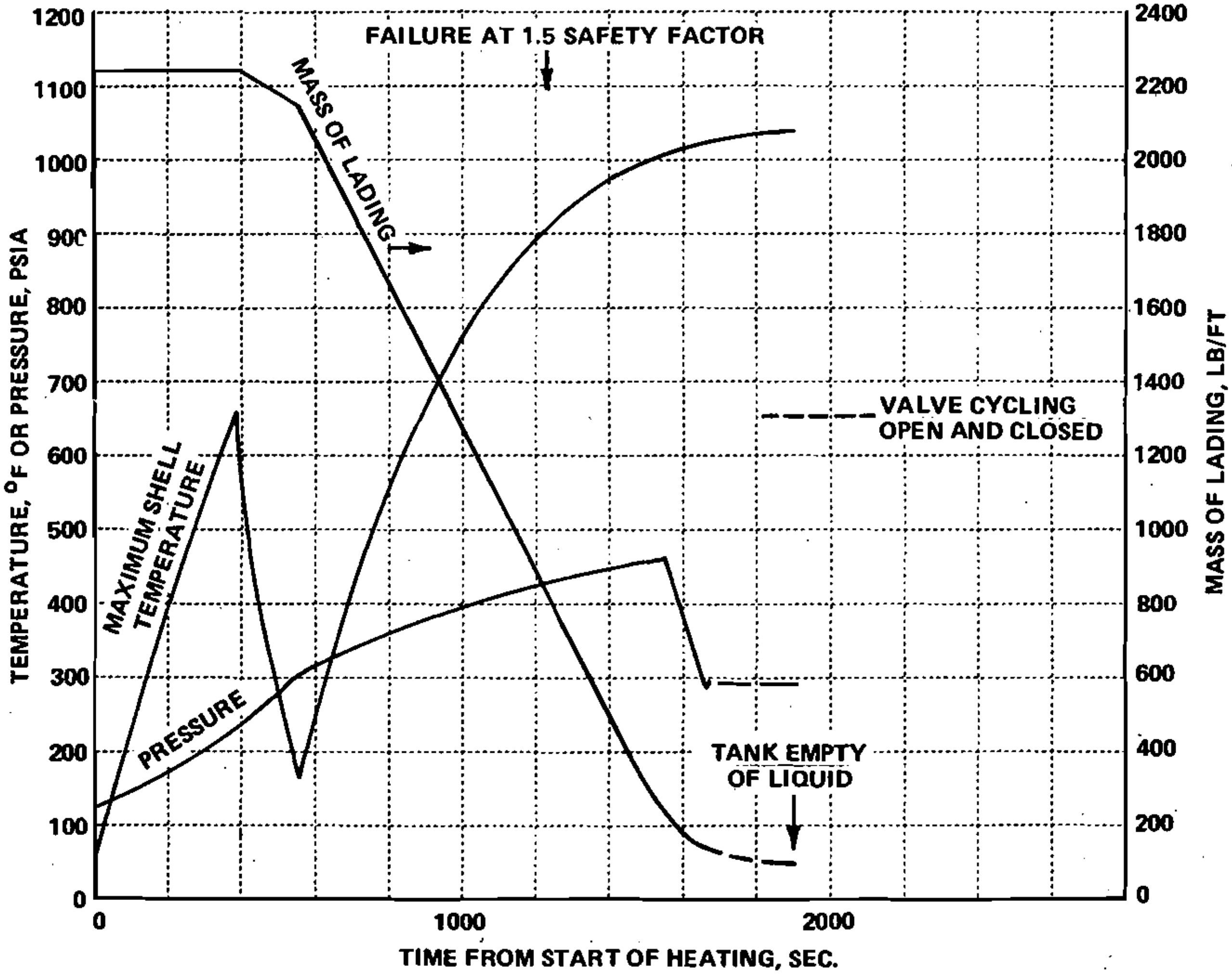


Figure 13 PRESSURE, MAXIMUM SHELL TEMPERATURE, AND MASS OF LADING IN TANK WITH SUMMER PROPANE LOADING SUBJECTED TO 17,000 BTU/HR FT^2 COLD WALL HEAT FLUX, 0.055 FT^2 VALVE AREA, VALVE AT 150°, SEE TABLE XXII FOR OTHER CALCULATION CONDITIONS

liquid flow circumstance, the pressure increases to a much higher value until the lading level reaches the valve and gas flow begins. The pressure then decreases rapidly. Even though the flow rate is higher for liquid flow, tank pressures are observed to be greater. Tank failure at 306 psig, the maximum allowed in the specifications, occurs at 600 sec. Failure at 340 psig, tank test pressure, occurs at 800 sec and at a 1.5 safety factor at 1200 sec.

C. N.O.L. Fire Testing and Results

A series of fire tests were conducted by the Naval Ordnance Laboratory (N.O.L.) for the Federal Railroad Administration (FRA).[26a] The purpose of the tests was to gain information regarding:

1. The fire environment to which tank cars might be subjected.
2. The thermal effects of fire on the tank contents and tank shell.
3. The performance of the Midland A-3480 safety valve.
4. The effectiveness of thermal shield coating materials toward reducing the thermal load to the tank shell and lading.

In pursuing the study program, 1/5 model tank cars containing water and propane were subjected to the fire environment produced by continuous feed of JP-4 fuel into a fire pit. A total of six fire tests were conducted for which test data for the last four tests of tanks containing propane only have been procured. Data for the first two tests of tanks containing water have not yet been obtained but are believed of little value, in any event, due to stated difficulties with temperature instrumentation in those tests. This section, therefore, deals with the results of the four propane tests.

1. Test Instrumentation

In seeking information by which tank car performance in fire environments could be assessed, a number of measurements were desired. Although instrumentation techniques for the above mentioned four 1/5 scale fire tests were not identical, the information desired was the same. For this reason, the description of instrumentation given here is general and essentially relates to all four fire tests.

Temperatures were measured by means of thermocouples. Temperatures of the shell, lading, fire and, if appropriate, insulation, generally were obtained. Shell temperatures were measured at a sufficient number of circumferential locations that good representation of thermal gradients around the shell could be obtained in addition to indication of the angle to the propane liquid by observation of shell temperature break* points. The lading temperatures were indicated through a gridwork of thermocouples placed within the tank. A number of these thermocouples were provided with radiation shields in an effort to prevent overtemperature indications in the vapor space due to radiation from the hot shell walls. Fire temperature indications were obtained by a ring of thermocouples placed approximately four inches from the tank outer shell. Temperatures in the intumescent mastic insulation were obtained at a number of circumferential locations. These thermocouples were topcoated with the insulation system under test.

Tank pressures were obtained by use of two redundant pressure transducers placed within the tank.

The lift of the valve or valve displacement was obtained by use of a linear differential transformer interior to the tank and a rectilinear potentiometer external to the tank. These units were calibrated prior to test and their signals were recorded continuously during test.

Heat flux from the fire was indicated by use of a Hycal heat flux calorimeter capable of continuous recording through the test period, and a large calorimeter bottle containing water by which gross measurements of heat flux could be obtained. In addition to these devices, total heat flux to the shell could also be estimated by determination of rate of propane loss from the tank.

*Discussed later.

2. Description of Tests and Observations

As mentioned above, the first two water tests will not be considered. The third fire test, conducted March 17, 1972, consisted of an uninsulated 1/5 scale tank having the relief valve placed vertical. In this test, 239 gallons of propane were loaded into the tank and the fire test was initiated under calm wind conditions. Venting of propane was observed at about 78 seconds after initiation of the test. Temperature of the lading, shell, and fire were recorded as were the pressures in the tank. Due to an oversight, water cooling of the heat meters was not provided. Hence, their output for the first minute only was considered reliable. Liquid level devices were inoperative. Valve lift measurements were highly questionable.

The fourth fire test, conducted June 20, 1972, consisted of a 1/5 scale tank outfitted with a nominal 7/16 inch thickness of the N.O.L. insulation system. As in test No. 3 the full sized A-3480 Midland valve was in the vertical position. In this fourth test, 226 gallons of propane were loaded into the tank and the fire test was initiated under calm wind conditions. Temperature data were obtained for the lading and shell. Heat flux data for the first 24 minutes of test were obtained by the Hycal heat flux sensor. The calorimeter bottle also functioned satisfactorily. Application of secondary pressure instrumentation after failure of the primary pressure transducers prior to the test was apparently unsuccessful in that questionable pressure data was obtained. The linear transformer was inoperative and valve lift measurements as given by the rectilinear potentiometer appeared erroneous. Temperatures at the exterior surface of the insulation were also obtained through the majority of the test but were no doubt indicating essentially the flame temperature of the fire.

The fifth fire test was conducted on July 12, 1972, for the purpose of investigating the effectiveness of a proprietary coating system consisting of a mastic and an overcoat. Again, the A-3480 Midland safety valve was in the vertical position. Temperatures of the lading and shell were obtained as were temperatures of the fire. Temperatures recorded at the exterior surface of the insulation were unreliable. The heat flux gage failed shortly after the start of test, and water temperature within the calorimeter bottle exhibited questionable behavior. Tank pressure as a function of time was obtained for thirty-seven minutes of test time. The rectilinear potentiometer failed after only about 30 seconds of operation, but the differential transformer indicated essentially no valve motion throughout the test.* Post test examination of the spring of the safety valve indicated it to be approximately 9 percent longer than that of the valve spring of the fourth fire test. The effects produced by overheating of the spring were also evidenced.

A sixth 1/5 scale fire test was conducted on November 1, 1972. The purpose of the test was to establish the effects of fire on an uninsulated tank containing 226 gallons of propane, having the safety valve oriented at 90° from the vertical. Data regarding temperatures, heat fluxes, tank pressures, and valve displacement were obtained in this test. It was observed that mild venting of the valve occurred for 2 seconds after about 45 seconds into the test. After 74 seconds, near continuous cyclic discharge was noted with sound described as that of a steam engine.

3. Reported Test Results

Data obtained in the tests described above were reduced, analyzed, and submitted through several brief notes and progress reports issued by N.O.L. to the FRA. Items included in these analyses were exterior heating rates, fire temperatures, tank pressure, heating rate to lading, valve opening time and displacement, shell temperatures, and lading temperatures. Unfortunately, as noted above, information on all of these items was not obtained in each test due primarily to equipment failure. In addition, analysis of each test did not always include determination of the same quantities. For the most part, however, information regarding fire environment and tank car thermodynamics was reported. Table XXIII illustrates the reported data for fire tests No. 3 through 6. Simple comments with respect to values cited are given at the foot of the

*At 500 seconds into the test, a single impulse was observed.

Table XXIII

REPORTED RESULTS FOR FIRE TESTS OF 1/5 SCALE TANKS CONTAINING PROPANE

Test No.	Test Type	Valve Position	Fire Temp.	Max. Shell Temp.	Heat Flux (meter)	Heat Flux (bottle)	Heat Flux (lading)	Valve[1] Time	Valve Displacement	Average Discharge Rate
3	non-insulated	vert.	15-1900°F	1380°F	45,000[2]	----	33,600[3]	78 sec	----	----
4	NOL insulation	vert.	----	1000	26,500	30,300	2,600	60[4]	----	----
5	Proprietary Coating Sys.	vert.	16-1800	1000	----	21,000	----	420	----	0.55 lb/sec
6	non-insulated	horiz.	----	980	38,300	----	39,000[5]	45	0.25 in.	----

General Note: Tabulated values rounded for simplicity.

[1] Specified time to first valve action

[2] Average of two heat meter readings in the first minute of operation, Btu/hr-ft^2

[3] As given in "Analysis of 1/5-Scale Fire Test Data," by L.J. Manda, Railroad Tank Car Safety Research and Test Project, RPI/AAR Report RA-11-2-14, April 12, 1972.

[4] This reported value believed to be reported in error by ten to one; better estimate is 600 sec.

[5] Some question regarding accuracy of this figure due to insufficient data.

table. The following section gives a more complete commentary regarding methods utilized in generation of the reduced results and gives added perspective concerning information gained as well as possible conduct of future tests.

4. Examination of Results and Discussion

In reviewing the reported results, one must consider the degree to which the purposes of the test program have been met, as well as the precision of the reduced results. The following is intended to form a brief summary and discussion of the information thus far gained with some supplementary analysis of data. Suggestions for possible future testing are also included in the section.

a. Fire Environment

The two chief items of importance relating to fire environment are the fire temperatures and the heat fluxes (both radiative and convective) produced. Fire temperature is of importance in the sense that it is the controlling or driving potential for heat flow to the tank or insulation surface. In this respect it is to be noted that it is, therefore, the highest possible temperature to which any exposed object can be heated. The heat fluxes are of importance in that they govern the rate of change of temperature of exposed objects and/or the rate of vaporization of material. In this regard, it must be noted that heat flux is not solely a property of the fire environment but is also governed to a degree by the geometry and temperature of the exposed object. Convective heating is differently affected by object size and temperature than is radiative heating, and therefore discrimination between convective and radiative flux is necessary. Relative to observed data (see Table XXIII), it is evident that JP-4 fire temperatures approached peak values near 2000^{o}F with a reasonable average of 1700^{o}F over the majority of the test time. While JP-4 is certainly not propane, it may be taken as a reasonable representation of hydrocarbon fuel fires including that of propane.

The heat fluxes generated by the fire in the 1/5 scale tests were evaluated by a number of methods, but unfortunately no actual determinations of radiant heating were obtained. Heat flux values reported are gross measurements in which both convective heating and radiative heating are combined. In generalizing the heat flux data (Table VII), one observes the total heating rate to be of the order 35,000 Btu/hr-ft^2. There is, however, significant dispersion in the heat flux data depending upon the method by which it was established. It must be noted that heat flux measuring devices respond to net incident flux <u>to the devices</u> not necessarily to the object under test. Hence, probably the best measure of heating rate is that obtained by evaluation of lading loss by vaporization signified by rapid change in circumferential shell thermocouple outputs (break points) which were taken to indicate height of liquid in the tank as well as wetted surface area. The range of values for the lading appears to be from 33,000 to 40,000 Btu/hr-ft^2 in the number of non-insulated tank experiments conducted (two).

There are discrepancies in the heat flux data obtained from the bottle calorimeter. For example, the heat flux determined in test No. 4 was 30,300 Btu/ft^2 hr. However, more detailed inspection of the temperature output of the calorimeter bottle in this same test shows a definite arrest at the boiling point of the water. This arrest is found to exist for no more than 850 sec. Inasmuch as there was 10.66 lb. of water in the bottle and the heat input was presumed to be over the exposed area of a 7-1/2 inch diameter circle (0.305 ft^2), one gets

$$\text{vaporization} \approx \frac{10.66\ (970)\ 3600}{850\ \ (0.305)}$$

$$\approx 143{,}000\ \frac{\text{Btu}}{\text{hr-ft}^2}$$

This value is inconsistent with the above and further appears much too large. It would appear that values generated by the calorimeter bottle can be considerably in error possibly due to heat penetration of the asbestos paper insulation surrounding the container.

As noted above, the individual components of the surface heating rate convection and radiation were not established and this limits the usefulness of the data toward establishing tank temperatures and resulting net heating of the unwetted tank shell. There was, however, observed a maximum inner shell temperature of about 1380^{o}F in fire test No. 3. It would be expected that this shell would radiate at near blackbody conditions. Hence, the reradiation at this shell temperature is about 20,000 Btu/hr-ft^2. To be supplied solely by convection from a source at 1700^{o}F (approximate flame temperature) would require a convection coefficient of about 62 Btu/hr-ft^2-^{o}F. This would result in an unreasonable initial tank heating rate of 62(1700-70) = 100,000 Btu/hr-ft^2. Hence, a large portion of the heating of the tank must be by radiation. Assuming the initial total heat flux to be about 40,000 Btu/hr-ft^2 (as indicated by lading vaporization) and the average flame temperature to be 1700^{o}F, it follows that the initial radiation contribution must be at least 15,000 Btu/hr-ft^2 in order to produce a maximum shell temperature of 1380^{o}F. This requires a fire emissivity of about 0.4. Further, the convective heat transfer coefficient must be approximately 15 Btu/hr-ft^2. Table XXIV summarizes the best estimates of the fire source parameters.

The values in Table XXIV corresponds favorably with those estimated and utilized in Reference 9 for evaluation of thermal effects on tank cars.

Table XXIV

FIRE SOURCE PARAMETERS BASED ON ANALYSIS OF NOL DATA

Initial Total Heat Rate	40,000 Btu/ft^2-hr
Initial Convective Heating Rate	25,000 Btu/ft^2-hr
Initial Radiant Heating Rate	15,000 Btu/ft^2-hr
Convection Coefficient	15 Btu/ft^2-hr-^{o}F
Flame Temperature (JP-4)	1700^{o}F
Flame Emissivity	0.4

b. Thermal Effects

The thermal effects of the fire environment on the tank and lading were observed quantitatively by measurements of tank shell and lading temperatures as well as tank pressures. Two major items of importance were noted. First, the vapor in the vapor space above the liquid was considerably superheated at the existing tank pressure. The degree of superheat increased in proportion to the vapor space shell temperatures. There are counterbalancing effects of superheat in the vapor space, (1) more energy is absorbed per pound of material vaporized and superheated, and (2) less material can be expelled as a vapor through the relief valve with access to the vapor space or more liquid must be expelled through a valve communicating with the liquid. There is little doubt that vapor superheat exists in the tank and recent efforts have been directed to include the superheat effect in the thermal mathematical model which describes tank cars subject to fires. It must be noted, however, that the counterbalancing effects described above are nearly complete for propane. That is, the additional heat absorption in the superheat nearly counterbalances the reduction in flow associated with the vapor volume expansion. Thus, earlier simplified analyses are probably nearly correct for propane.

A second observation deemed of greater importance than that of the vapor superheat is the apparent existence of significant compressed liquid in the tank during the heating period prior to and during the initial valve discharge period. This effect was particularly evident in fire test No. 6 in which the initial discharge was presumably liquid inasmuch as the valve was at 90^{o} from the vertical. In that test, the pressure increase rate within the tank was several times greater than that expected with uniform temperature rise in the lading. This is illustrated in the following:

Total Mass of Lading In Tank	980 lb.
Pressure in Tank at 34.3 sec after Fire Initiation	160 psig
Pressure in Tank at Initial Opening Valve (44.9 sec after Fire Initiation)	270 psig
Total Exterior Surface Area	70.5 ft^2

If one were to assume saturation conditions to prevail throughout the heating period prior to valve action, the approximate change in internal energy of the liquid lading in the tank from 160 psig to 270 psig would be

$$\Delta u \approx 980\,(289.5 - 266.2)$$

$$\approx 19{,}300 \text{ Btu}$$

Further, this heat input would enter through, at most, the total shell area in a time of 10.6 sec. Hence,

$$q \geq \frac{(19{,}300)(3600)}{(70.5)(10.6)} = 93{,}000 \text{ Btu/hr-ft}^2$$

This value is more than twice the observed flux during other portions of the test and indicates that only about one half of the tank contents is heated and this in turn does not appear to mix sufficiently with the rest of the contents to transfer its heat of vaporization. The pressurization of the tank is, thus, produced with great stratification between vapor and liquid (approaching almost complete thermal separation). The effect of this stratification is to reduce the time to pressurization of the tank to the valve set-point. Subsequent effect is to reduce the required valve capacity while the remaining liquid is being heated to the saturation temperature. This effect is clearly evident in inspection of the valve action of test No. 6 where after the initial valve discharge at 44.9 sec, essentially no further valve action was observed for an additional 30 sec in which the remaining liquid contents were absorbing nearly the entire external heat load with possibly some minor (unobservable in the data) fluid efflux. The effect of stratification is also witnessed through inspection of the lading temperatures, whereby temperatures much below saturation for the prevailing pressure were observed in the liquid lading.

Because significant heat capacity remained in the compressed liquid at the valve opening during discharge of liquid in Test No. 6, it is to be expected that less fluid flow is required to reduce tank pressure than is necessary if the discharge had occurred at saturation conditions within the tank. Tank cars exposed to actual fire conditions may be preheated to saturation conditions by moderate heating. With full fire conditions then applied during discharge at saturation conditions, greater burden would be placed on the safety valve than produced in the 1/5 scale test. One therefore questions whether the 1/5 scale fire test as conducted is sufficiently representative, especially for nonvertical valve placement. Further testing is suggested to explore effects of preheating of the tank.

C. Valve Performance

The full sized A-3480 Midland safety valve at a discharge coefficient of 0.6 should be capable of discharging (full open) a minimum of 40 lb/sec of vapor or 100 lb/sec of liquid. Because the 1/5 scale tank contained nominally 1000 lb of material a maximum of 25 sec would be required for a complete discharge of vapor and 10 sec for complete discharge of liquid providing that sufficient heat input was available to support these loss rates.* When this is compared to the maximum times of 3000 and 1200 sec for the full sized tank car, one can readily appreciate the inherent safety factor in the 1/5 scale tests. Because the 1/5 scale tank was constructed with the same size wall thickness as the full sized car, but with only 1/5 the diameter, the strength and safety of the scaled car was further improved. Nonetheless, the performance of the valve can be partially assessed for those instances in which valve displacement and tank pressure data were obtained. Unfortunately, of the total of four propane tests conducted, only the last test provided reliable data concerning both valve displacement and tank pressure as well as discharge time. From this information, valve performance has been assessed in the following:

The data obtained from fire test No. 6 allows determination of approximate values of discharge coefficient for the Midland valve when discharging liquid or vapor. Several items of importance were obtained in this fire test of an uninsulated 1/5 scale tank car having the valve oriented at 90° from the vertical: first, the approximate tank pressure at which the valve opened; second, the approximate valve displacement during each valve action; third, the number of valve actions and the duration of each during the discharge period; and finally, a reasonable approximation of amount of material discharged during the discharge period and the distinction between liquid and vapor flow. Table XXV illustrates the best estimated values of the above items for both liquid and vapor flow. It is now of interest to evaluate the approximate discharge coefficients for the valve. The theoretical discharge rate for venting of liquid has been presented in Reference 9 (Figure 11) at a pressure of about 270 psia to be:

$$\dot{m}_L \approx 2800\ C_L A \qquad \text{lb/sec} \qquad (5)$$

For the valve, the discharge area is:

$$A = \pi D\ h \qquad \text{ft}^2 \qquad (6)$$

* A discharge time of one second was erroneously given in several of the NOL reports.

Table XXV

VALVE-RELATED TEST DATA

		Liquid Discharge	Vapor Discharge
1.	Approximate Average Tank Pressure During Valve Action	268-245 psig	230-200 psig
2.	Approximate Average Valve Displacement	0.25 in.	0.25 in.
3.	Number of Valve Actions	42	34
4.	Approximate Total Open Period	11.5 sec	30.75 sec
5.	Approximate Amount of Material Discharged	465 lb	430 lb
6.	Total Time to Discharge After Continuous Cyclic Valve Action	59 sec	200 sec
7.	Average Discharge Rate	40.5 lb/sec	14 lb/sec

where: D is the nozzle diameter (3.25 in.),
h is the valve displacement.

Combining Equations 5 and 6 we get:

$$\dot{m}_L = 2800 \; C_L \, \pi D \, h \tag{7}$$

BUT $h \approx 0.25$ in.
$D \approx 3.25$ in.

therefore $\dot{m}_L = 2800 \dfrac{\pi(3.25)(.25)}{144} \; C_L$

$$\dot{m}_L = 50 \; C_L \qquad \text{lb/sec, theoretical}$$

From the data (Table XXVI):

$$\dot{m}_L = \frac{465}{11.5} = 40.5 \text{ lb/sec}$$

Hence,

$$C_L = \frac{40.5}{50} = 0.81$$

The theoretical discharge rate during venting of vapor is again taken from Reference 9 (pages 96-97), and at an average pressure of 230 psia is:

$$\dot{m}_V = 865 \; C_V A \quad \text{lb/sec}$$

But

$$A = \frac{\pi(3.25)(.25)}{144}$$

Then

$$\dot{m}_V = 15.3 \; C_V \text{ lb/sec, theoretical}$$

From the data (Table XXVI):

$$\dot{m}_v \approx \frac{430}{30.75} = 14 \text{ lb/sec}$$

Hence,

$$C_v = \frac{14}{15.3} = 0.91$$

Evidently, relatively high discharge coefficients prevail during periods of low rates of discharge. Detailed inspection of the data observations of fire test No. 6 suggest that the above computed values of discharge coefficient may be higher than those which would actually be realized in a full-scale fire test. First, as mentioned earlier, it was found that during a major portion of the period of discharge of liquid, a majority of the tank contained liquid at less than the saturation temperature for the prevailing tank pressure. Hence, during this period, liquid under partial compression was being discharged. Equation 5 is based upon saturation conditions with likely increase in the amount of theoretical discharge expected during passage of compressed liquid due in part to less conversion to vapor. The net result is to estimate a lower theoretical flow than actually prevails and a corresponding increase in liquid discharge coefficient. Second, the rates of discharge of liquid are much less than would be expected for a full-sized tank. For this reason, the valve displacement was only a small portion of that required for a full-sized tank. Small displacements of the valve closure would place the restriction on flow near the seat of the valve where flow contours appear to be excellent and with little change in effluent flow direction. With greater discharge, the limit on flow becomes that of the valve orifice which may operate more nearly like that of a sharp-edged orifice. Furhter, the presence of the valve closure creates a significant flow obstruction in that a 90° change in the exhaust flow is required for exit from the valve. This might severely limit actual flow by introducing completely irreversible flow patterns and large momentum changes.

Relative to vapor discharge, it must be noted that the second of the above explanations also applies to it. It is reasonable to believe that discharge coefficients approaching unity can be obtained for vapor discharge, especially during use at far from maximum flow conditions. Once maximum flow conditions are realized, it is to be expected that the flow coefficient would be reduced. One must observe as well that the measured pressure range of operation of the valve during vapor discharge (Table XXV) was much below that expected. This might indicate either a change in the pressure-output characteristics of the pressure transducer due to heating or a difference in the valve construction. Either or both of these, if erroneous, could partially account for the high discharge coefficients derived from the data.

The above comments on possible extraneous effects on computer discharge coefficients are intended to caution the reader with respect to the accuracy of the coefficients derived. For the most part, such extraneous effects would be expected to be minor. Therefore, the analysis of data from fire test No. 6 seems to indicate relatively good valve functioning during small valve displacements. Yet to be established is its operation at conditions approaching maximum displacement and the pressure range over which this displacement takes place.

Valve performance investigations would be improved while utilizing the 1/5 scale tank if a smaller safety valve were to be tested. There is, in fact, sufficient justification for such a test. One notes that insulated 105A series tank cars subject to fire conditions have been observed to undergo catastrophic rupture which perhaps is attributable to an underdesigned safety valve. Inasmuch as the valve on the insulated cars has approximately one-tenth the capacity of the A-3480 valve and is of equivalent design, it is suggested that this valve be tested in carefully controlled 1/5 scale fire tests in an attempt to generate data more nearly approaching maximum valve design requirements. From this information, it is reasonable to expect that valve deficiencies, if any, might be observed and improvements implied.

d. Thermal Shield Effectiveness

The effectiveness of thermal shield materials tested, namely the

NOL insulation system, and the Albi 89X mastic with Albi 144H overcoat was found to be excellent for both materials. Reductions in heat input to the lading approaching 10:1 were observed. On a thickness basis, the coating effectiveness was about the same for each type of coating, although a greater reduction in total heating was obtained with the thicker (7/16-inch) NOL insulation system. Significant reduction of heat input to the lading through the use of an insulative mastic coating of the order of 0.3 inch thick would be expected, as illustrated in the following simplified analysis.

The effective combined radiant-convective coefficient for the fire-tank combination was about:

$$h_{comb} = \frac{40,000}{1700} = 23.5 \text{ Btu/ft}^2\text{-hr-}^\circ\text{F}$$

Considering the conductance of the coating to be given by:

$$h_{coating} = k/\delta$$

Where k is the thermal conductivity of the coating; δ is the coating thickness. The composite or net effective heat transmission coefficient is given by:

$$h_{eff} = \frac{h_{comb}k}{k+h_{comb}\delta} = \frac{23.5k}{k+23.5\delta}$$

The ratio of heat input to the lading for an insulated versus a non-insulated tank is given by

$$q_{ins}/q_{non} = \frac{h_{eff}}{h_{comb}}$$

$$q_{ins}/q_{non} = \frac{k}{k+23.5\delta}$$

The thermal conductivity of the insulation is not known with any degree of precision but for reasonable insulation would be expected to range from 0.1 to

0.3 Btu/ft-hr-^{o}F. For an insulation thickness of 7/16 inches and assuming no intumescence, we get

$$0.105 \leq q_{ins}/q_{non} \leq 0.26$$

Hence, substantial reductions in heating should be expected if the insulation remains in place. The apparent success of the insulation in the 1/5 scale tests indicates that a substantial portion of the insulation must have been effective throughout the test period. Post-test descriptions of the quality of the insulation residue are, however, lacking in the reported results. It is suggested that post-test data relative to insulation strength be gathered in future fire tests.

D. Full Scale Fire Test

Ballistic Research Laboratories conducted a full scale fire test of a 112A (uninsulated) tank car containing LPG.[27] (An insulated tank car test was subsequently conducted but results were not available for this report.) The car was a 33,000 gallon 112A340W modified with an extra, remotely operated valve intended for excess pressure relief. The tank car was located in a 150 ft. x 100 ft. x 26 ft. excavation. An 80 ft. x 30 ft. dike under the car contained the JP-4 jet fuel for the fire. The safety relief valve was vertically up and communicated with the vapor space throughout the test. Temperatures and pressures were measured within the tank throughout the test. The test data has not been examined in great detail but several conclusions can be stated based on a preliminary examination of the data even though there are a number of anomalies in the data. Preliminary observations indicate:

1. The tank did not become shell full prior to initial valve relief.
2. Heat flux to the tank was of the order of 25,000-35,000 Btu/ft^2-hr to the wetted surface or roughly 4 times that assumed in the AAR Specifications for Tank Cars determined from $Q = 34{,}500A^{0.82}$.

3. Tank rupture occurred at 24.5 minutes after fire initiation with about one-half the lading still in the car.

4. Based upon time and propane loss, it appears that the valve on the average was discharging 50-70 lb/sec. This is approximately the value estimated for the valve using a discharge coefficient of 0.8 for vapor flow.

5. The relief valve had insufficient capacity to limit the tank pressure to 306 psig as required by the Specifications or to below the tank test pressure (340 psig).

6. At the time of rupture, the peak indicated tank shell temperature was about 1200°F with a tank pressure of about 335 psig.

7. Tank rupture occurred at about pressure-temperature conditions indicated by uniaxial strength data for the TC 128 steel as given by Figure 8.

8. The tank pressure was found to change from 125 psig to about 270 psig in a period of about 125 sec, after which relief valve action was observed. Had saturation conditions prevailed within the tank through this initial period, the required heat flux to the tank would have to have been about 60,000 Btu/ft^2-hr. Because this is much in excess of other more reliable heat flux indicators, one concludes that non-equilibrium conditions were present during the initial stages of heating due to temperature stratification within the liquid lading and/or the presence of non-condensibles in the vapor space.

9. Generally, saturation conditions prevailed within the tank during a majority of the test duration, after valve action.

E. Effect of Insulation on Safety Relief Valve Sizing

Safety relief valve sizing as given by DOT Regulations are based on equations derived in AAR Specifications for Tank Cars, effective October 1, 1972. The equations are based on gaseous flow and an average heat input to $34,500A^{0.82}$

for an uninsulated car, both of which are not believed to be the limiting conditions as discussed in Reference 9. However, it may be instructive to compare the valve sizing requirements as given in the AAR Specifications for insulated and uninsulated tank cars.

The formula given in the Specifications for the required flow of a valve on an uninsulated car is $Qa = GuA^{0.82}$ (Formula A8.01(b)) where Qa is the required air flow in cubic feet per minute at standard conditions, Gu is a constant for each commodity at the flowing conditions, and A is the outside area of the tank in ft^2. The value of the constant Gu is tabulated in the Specification. For LPG with a flow rating pressure of 306 psig (112A340W and 114A340W cars), Gu is 54.98. The outside area of a 112A340W car of about 33,000 gallons capacity is about 2000 ft^2. Therefore, Formula A8.01(b) indicates that the required air flow is $Qa = (54.98)(2000)^{0.82} = 27{,}980$ scfm.

For insulated cars, the Specification requires less valve capacity because of the reduced heat input to the car. The resulting value of required air flow is given by $Qa = 2GiUA^{0.82}$ (Formula A8.02(b)) where Gi is a constant for each commodity at each flow condition for insulated cars, and U is the insulation conductance in Btu/hr-ft^{2}^{o}F. For LPG with a flow rating pressure of 306 psig, Gi = 6.733. The conductance, U, is limited to a maximum of 0.075 Btu/hr-ft^{2}^{o}F in the Specification. Therefore, the required valve capacity for an insulated 33,000 gallon 112A340W car would be

$$Qa = 2(6.733)(0.075)(2000)^{0.82} = 514 \text{ scfm}$$

Insulation is actually not permitted on 112A cars by the DOT Regulations but it is interesting to note the large reduction in flow capacity that is calculated using the formulas in the Specifications. The reduction in valve capacity is 54 to 1.

DOT 105A300W cars are insulated and utilize valves which meet the criterion of Formula A8.02(b). An 11,000 gallon car has an outside area of about 1000 ft^2. For this car carrying LPG, Gi is 6.442 and the required air flow is

$$Qa = (2)(6.442)(0.075)(1000)^{0.82} = 278 \text{ scfm}$$

In practice a valve of larger capacity than required is actually used. For 105A300W cars, valve no. A-1247 made by Midland Manufacturing Company has been used. This valve has a nominal capacity of 3070 cfm. That is, this valve has an excess capacity according to formula A8.02(b) of 3070/278 = 11/1.

The Midland valve used for 33,000 gallon 112A340W cars has been valve No. A-3480 with a nominal capacity of 36,640 cfm. This valve has an excess capacity according to Formula A8.01(b) of 36,630/27,980 = 1.3/1. Perhaps the larger safety factor of the valves used on the insulated 105A300W cars partially accounts for the lower incidence of ruptures for these cars during fire exposure than for uninsulated 112A340W cars.(See Appendix C - in which it is shown that during the time period 1965-1970, 3 of 20 (15%) insulated cars which had been exposed to fire eventually ruptured and of 55 series 112A/114A cars exposed to substantial fires, 50 ruptured (91%).

The validity of the valve sizing formulas has been questioned previously (Reference 9). For example, instead of the average heat flux being proportional to $A^{0.82}$, it has been shown that the heat flux is more nearly proportional to A. With this being the case, Formula A8.01(b) should be modified for gas flow to be Qa = GuA. The required air flow for a valve on a 33,000 gallon 112A340W car would then be

$$Qa = (54.98)(2000) = 110,000 \text{ scfm}$$

and Midland valve No. A-3480 would be severely underdesigned. Of course, liquid flow requires even larger valve capacity as discussed in other portions of this report and in Reference 9.

Similarly, for an insulated car Qa = 2GiUA and the required flow for an insulated 11,000 gallon 105A340W car would be:

$$Qa = (2)(6.733)(0.075)(1000) = 1010 \text{ scfm}$$

and Midland valve No. A-1247 would have sufficient capacity for this vapor flow by a factor of 3070/1010 = 3/1.

It has been shown, Reference 9, that the ratio of required liquid to vapor mass flow rates for an overturned car is

$$\frac{M_{L,MIN}}{M_{V,MIN}} = \frac{V_g}{V_f}$$

where $M_{L,MIN}$ is the minimum required liquid mass flow rate, $M_{V,MIN}$ is the minimum required vapor mass flow rate, and V_g and V_f are the specific volume of gaseous and liquid propane, respectively, at the discharge conditions. At the discharge conditions of 105A300W tank cars, the flow ratio is

$$\frac{M_{L,MIN}}{M_{V,MIN}} = \frac{0.370}{0.0361} = 10.2 \text{ (Required ratio)}$$

Reference 9 also includes the calculation procedure for determining the liquid and vapor flow capacity of any particular valve. For a valve on a 105A300W car with equal liquid and vapor flow coefficients, the ratio of liquid to gas flow has been calculated to be

$$\frac{M_L}{M_V} = 2.4 \qquad \text{(Capacity ratio)}$$

It was shown above that the A-1247 valve is overdesigned for vapor flow by a factor of 3/1. Therefore, the amount of liquid flow capacity of valve A-1247 is (2.4)(3) = 7.2 times the required vapor flow. However, the required liquid flow is 10.2 times the required vapor flow (see above). Therefore, the valve is underdesigned for liquid flow when installed in an insulated 105A340W car by the ratio 7.2/10 = 0.72. In the above analysis, the Specification values of the effectiveness of insulation have been used even though Reference 9 indicated several reasons why the insulation may not be as effective as the Specification suggests, particularly due to increase in insulation conductivity at elevated temperatures. However, the above analysis does suggest some possible problem areas. The calculations show that present valves used on 112A340W tank cars do not have sufficient capacity to prevent pressure buildup for either liquid or vapor flow. Also, the valve on the insulated 105A300W car has sufficient capacity for vapor flow but insufficient capacity for liquid flow. These calculations tend to be verified by the fact that in actual service when exposed to fires, 15% of the insulated 105A cars ruptured and 91% of the 112A's ruptured.

Safety Valve Sizing

For every safety valve there is a corresponding maximum heat input rate to a tank for which the safety valve can pass sufficient commodity to prevent pressure buildup. The equations for estimating this heat input rate are given in Reference 9. The allowable heat input rate is based on liquid flow through the valve. The governing relation for this circumstance is:

$$Q = \frac{M_L L}{(V_g/V_f - 1)} \qquad (8)$$

where Q is the total heat input rate, M_L is the mass flow rate through the valve, and L, V_g, and V_f are the latent heat, and specific volumes of the gas and liquid at the internal tank conditions. For LPG, the mass flow rate is given by:

$$M_L = 3000\ C_L A_V \quad \text{lb/sec} \qquad (9)$$

where C_L is the liquid flow coefficient through the valve, and A_V is the valve area in ft^2.

Combining Equations (7) and (8), the maximum heat input rate that will not cause increasing tank pressure is given by:

$$Q = \frac{3000\ C_V A_V L}{(v_g/v_f - 1)} \qquad (10)$$

For a safety valve for a 112A340W car, C_L has not been precisely determined. RPI/AAR Phase 6 tests for water flow indicate a value of only 0.2. We have calculated for test 6 of the 1/5 scale fire tests conducted by the Naval Ordnance Laboratory with liquid propane flowing that C_L is 0.81. The maximum valve flow area is 0.057 ft^2. At the maximum allowable pressure of 306 psig (DOT Specification for Tank Cars, 179.102-11), V_g is 0.305 ft^3/lb, and V_f is 0.0375 ft^3/lb, and L is 111 Btu/lb. Therefore, by Equation 10, assuming C_L = 0.81, the allowable heat input rate is Q = 2060 Btu/sec = 7,420,000 Btu/hr. The above heat input rate corresponds to an average flux over the whole car (2000 ft^2) of 3710 Btu/hr-ft^2. This is in contrast to the heat flux of about 35,000 Btu/hr-ft^2 that has been measured from a fire. That is, the valve may be underdesigned by the ratio 35,000:3710 = 9.4:1 for liquid flow under the criterion of no pressure increase above 306 psig.

G. Review of Edwards Air Force Base Valve Test Facility

The Edwards Valve Test Facility is a facility for measuring flow characteristics of the tank car relief valve that should permit excellent control of the conditions of each test. However, the possible range of the conditions are limited. The arrangement represents a compromise between reservoir volume and capacity for heating the propane. First, the steam generating plant, while of sufficient capacity (100 HP) to heat the lading to cause valve opening in a reasonable time, cannot approach the heat input/pound of lading that can be produced by a fire because the reservoir volume is so large. Consequently, blowdown runs are planned for determining flow characteristics. These will be initiated by release of a hold down latch. Valve functioning tests are also planned whereby the valve will be allowed to open by itself, but the extent of opening achieved (the displacement of the valve) will not be as great as in a large fire.

A depth of 2 ft. of liquid is required to cover the steam pipes of which there are 16, and they extend the full length (54 feet) of the tank. The upper flow nozzle extends down from the top of the tanks about 18 inches so that the working range of depth change is about 3 feet (the tank is 84" in diameter).

The flow indicating nozzle has an 18" diameter throat with four pressure taps around its periphery. Very sensitive transducers are connected to the taps with a short run of tubing. The recorder for this transducer has extremely high resolution (1 part in 10^5 over a range of 2 psi). If this proves to be the case and if damping of the system is good subsequent to valve opening, good measurements of flow rate should be possible in spite of the low pressure drop of the nozzle (1/2" of water for maximum flow rate). However, we believe that a greater probability of success would be obtained in the measurement of flow rate by using a smaller flow nozzle. This would relax the requirement for a highly sensitive and delicate transducer. For example, the maximum flow rate through a nozzle having a 5-1/2 inch diameter throat would produce a manometer deflection of 45 inches of water. (The diameter of the valve housing at its entrance is 5-1/2 inches.) The maximum nozzle diameter that is recommended for good practice is 10 inches, which would yield a manometer reading of 4 inches of water at the maximum flow rate. Use of this nozzle would greatly facilitate measurement of pressure, especially at conditions leading to partial valve opening. The pressure change would have to be much greater than 4 inches of water to produce any detectable effect in the quality of the vapor i.e., the amount of condensation would be negligible when the pressure is decreased from 290 psia by several inches of water.

Installation of load cells was investigated as a means of providing redundancy in measurement of mass flow rate. After discussing the difficulty of installation and the limitations of load cells (considerable experience has been accrued at Edwards Air Force Base on their rocket engine test stands and then have a very large (10^6 lb) calibration facility), it was decided that this was impractical in spite of the low probability of success anticipated for the present flow rate measurement.

Data recording and processing facilities are excellent and impose no limitations on the quality of data to be expected. Calspan suggests, however, that tests be run on smaller commercially available valves. This would permit investigation at conditions approaching maximum flow and for substantial test duration.

V. AREAS REQUIRING FURTHER RESEARCH

A. Tank Car Thermal Environment

1. External

Continued investigation using small models and full scale tank cars is required to define the effects of external fires on the tank insulation, structural material, and lading. In particular, the effects of localized heating at fluxes as high as 150,000 Btu/ft^2-sec should be investigated. Research should be carried out to determine if a coating and/or insulation can be found which, when applied to the 112A/114A car, will be cost effective. If no cost effective solution is available, research into minimizing the cost of applying insulations should be undertaken. The application of an insulative coating to the 112A/114A type car is one plausible solution which may have the greatest payoff in terms of reducing catastrophic type failures.

2. Internal

Investigations should be continued into defining the thermodynamic state of the lading during heating. For instance, the present sequence of tests indicates that lading does not heat up uniformly. The sequence of events which occurs during heating has an effect on the type of pressure relief device which should be specified for tank cars. The tank car thermal model should be modified to account for this non-uniform lading temperature.

B. Valve Designs

The 1/5 scale tests and the full scale tests at White Sands tested the safety relief valves under the most adverse conditions. Most of the testing has been conducted with the valve flowing vapor whereas the condition of the valve flowing liquid presents the most severe condition. The operation of the present valve has not been fully determined under flashing liquid flow conditions. Do the valves pop open or do they act as proportioning valves, slowly opening?

What are the ramifications and expected effectiveness of specifying latching lading-temperature-actuated valves? A low lèvel of heating followed by a sudden burst of heating may cause vaporization of the lading at a rate higher than the valve is capable of discharging. Different heating schedules may affect the valve's ability to protect the tank from rupture. Continued testing with different heating schedules should be carried out.

C. Couplers and Head Shields

No full scale crash tests have been conducted to define the performance of Modified E and F couplers in the derailment situation. There is, however, an RPI/AAR "Railroad Coupler Safety Research and Test Project" which has proposed some laboratory test work. To what extent this will include modified coupler and full scale testing is not known.

Optimization of head shield location, size and thickness have yet to be accomplished. Full scale tests and/or computer simulation are needed to define the effectiveness of modified couplers and head shields in preventing head punctures. These problems should be investigated analytically and with full scale confirmation tests.

D. Car Structural Design

The number of ton-miles carried by all freight cars per year has generally been increasing over the last ten years (Reference 28). This is particularly true of the new tank cars which have entered service during this period. Further, the speed at which trains are being run is also increasing. As a result, the loading and stress level at which tank car components are expected to operate have increased significantly and failures due to fatigue and general overloading are increasing.

The design manuals used to design tank cars (References 29 and 30) are generally based on static-elastic types of analysis. It is recommended that a thorough review of tank car design be made which will utilize the refinements developed in the last ten years. Such a review would include:

1. Structural analysis of the tank using finite element techniques.

2. Estimation of fatigue life of each critical tank car component and a comparison with present life cycles to determine weak points.

3. Experimental determination of force levels developed by in-service tank cars.

4. Analysis of the experimental data generated above by statistical methods so that fatigue factors and component loads limit can be set on a rational basis.

5. Review of quality control and manufacturing procedures.

E. Thermal Shields

A considerable amount of research work is being invested in developing coatings which can be applied to tank cars as thermal shields. The majority of this research is concerning itself with the thermal properties of the coating, while its ability to withstand ten years of railroad type service has not been proven. Recent experience with painted tank cars has indicated that organic coatings exposed to railroad service are not performing as expected. The economics of applying thermal shields to tank cars will be greatly affected by its ability to withstand the railroad environment. Evaluation of a coating's ability to withstand this environment should be included in the screening process.

Alternate methods of constructing thermal shields should be investigated. Construction similar to that now used on the 105A series cars should be considered for the 112A/114A type car. This type of construction may not be strictly cost effective but because of the large increase in the time to rupture produced by even a small amount of insulation, this car modification should be given serious consideration.

F. Accident Statistics

The compilation of accident statistics should be continued so that more complete and more recent data are available for the evaluation of proposed modifications. Particular effort should be placed on the correct estimation of losses suffered by non-railroad connected persons including injuries and deaths and damage to property. A small number of large loss accidents include a major portion of the total losses and because of their influence on overall loss figures, a proportionate amount of effort should be expended on obtaining loss data for these accidents. Re-evaluation of past accidents should be a continuing effort so that as more information on losses becomes available, such as settlements of litigation, it can be included in the loss estimates.

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APPENDIX A

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APPENDIX B

RE-EVALUATION OF LOSS DATA FOR FIVE ACCIDENTS

Calspan has re-evaluated the loss data for five accidents chosen to be representative of a range of dollar losses per accident. Columns 1 and 2 in Table A-I show a comparison between the RPI/AAR values used in their cost-benefit analysis and the re-evaluation. The bases for these figures are discussed in subsequent paragraphs.

Both the RPI/AAR and Calspan estimates are based on those costs of the accident traceable to the presence of tank cars in the accident. Tank cars did not initiate the accidents examined, and therefore, the costs shown are less than the total cost of the accident. The RPI/AAR totals do not include the cost of mechanical damage to tank cars that lost lading on the assumption that the cost to repair a mechanically damaged tank car is the same, whether or not it has been modified. The Calspan reassessment accepted this assumption unless a high cost car of unusual construction (e.g., 113 series double-wall car) was involved. For the special case, a differential value was used (reduction in damage to the car due to a modification less the cost of modification replacement). Totals for lading losses were in reasonable agreement. Calspan included a billing increment added to FOB carload prices. RPI/AAR values for lading were high enough to include some transport increment in some cases, although this is not so stated. Estimates on thermal damage to tank cars stated in the RPI/AAR report[23] were used by Calspan.

The principal cost differences in the reassessment show up in what the RPI/AAR report termed the "other loss" category. This category has the highest dollar loss when considering the total accidents in the 1965-1970 base period and contains the pivotal figures in the RPI/AAR cost/benefit analysis. Before proceeding further with general discussion, we will consider the five specific cases in Table B-1.

Table B-1
COST OF ACCIDENT COMPARISONS BETWEEN SELECTED SOURCES

Accident Location and Date	Cost and Source					
New Athens, Ill. - 4/9/70	$ 84,000[1]	$ 128,000[2]	$ 450,000[3]			
Armitage, Ohio - 4/25/70	$ 4,800[1]	$ 11,100[2]	$ 66,000[4]			
Crescent City, Ill. - 6/21/70	$1,900,000[1]	$2,200,000[2]	$6,800,000[5]	$1,500,000[6]	$1,700,000[7]	$3,000,000[8]
South Byron, N.Y. - 8/27/70	$ 119,000[1]	$ 146,000[2]	$ 121,000[9]			
Crete, Neb. - 2/18/69	$2,000,000[1]	$2,000,000[2]				

NOTE: Cost from [1] and [2] include costs derived from loss of lading from tank cars only, not entire cost of accident. Other sources are derived on different bases--see text.

1 RPI/AAR Tank Car Safety, Research and Test Project Report Nos. RA-02-1-10 and RA-02-2-18.
2 Calspan Review.
3 FRA Office of Economics estimate
4 Railroad "Unusual Occurrences Report" with appended costs plus cost of lading - RPI-AAR General File.
5 Calspan estimate including projected lost business receipts discounted to present worth.
6 TP & W R.R. claims estimate - August 1971, as reported in St. Louis Globe Democrat.
7 National Transportation Safety Board Release - 13 July 1972.
8 National Fire Protection Association - Fire Journal, November 1970.
9 AAR Bureau of Explosive Tank Car Incident Report (Form 25-T-1).

Crete, Nebraska - 18 February 1969

This accident produced a very high death toll (nine) and numerous serious injuries. Property damage assessable to lading release was slight. Death and injury resulted from the rupture of a car of anhydrous ammonia in a standing train struck by a derailing train on an adjacent track.

The RPI/AAR figures were apparently derived principally within the project group, rather than from railroad figures. Even though more than four years have elapsed, litigation from this accident continues. Therefore, it is not surprising that "official" estimates were not available. According to sources contacted by Calspan, settlements on the order of three-quarters of a million dollars have been paid to date, with the largest single settlement to date being $290,000. Three major damage suits are just coming to trial, Calspan assessment is based on projected future earnings of victims discounted to a present sum plus estimated injury settlements. The total projected earnings were relatively low for an accident with the number of deaths (nine) involved, due to the fact that three victims were transients for which no earnings were projected.

New Athens, Illinois - 9 April 1970

Thirty-three cars, including a variety of liquid chemical cargo, derailed in this community of 2000. Fire from a punctured vinyl chloride car caused failure of four loaded, nonpressure cars. The town was evacuated, and the water supply shut off for fear of contamination from leaking chemical cars. Ten fire departments responded to the scene, including a foam truck from a nearby air base. This accident was extensively covered by the RPI/AAR project group.

The Calspan assessment for lading loss is higher than that shown in Reference 1. There was no gross lading release from one car of vinyl chloride; however, the car was considered to have been heated sufficiently to impair the quality of the lading. Therefore, the lading was presumed "lost" in the Calspan estimate even though the tank shell maintained its integrity. Calspan used replacement cost figures for homes and vehicles destroyed. Cost of evacuation, manhours expended by public safety personnel, and loss of earnings resulting from temporary evacuation of businesses were utilized in the Calspan estimate, which was 52 percent greater than that estimated by RPI/AAR. The $450,000 estimate from FRA is believed to be a direct loss estimate for the total 33-car derailment and therefore not directly comparable.

Armitage, Ohio - 25 April 1970

This was a small accident involving double walled DOT 113 series cars containing liquefied ethylene. The railroad requested and received technical assistance from Union Carbide (shipper) personnel sent to the scene from Charleston, West Virginia.

For this special car type, Calspan considered that a mechanical damage assessment should have been made. A cost increment of $5000 was added as the difference in repair cost between a modified and unmodified car. Travel expenses for Union Carbide personnel were also considered as an assessable accident cost. The total Calspan estimate was a 131 percent increase over the RPI/AAR estimate. Direct losses for the total accident were $61,000 in equipment destroyed plus the cost of the lading of one car.

South Byron, New York - 27 August 1970

Twenty-eight cars of a Penn Central freight derailed in an unpopulated area of western New York. One car of vinyl chloride was punctured and caught fire. A second car violently ruptured. A third car of vinyl chloride adjacent to a burning boxcar received moderate fire damage. The threat of explosion of this car and the presence of a fourth pressure car which had been dented slowed up clean-up operations on the heavily traveled two-track main. Sixteen hours elapsed until limited service was restored. FRA, AAR, state officials, and a Calspan representative were at the scene.

The Calspan reassessment is approximately 23 percent higher than the RPI/AAR loss calculation. The principal difference lies in the fire damage estimate to other property. Note 10, page A-45, of Reference 23 is incorrect in stating fire damage was confined to the tank cars. Five box cars burned, two of which were not damaged heavily in the initial derailment. The lading in at least two of these cars was destroyed by fire, one containing canned goods and a second with fiber-bituminous pipe. Reference 1 does show a $51,000 "other" loss, but this does not cover fully the loss attributable to loss of a tank car lading.

Crescent City, Illinois - 21 June 1970

Crescent City was a major disaster resulting from fire and explosion of LPG cars involved in a derailment within the city limits. The business district of the small town of 650 persons was virtually destroyed, in addition to 25 residences. Sixty-six persons were injured, but no deaths resulted.

The RPI/AAR project made an extensive investigation of this accident and made several revisions in their cost estimates. The $1.9 million total is published in August 1972 reports.[1,23] Components of the Calspan $2.2 million estimate included the replacement value of homes and personal property destroyed, business property destroyed (including equipment and inventory), lost business receipts in the immediate post-accident period, and home-owner insurance living expenses for temporarily displaced persons. The TP & W railroad figure of $1.5 million for the accident is presumed to include only damage claims paid. The source of the $1.7 million NTSB figure is unknown, but it may have come from Reference 23. The $3.0 million estimate by the NFPA, an experienced assessor of damage from fire, was a total loss figure.

The same Calspan representative who visited Crescent City at the time of the accident recently revisited the scene to assess recovery of the community. Crescent City now has a modern business district but, unfortunately, much reduced in size from that before the accident. Only about half of the business establishments returned. It may be instructive to examine some of the cost potential in lost business to the community of Crescent City. For example, if eight permanently lost establishments in Crescent City had average annual gross receipts of $50,000 over a projected lifetime of 20 years, accumulated losses of $8,000,000 would accrue. Discounting this sum at 6 percent to a present value of $4,600,000 and adding the previously estimated $2,200,000 direct loss gives an estimated total loss of $6,800,000. This might be considered to be the loss to Crescent City but it is not necessarily a loss assignable to the accident because presumably the businesses would move to a different location.

APPENDIX C

HISTORICAL LOSSES OF INSULATED AND UNINSULATED TANK CARS DUE TO FIRE

As an aid to estimating the efficiency of thermal shield coatings for reducing losses due to fire, this Appendix examines the losses that have been experienced historically by 105A (insulated) tank cars in comparison with 112A/114A (uninsulated) tank cars.

Prior to the introduction of the 112A/114A series tank cars, series 105A tank cars were utilized for compressed gas service and 105A cars are still utilized for this service. The 105A cars consist of an inner tank which is covered with insulation. The insulation is covered with a metal jacket. The insulation is required to "be of sufficient thickness so that the thermal conductance at 60°F is not more than 0.075 Btu per hour, per square foot, per degree F temperature differential" (49) CFR 179.100-4) but is otherwise unspecified. Typically, the insulation is rock wool, glass wool, cork, or a foamed in place synthetic material. The major concern of this report is insulators which are coated directly on the tank shell without an outside metal jacket. However, the purpose is thermal insulation similar to that of 105A cars. It is informative to look at the history of losses of 105A insulated cars compared to 112A/114A uninsulated cars that have been exposed to fires.

Insulated cars are allowed to have smaller safety relief valves because of the reduced heating load through the insulation. For safety valve sizing, the assumed heating load is increased over that through an insulator of 0.075 Btu/hr ft^2°F because of the increase in conductivity at elevated temperatures, the possibility of losing insulation in an accident, and the heat transferred through connections and fittings. The sufficiency of the assumed increased heating has been found to be somewhat dubious (Ref. 9, p. 47). That is, the supposed large safety factor in safety valve sizing for insulated cars (105A's) may not exist. However, in actual practice, the valves that have been used for uninsulated 112A/114A cars are only 30 percent larger than the minimum allowable according to the specifications whereas the valves that have been used for insulated 105A cars are 11 times the minimum allowable (see Section IV-E). If insulating coatings were put on 112A/114A cars without changing the valve from the one used for the uninsulated cars, then the valve would also be substantially oversized compared with the specified minimum allowable. Therefore, looking at the efficiency of the thermal shield on 105A cars would appear to be indicative of the efficiency that could be expected for one particular specification of thermal shield on 112A/114A cars with the present safety relief valves.

Reference 23 gives the RPI/AAR data on loaded tank cars exposed to fires for 1965 through 1970. "Exposed to fire" is defined as suffering visible fire damage, i.e. at least blistered paint. Loaded tank cars includes all cars "which were known to have been loaded when exposed to fire as well as those where it was not known whether the tanks lost lading prior to the fire exposure due to a puncture in the initial accident. Tank cars punctured initially are excluded only if they were known to be essentially empty when later exposed to fire." Thirty series 105A cars were reported to have been exposed to fire. Of these, 26 lost all of their lading due to fire including 9 that ruptured. Further examination of the data in Reference 3 indicates that 5 of the cars exposed to fire were 105A100W's containing ethylene oxide and all five of these cars ruptured (actually stated as exploded in this instance). Ethylene oxide may polymerize when heated. It is not a commodity that is shipped in 112A/114A cars. LPG is the major commodity shipped in 112A/114A cars and it does not polymerize. Commodities that may polymerize and which are commonly shipped in both 105A and 112A/114A cars are vinyl chloride and butadiene. However, explosive polymerization in tank car fires of these commodities is rare, whereas it is not in the case of ethylene oxide. Therefore, 105A100W cars loaded with ethylene oxide should be eliminated from historical data on the efficiencies of thermal shields considered for 112A/114A cars. In addition, one of the cars that ruptured was actually an ARA V series car which is a predecessor of 105A series cars and probably not indicative of modern insulated car technology. Eliminating the five 105A100W cars loaded with ethylene oxide and the ARA V car leaves 24 of the series 105A cars which were exposed to fire. Twenty of these lost all of their lading, including 3 that ruptured. Of the three that ruptured; one was loaded with anti-knock compound and had been heated an unknown time before rupture, one was loaded with butadiene and had been heated for 8 hours and 52 minutes, and one was loaded with vinyl chloride, and had been heated for 10 hours and 15 minutes. Valve operation or lack of operation is not known for two of these ruptures but in the third, the valve was known to have remained closed, an anomaly which may indicate faulty valve operation and which may have influenced car rupture. Also, none of the ruptured cars contained LPG, the major commodity shipped in 112A/114A cars and formerly shipped in great quantities in 105A series cars. In any event, the vast majority of 105A cars did not rupture during exposure to fires.*

By comparison with 105A series cars, of the 65 112A/114A series cars exposed to fires, 56 lost their lading including 50 that ruptured. Of the 15 cars without ruptures, 7 did not vent at all, i.e., they must have been very small fires as the contents could not have heated to even 115^{o}F, the maximum summer loading condition. For three cars, it is not known if they vented or not and it was assumed by RPI/AAR that one or more of these were punctured in the initial accident. Eliminating the 7 cars in which there was no venting and the 3 cars for which venting was not known, leaves 50 ruptures of 55 cars exposed to fires which were at least of sufficient severity to cause venting.

In summary, during the time period 1965-1970, 3 of 20 (15%) insulated cars which had been exposed to fire eventually ruptured and some of these ruptures may have had safety equipment failures. Of 55 series 112A/114A cars exposed to substantial fires, 50 ruptured (91%). It is evident that 105A cars are very much less apt to rupture on fire exposure than 112A/114A cars. This information suggests that the insulation combined with safety relief valves larger than called for in the Tank Car Specifications result in cars less prone to rupture during fire involvement.

*In the decades prior to 1965 some 105A cars loaded with LPG have ruptured when exposed to fire, but it has been a rare event. The Shattuck, Oklahoma, accident of March 4, 1958, being about the only notable major accident.

APPENDIX D

MATHEMATICAL MODEL OF A TANK CAR EXPOSED TO FIRE

1. Fortran Nomenclature

Input Nomenclature

1. Input Specific to Lading

ALPHA	Thermal expansion coefficient of liquid	cu, ft/^oF
GAMMA	Ratio of specific heats	
GASCON	Gas constant	
HGT	Gas heat transfer coefficient for internal tank car environment	Btu/ft^2-hr-^oF
HLT	Liquid heat transfer coefficient for internal tank car environment	Btu/ft^2-hr-^oF
HFT	Specific enthalpy of saturated liquid	Btu/lb
LT	Heat of vaporization	Btu/lb
MTOT	Total mass in tank car per unit length	lbs/ft
PLT	Pressure values for enthalpy and volume data	psi
TLT	Temperature values for enthalpy and volume data	^oF
TS	Sonic temperature	
VFT	Specific volume of saturated liquid	cu.ft/lb
VGT	Specific volume of saturated vapor	cu.ft/lb

2. General Input

A	Relief valve flow area	ft^2
ANG	Angle values for HEATX data	degrees
CINS1,2	Slopes of lines describing variation of thermal conductivity with temperature. 1 refers to outer layer, 2 to the inner one.	

CP	Specific heat of tank car shell material	Btu/lb-^{o}F
CD	Relief valve flow coefficient	
DELX	Longitudinal element size	
DELTA	Computing interval	sec
EI	Emissivity of inside surface of tank car shell	
EMO	Emissivity of outside surface of tank car shell	
FK1,2	Thermal conductivity of insulation at reference temperature	Btu/ft-hr-^{o}F
FKS	Thermal conductivity of shell material	Btu/ft-hr-^{o}F
HEATX	Heat transfer coefficient external to tank car	Btu/ft^{2}-hr-^{o}F
LAGTH1,2	Thicknesses of insulation layers, 1 refers to the outer layer, 2 to the inner one	inches
NEL	Number of tank car shell elements around circumference	
NRAD	Control number for circumferential element temperature plots. NRAD = 1 means a plot will be generated for every element, NRAD = 2 will produce a plot for every second element, etc., starting with the first one	
NX	Number of elements along tank car (longitudinal)	
PBT	Burst pressure of tank	psi
PITCH	Pitch angle defining tank car attitude	degrees
PR	Reliev valve opening pressure	psi
PRL	Relief valve closing pressure	psi
RHOSK	Density of protective skin over insulation	lb/cu.ft
RHOTNK	Density of tank car material	lb/cu.ft
RTANK	Inside radius of tank car shell	ft
SKTHK	Thickness of protective skin over insulation	inches
TDCMP	Temperature where insulation loses its effectiveness	^{o}F

TEM1,2	Reference temperatures for thermal conductivities of insulation layers, 1 indicates temperature for outer layer, 2 for the inner one	^oF
TEMPX	Fire temperature values for each HEATX value	^oF
TIMET	Time table for HEATX data	sec
THICK	Tank car shell thickness	inches
TILT	Roll angle to relief valve centerline	degrees
TFI	Initial temperature of liquid	^oF
TLENTH	Length of tank car	ft
TLT	Temperature table corresponding to burst pressure	^oF
TPLOT	Time interval between points on pressure plot	seconds
VOL	Total internal volume of tank car per unit length	ft^3/ft
VENPOS	Distance to each vent from end of car defining vent position	ft

Computational Variables

AEL	Medial area of each element of the tank car shell	
CON	KK·THICK·DELTA/AEL	
CRV	C·RHO·THICK·AEL	
D	RTANK·DANG·DELTA	
DA	DELTA.AEL	
DAO	DANG·(RTANK + THK + THICK/2).DELTA	
DANG	Included angle of each tank car element	
DELX	Length of each tank car element	
FLG	Signal for valve closed (FLG = 0) or open (FLG = 1)	
FLIQ	Signal to indicate valve below liquid level (FLIQ = 1) or above (FLIQ = 0)	
HF	Specific enthalpy of liquid	Btu/lb
HG	Specific enthalpy of vapor	Btu/lb
HTCL	Liquid heat transfer coefficient	Btu/ft^2/hr-^{o}F
KK	Thermal conductivity of shell	
KP	GAMMA	
MG	Mass of gas in tank car per unit length	lb/ft
ML	Mass of liquid in tank car per unit length	lb/ft
MR	Mass flow of material through relief valve per unit length	lb/sec-ft
MR1	Mass flow of liquid relieved	lb/sec-ft
MR2	Mass flow of vapor relieved	lb/sec-ft
PL	Pressure in tank car	psi
PS	Sonic pressure for gas flow through valve	psi
QG	Gas heat transfer rate per unit area for one element of tank car shell	Btu/ft^2-hr
QGT	Heat loss from element of shell	Btu/ft^2-hr
QGSUM	Total heat input to the internal gas environment from the tank car wall	Btu
QINTO	Heat transfer rate per unit area applied to the outside wall of the tank car	Btu/ft^2-hr

QL	Liquid heat transfer rate per unit area for one element to the tank car shell	Btu/ft^2-hr
QLSUM	Total heat input from the tank car wall to the liquid	Btu
SIG C	Circumferential stress in tank car shell	lb/in^2
SIG T	Transverse stress in tank car	lb/in^2
T(N,IDELX)	Temperature of tank car shell element located at circumferential location N and length location IDELX	^{o}F
T'(N,IDELX)	Temperature of tank car shell element at time increment previous to T(N,IDELX)	^{o}F
TAU	Shear stress at 45^o plane in tank car shell element	lb/in^2
TE	Fire temperature	^{o}F
TG	Temperature of gas in tank car	^{o}F
THET	Angle from 0 = 0 to liquid-gas interface at tank car shell	degrees
THETA	Position of the centroid of each element of the tank car shell	radians
THK	Thickness of effective insulation	
TI	Temperature of inside surface of tank car shell element	^{o}F
TL	Temperature of liquid in tank car	^{o}F
TO	Temperature of outside surface of tank car shell element	^{o}F
TS	Sonic temperature for gas flow through relief valve	^{o}R
TSURF	Surface temperature of protective skin	^{o}F
UC	Critical velocity	ft/sec
VF	Specific volume of liquid in tank car	ft^3/lb
VG	Specific volume of liquid in tank car	ft^3/lb
VOLG	Volume of gas in tank car per unit length	ft^3/lb
VOLL	Volume of liquid in tank car per unit length	ft^3/lb

2. Program Logic and Computation

The computer program consists of a main routine and several subroutines. The bulk of the computing is done by MAIN, which calls the subroutines for special purposes. All input is set up by the INPUT subroutine, which also contains write and format statements for print-out of input data. Subroutine OUTPUT contains the general purpose write and format statements for printing the results that define the conditions in the tank. The FORTRAN names for all input variables are listed and identified in the section on "FORTRAN Nomenclature." The units cited are those which must be used for each variable. An explanation of the computational variables is provided, also.

Subroutine HUNTEM is a table look-up procedure used to obtain values from input data arrays.

Subroutine FPLT is a Lagrangian interpolation procedure for obtaining intermediate values for the thermodynamic properties of the lading from the input data array of specific volume (liquid and vapor), pressure, temperature, and latent heat versus enthalpy.

A printer plot subroutine is included called PLOTR, which provides the option of obtaining plots of shell temperature histories and tank pressure history. This option is achieved by giving the value 1.0 to the input variable, PLOT. Subroutine PLOTR calls subroutine PLOTTR, which contains most of the logic of the plotting scheme. It is supported by subroutines NORMAL, AXSCAL, and GRID in the plotting function. Each stage will be explained in detail.

The general organization of the program is illustrated in Figure D-1. Four stages, corresponding to the four main computations, have been designated as follows:

1. Stage A Computations required if tank is shell full.
2. Stage B Computation for shell and surface temperatures.
3. Stage C Valve state logic.
4. Stage D Iteration computations to establish tank pressure.

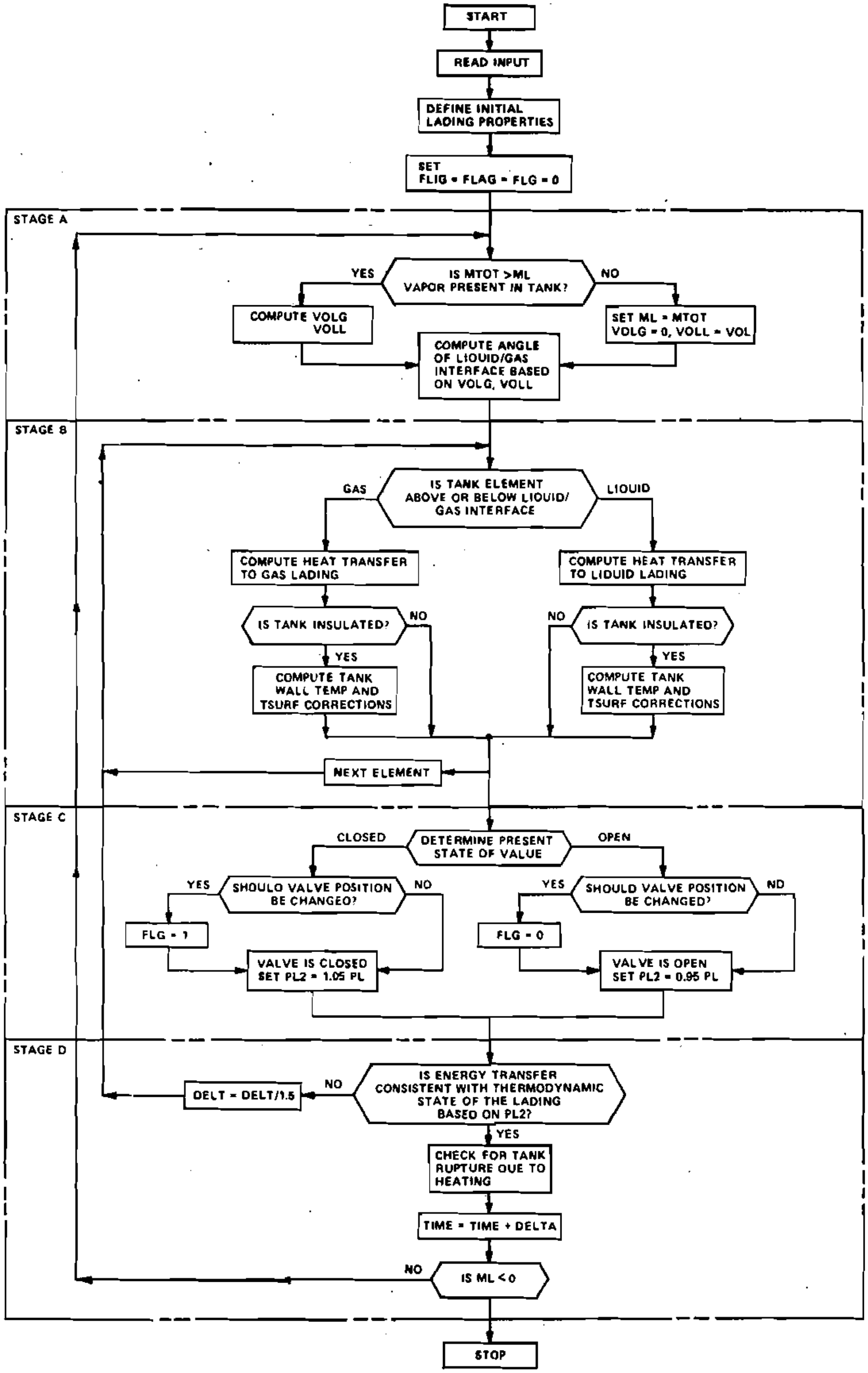

Figure D-1 ORGANIZATION OF TANK CAR PROGRAM

A careful study of Figure D-1 should be made before proceeding with this discussion. Each stage will be explained in detail.

The computation starts by establishing the thermodynamic properties of the lading based on initial temperature and pressure. Thus, the quantities PL, VF, VG, and LT are determined while HG is obtained from HG = HF + LT. All shell temperatures are initialized to TL. With the known volume of the tank and the mass of the lading in the tank, computation for the separate masses of liquid and vapor are made. Finally, the logical variables FLIG, FLL, and FLAG are set to zero and computation can begin.

Stage A (see Figure D-2) is entered to determine if the tank is liquid-full. Using the known volume of the tank and the mass and thermodynamic properties of the lading, Equation D-1 gives the mass of liquid present in the tank.

$$\mathrm{ML} - (\mathrm{VOL} - \mathrm{MTOT} * \mathrm{VGS}) \; / \; (\mathrm{VF} - \mathrm{VGS}) \qquad \text{(D-1)}$$

The mass of vapor in the tank is, then:

$$\mathrm{MG} - \mathrm{MTOT} - \mathrm{ML}$$

If MG is not zero, the tank contains both liquid and vapor, and the computation proceeds to Stage B. However, if the tank is liquid-full, a computation of the flow capacity of the valves is made to determine if the increased lading volume due to an expanding liquid can be relieved. If the valve does not have sufficient capacity, the tank ruptures and the computation is stopped. Otherwise, the volume of the liquid is set equal to the volume of the tank, and a new mass of liquid, ML, is computed. A new MTOT (resulting from loss of fluid by relief) is now set equal to ML.

A test is now made, using the parameter FLIO, to determine if the valve inlet is submerged. If it is, a second test is made to determine if the tank pressure, PL, is above the valve set pressure, PR. A negative result for this test causes the parameter FLAG to be set equal to one, indicating that the tank

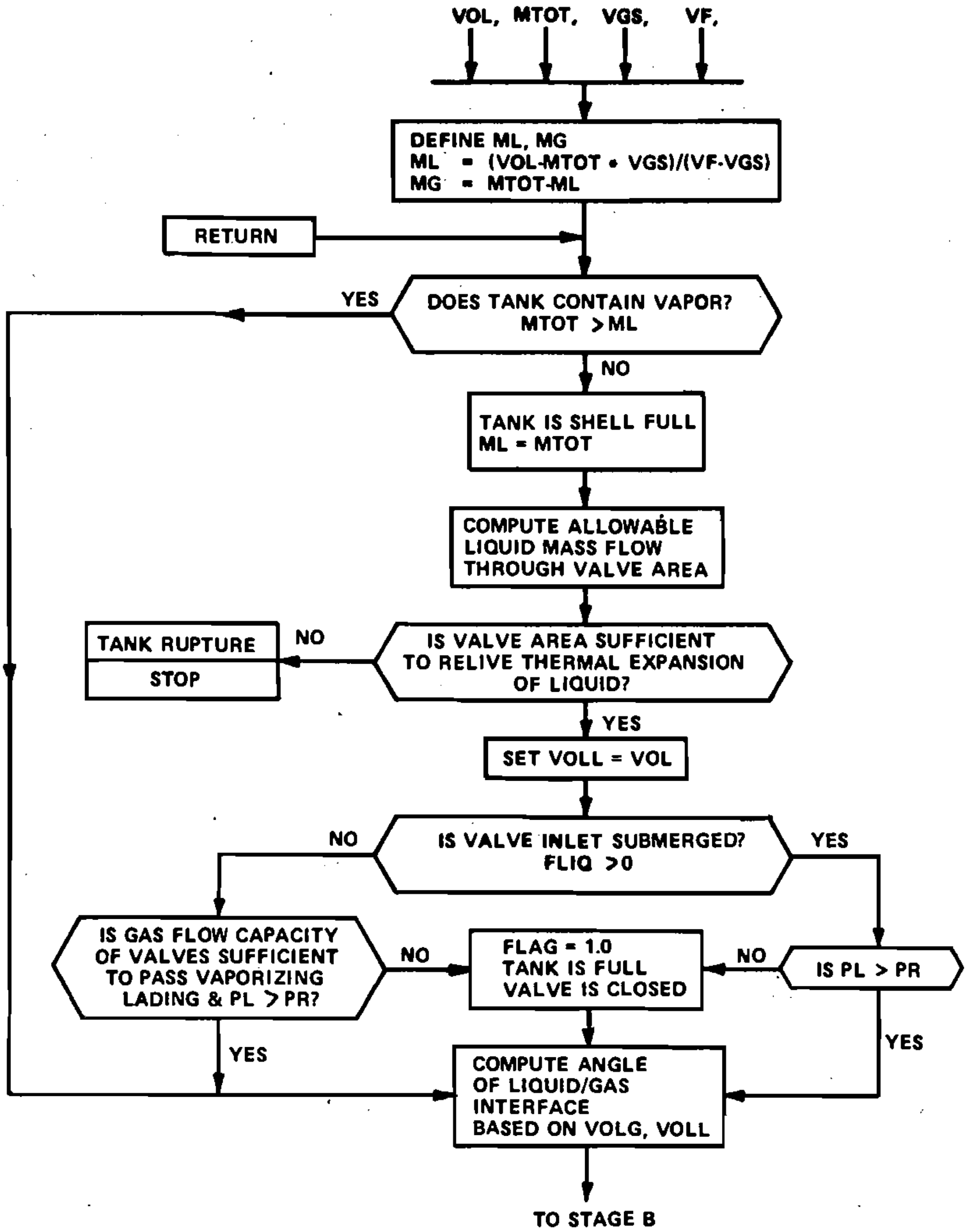

Figure D-2 STAGE A — SHELL FULL COMPUTATIONS is full and the valve is closed. A positive result allows the computation to proceed.

Returning to the test for a submerged inlet, a negative result leads to a second test on valve flow capacity. If the valve is not capable of passing the amount of lading which is being vaporized during this time step or the tank pressure is less than the valve set pressure, the parameter FLAG is again set equal to one and the tank is considered full of liquid with the valve closed. On the other hand, if the valve does have sufficient capacity and the tank pressure is above the valve set pressure, the computation is allowed to proceed.

The final step in Stage A is to determine the angle of the gas/liquid interface. This step involves the determination of the liquid level; specifically, the angle THET that measures its position. THET is half the included angle of the segment of a circle that is described by the points of the intersection of the level surface with the tank. The area of the segment of a circle is:

$$A - 1/2r^2 \ (Y - \sin Y) = 1/2r^2 \ (W)$$

Rearranging and multiplying by π/π,

$$\frac{A}{1/2r^2} = \frac{2\pi A}{\pi r^2} = \frac{2\pi \mathrm{VOLG}}{\mathrm{VOLG} + \mathrm{VOLL}} \equiv V$$

Once V is computed, it is used to step off in a search routine for Y. The test is on the integer difference between V and W, and as soon as it becomes less than 1×10^{-3}, THET is computed from THET = 0.5 * Y.

Stage B (Figure D-3) is now entered to estimate the tank shell heat transfer rates and the shell temperature. The shell has been broken into N circumferential elements, and each pass through Stage B makes computations for a single element.

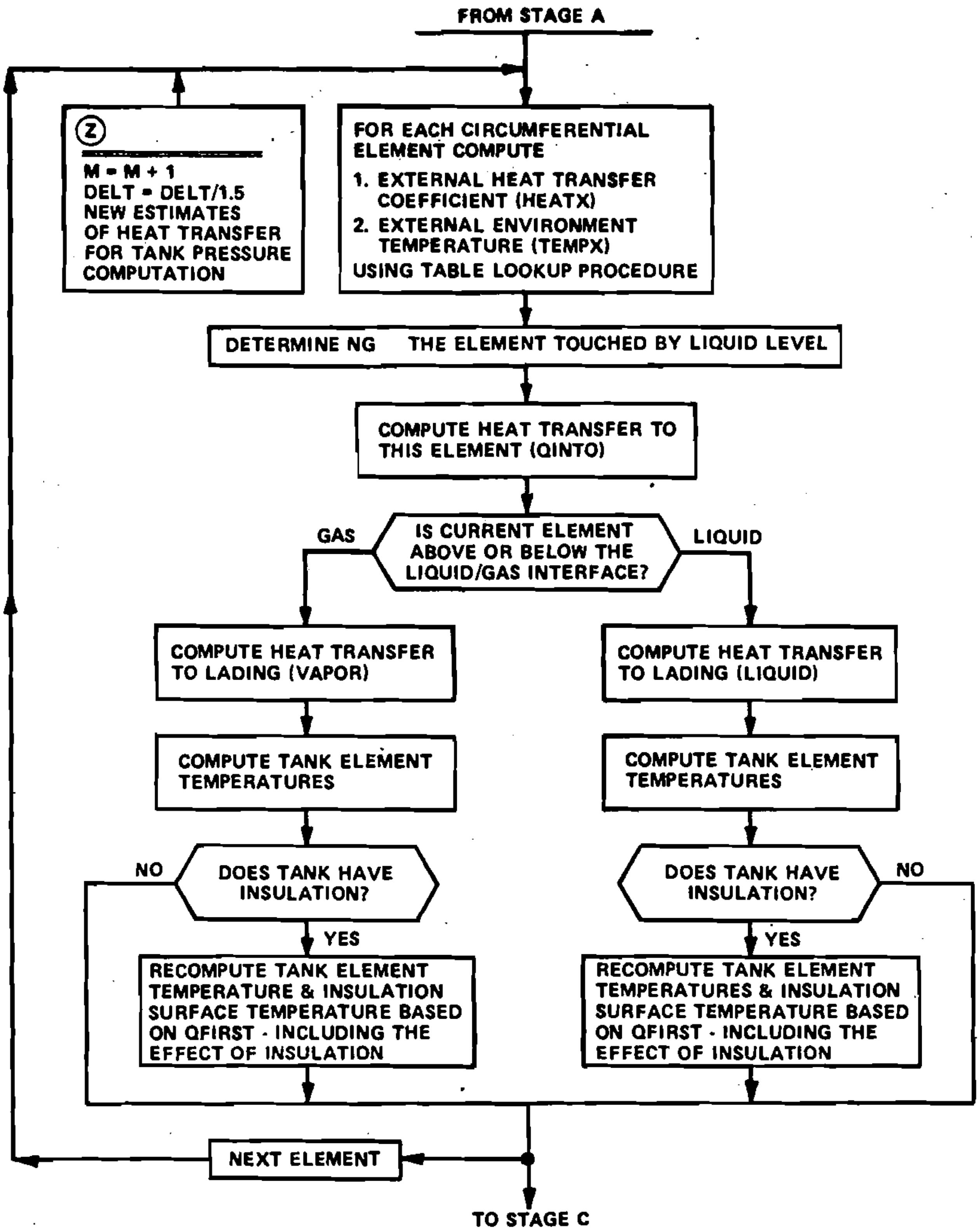

Figure D-3 STAGE B – SHELL TEMPERATURES

The first step in this stage is to obtain the heat transfer coefficient, HEATX, from the data by table look-up and interpolation (HEATX versus ANG need not be specified in increments of shell element width). The same is done for TEMOX, the fire temperature. Then, the element touched by the liquid level is identified (NG). The heat transfer rate into any element from the environment (QINTO) is computed from HEATX and TEMPX. If the element being examined is above the element NG, the gas heat transfer coefficient, NGT, is used to obtain QG, the heat into the vapor. QG is integrated as QGSUM during subsequent passes through the loop. QG is corrected to obtain QGT, the heat loss from the shell element. If the element is below NG, the computation for liquid heat transfer coefficient, HTCL, is made. It depends upon TI, inside surface temperature of the shell, as does the heat transfer rate. TI is estimated from the average shell temperature for the previous time, and from QINTO. The formula for heat transfer coefficient is a curve fit to experimental data, Reference 21, for propane exposed to a horizontal surface. It depends upon tank pressure as well as the temperature difference between shell surface and bulk of the liquid. The quality of the curve fit is demonstrated in Table D-I*.

Shell element temperatures are computed from a relation that is derived in Appendix D-4. This permits a new surface temperature to be computed. At the beginning of the computation, the surface temperature (TSURF (N, IDELX)) is set to a temperature slightly (30^{o}) above TL to induce a smoother start. The inside and outside surface temperatures of the shell, TI and TO (N,IDELX) are computed from relations representing a parabolic temperature profile passing through T (N,IDELX) for elements of the shell above the liquid level. It can be shown analytically that the parabolic profile is valid for a slab of finite thickness with the heat flowing out one side equal to a small proportion of the heat entering the other side. A proof for this is presented in Appendix D-5.

The temperature profile through the shell at elements below the liquid level is assumed to be linear, which is a valid approach for the case of a slab

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*A comparison of the tabulated values shows the lower limit of the range of good fit to correspond to a heat transfer rate of 20,000 Btu/ft^2-hr with degradation increasing as the rate decreases.

Table D-I

VALIDITY OF EQUATION FOR CURVE FIT TO BOILING HEAT TRANSFER RATE

Experimental Data from Reference 31			
Pressure - psia	θ_s, Temperature Difference, Wall and Liquid Bulk - °F	$\dot{q}$, Heat Transfer Rate - Btu/ft^2-hr	Computed Heat Transfer Rate from Equation of Curve Fit - Btu/ft^2-hr
168	18.4	28,850	27,300
170	14.9	19,380	16,300
168	11.0	13,180	7,500
245	23.3	65,600	64,100
240	19.5	40,300	40,100
248	16.1	27,600	25,000
245	14.7	17,060	19,070
295	15.6	31,200	29,300
295	13.3	20,100	19,400
378	15.2	41,200	41,200
375	19.9	88,300	82,600
375	17.7	54,750	60,800
375	13.3	28,100	29,500

$$\frac{\dot{q}}{\theta_s^{2.55}} = 15.0 + 0.0642 \times 10^{-6}\ P^{3.347}$$

or

$$h = HTCL = \frac{\dot{q}}{\theta_s} = \left(15.0 + 0.0642 \times 10^{-6}\ P^{3.347}\right)\theta_s^{1.55}$$

of finite thickness that is transmitting most of the net incident heat. This statement is similar to stating that the rate of heat storage in the shell shell is negligible compared to the rate of heat conducted through it. Consequently, in the heat conduction equation (Equation 1 of Appendix II) the term $\rho c\, \partial T/\partial t \rightarrow 0$. In addition, for the case of uniform heating around the tank, $\partial^2 T/\partial\Theta^2 = 0$, and the equation reduces to $d^2T/dr^2 + 1/r\, dT/dr = 0$. This may be simplified to $d^2T/dr^2 = 0$ for r large compared to δ, the thickness of the shell, which enters the problem as a boundary condition. Integrating $dT/dr = c$, which indicates linearity, and the boundary condition, $dT/dr_w = \dot{q}/k$ establishes the value of c. ($\dot{q}$ is the heat transfer rate at the surface, w).

The condition of negligible heat storage may be justified by comparing the heat storage rate with the heat transmitted to the liquid during a fire. Test results show that the rate of temperature rise of the portion of the shell that contacts liquid averages only $1/2^{o}F$/second when a heating rate equal to 40,000 Btu/ft^2hr is imposed upon it because the temperature of the shell is controlled by that of the liquid. The corresponding rate of heat storage, $\rho c \delta\, dT/dr \simeq$ 3600 Btu/ft^2hr. Subtracting this from the imposed heating rate yields the rate of heat transmission to the liquid, which is over 90 percent of the total.

Provision is made in the program for the variation of thermal conductivity with temperature of any insulation used to cover the tank shell. Two separate layers of different materials are allowed. In preparation for computing thermal conductivity of the insulation, its average temperature (either Tk1 or Tk2) is defined in terms of the prevailing heat transfer rate, thicknesses, and outside and inside surface temperatures. Then thermal conductivity of each layer is computed (KK1 and KK2) as a linear variation from a reference value (FK1 and FK2), which is specified as input data at the reference temperatures, TEM1 and TEM2. This permits TSURF (N,IDELX) to be computed from heat transfer rate, thicknesses, and temperatures of the outside surface of the shell.

As soon as the outer surface of the insulation reaches the decomposition temperature, TDCMP, the program is directed to compute a reduced thickness of insulation. The first calculation for thickness, (TFK) reduces the outer layer to a thickness that will just support the established temperature gradient with an external temperature of TDCMP. (Temperature gradient is dictated by QINTO/KK1.) After THK reduces completely to LAGTH2, the thickness of the inner layer of insulation, a second computation, for THK, dominates the procedure and it operates by using the ratio QINTO/KK2. The insulation surface temperature, TSURF(N,IDELX) is maintained at TDCMP as long as any insulation remains, and this is defined by statement 160. TINT(N,IDELX) is an indexed variable for internal surface temperature of the shell to be stored for print-out.

It is conceivable that all insulation can be decomposed after a time, in which case TSURF is equated to TO(N,IDELX).

When computations have been made for each tank element, Stage C, Figure D-4, is entered. Upon entering, this stage, a signal is set FLQ to indicate whether vapor or liquid will be flowing from the valves. This parameter depends upon TILT, the roll orientation of the valve, and THE, the liquid level. The previously set parameter FLAG is tested to determine if it is greater than zero. If it is, this indicates that the tank is liquid full and the valve is closed. Immediate exit is made to the valve closed routine of Stage D. However, if FLAG is not greater than zero, the valve state logic is entered.

If the valve was not previously open and the tank pressure is still less than the valve set pressure, the valve remains closed and exit is made to Stage D. When the tank pressure is above the set point, FLG, is reset to one, indicating valve is now open, and the parameter FLIQ determines if liquid or gas equations will be used to compute flow through the valves.

A second case is possible, namely, the valve was previously open. If the tank pressure is not greater than the valve closing pressure, PRL, FLQ is set to zero, indicating that the valve is closed, and exit is again made to Stage D. If the tank pressure is above the closing pressure, FLIQ is used to determine if liquid or gas equations will be used to compute flow through the valves.

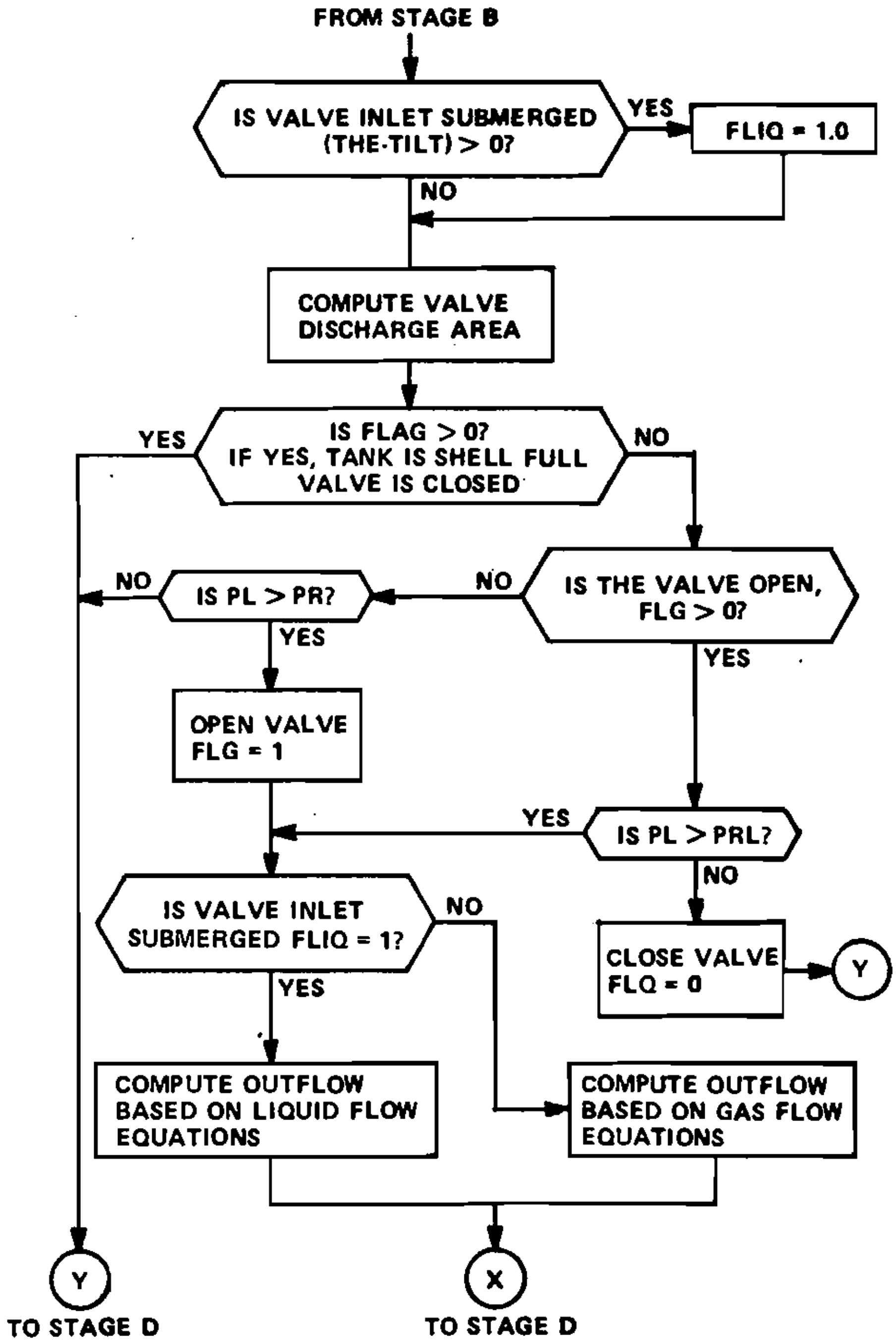

Figure D-4 **STAGE C – VALVE STATE LOGIC**

The primary function of Stage D (see Figure D-5) is to iterate on estimated tank pressure until a lading state is reached which is consistent with the energy transferred to the tank from the environment. There are two entry points to Stage D, depending on whether the valve is open or closed.

Iteration on pressure, for the open valve case, starts by arbitrarily assigning it a new name, PL2, and 95 percent of its previous value. The property subroutine FPLT is then called to get the corresponding equilibrium values for VF and HF. Correct values for liquid and vapor masses are established and enthalpy of the vapor is then computed from a heat balance equation for the vapor. The specific heat of the vapor, CPG, is obtained from HG and PL2 by an equation which is a curve fit to data tables (Reference 32). (Its quality of fit is evaluated in Table D-II).* This permits the computation of TG. Next, VG is computed from the equation of state for the vapor, which was also obtained by a curve fit to the tables of Reference 32. (The validity of this equation is demonstrated by Table D-III).** The revision of this value requires revision, in turn, of the mass of liquid and the mass of vapor.

At this point, all requirements have been satisfied for computing QIN, the heat input to the lading during the computing interval that is necessary to justify the pressure rise to PL2. After assigning QIN a new name, TEST, it is used to find the departure of QIN from the actual heat input over the computing interval, PREV. The difference is called DELQ2. A test is made whereby DELQ2 is compared to a small percentage of the absolute value of (PREV + 10.0). If it is greater than this percentage, the test is not satisfied and PL is corrected by means of a linear extrapolation to the value of PL for which DELQ2 goes to zero. DELQ1 and PL1 are reset to DELQ2 and PL2, respectively. The iteration

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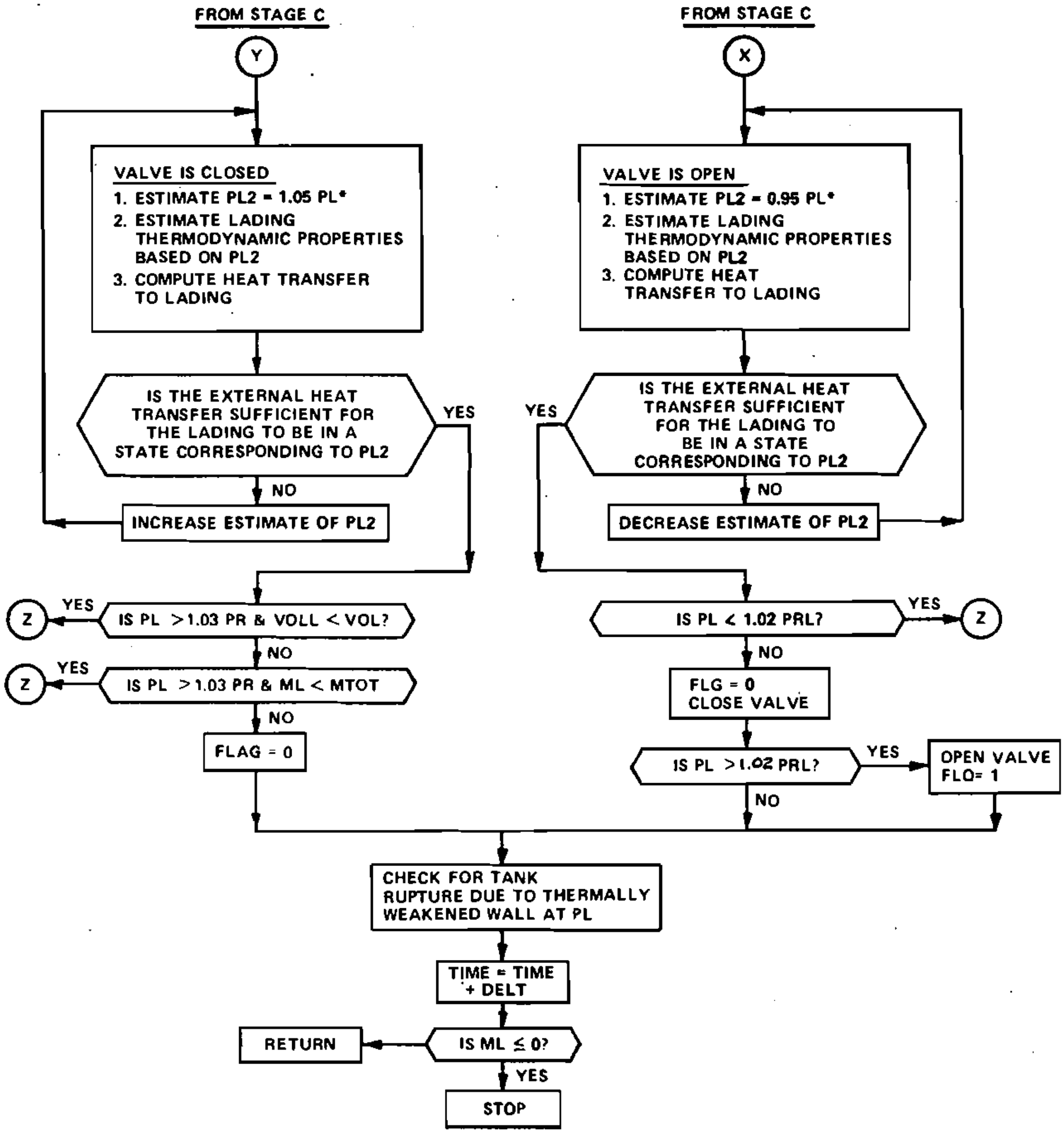
*The tabulated values demonstrate a good fit to specific heat data from the saturation condition to 400°F and 400 psi. As either pressure or temperature are increased beyond this, the fit degrades slowly.

**A good fit is demonstrated by the table over the whole temperature and pressure range of interest to present tank car studies.



*THIS ESTIMATE IS MADE ONLY DURING UPON ENTRY. SUBSEQUENT ESTIMATES ARE PROVIDED IF THE TESTS ON HEAT TRANSFER IS FAILED. SEE TEXT.

Figure D-5 STAGE D – ITERATION COMPUTATION TO ESTABLISH CORRECT TANK PRESSURE

Table D-II

VALIDITY OF EQUATION OF CURVE FIT FOR AVERAGE SPECIFIC HEAT

P, Pressure psi	T, Temperature °F	H, Enthalpy Btu/lb	$\bar{C}_p$, Average Specific Heat	
			Ref. Value from H/T	From Equation of Curve Fit
200	200	450.0	0.682	0.681
	240	470.1	.672	.676
	280	492.6	.666	.669
	300	503.7	.663	.665
	340	526.6	.658	.659
	360	538.3	.656	.655
250	160	420.4	.678	.684
	200	443.5	.672	.678
	240	466.0	.666	.672
	340	523.6	.654	.655
300	160	424.4	.684	.680
	200	446.2	.676	.673
	240	468.0	.669	.667
	340	524.3	.655	.650
400	240	459.2	.656	.660
	400	555.9	.646	.633
600	240	439.6	.628	.649
	400	548.4	.637	.616
	600	649.4	.612	.576

$$\bar{C}_p = 0.829 - 0.000298\,H - 0.00009\,(P - 50)$$

Table D-III

VALIDITY OF EQUATION FOR CURVE FIT FOR VAPOR STATE DATA

T, Temperature °F	V_g, Specific Volume ft^3/lb	P, Pressure - psia	
		From Ref. 32	From Equation of Curve Fit
100	0.5144	200.0	207.0
200	.7038	203	203.7
250	.6080	250	248.3
300	.6706	250	247.7
300	.3910	400	395.6
400	.4731	400	393.8
450	.3683	500	491.4
500	.4312	500	492.0
600	.4043	600	592.0
700	.3858	700	690.5
800	.3721	800	791.2
1000	.3532	1000	994.5

$$P = \frac{0.2433\ T}{V_g - 0.052} - \frac{23.081}{V_g^2}$$

count, ICOUNT, is raised by one, and the computation is repeated with the new estimates. If ICOUNT is excessive, i.e., if it exceeds 10, computation is stopped. When the test on DELQ2 is passed, the tank pressure is tested to prevent this pressure from dropping to a value less than the value closing pressure, which would be unrealistic. If PL has dropped to less than PRL in order to satisfy the iteration procedures, the time interval is reduced and the computer is routed back to the beginning of Stage B, the pivot where new computations start for all quantities dependent upon DELTA. It is necessary to reduce the time interval until the mass lost by relief valve flow is satisfactory for a reduction of tank pressure to PRL. An index, M, counts the number of times the computing interval is divided so that the initial computing interval is divided so that the initial computing interval can be restored.

The closed valve case starts with an arbitrary increase in tank pressure, called PL2. Then FPLT is called, for HF and VF. As in the open valve case, ML2 and MG2 are defined, HG, CPG, TG, and VG are computed. Then ML2 and MG2 are recomputed preparatory to computing QIN. In the case when valves are open, QIN is an enthalpy difference; but for valves closed, it is an internal energy difference, which must be determined by subtracting the flow work terms, (PL x VF) + (PL x VG), from the enthalpy, because internal energy is not otherwise available. The constant is obtained by converting square inches to square feet and ft/lbs to Btu's, i.e., 144/788 = 0.1851.

After setting QIN = TEST, the departure of QIN from PREV is computed as DELQ2. (PREV is the total heat input over the computing interval.) The same kind of test is made comparing PL to 1.03 times PR, and the subsequent logic is similar to that of the valve open case, except that the limiting tank pressure for the valve closed is the set point for valve opening, PR. The computing interval is shortened until this pressure limit is satisfied.

A check is now made for tank rupture due to overpressure by vapor expansion. For each shell element, a table look-up rpocedure gets the tabulated values for temperature in the burst pressure versus temperature table that brackets the temperature of the elements. Then, a linear interpolation is made between their corresponding pressures to find the burst pressure, PB, for the temperature of the element.

A term is computed to account for thermal stress, called THSTRS. This actually performs as a decrement in allowable tank pressure and is used by adding it to tank pressure to get a working pressure that may then be compared with the burst pressure. This procedure is valid because thermal stress may be superimposed upon stresses caused by mechanical loading. The formula for maximum thermal stress from Reference 33, page 174 , is

$$\sigma_{max} = \frac{\alpha E}{2} \frac{(T_1 - T_2)}{(1 - \mu)}$$

where T_1 and T_2 are outer and inner surface temperatures, α is the expansion coefficient, E is Young's modulus, and is Poisson's ratio. For the tank car steel this becomes $\sigma_{max} = 128.0\ (T_1 - T_2)$. The internal pressure that would produce the same stress is 128 t/r $(T_1 - T_2)$, where t is tank thickness and r its radius. The working pressure, PALL, is obtained by adding this to tank pressure and multiplying by the factor RTANK/5, which scales the burst pressure tables to the model tank. (The burst pressure tables are based upon the full size tank car which has a 5-foot radius.)

Finally, a comparison is made between PALL and PB. If PALL is greater than PB, burst is indicated and computation is terminated.

Time is updated and the output subroutine is called before the statements in the program set up the temperature parameters to be plotted. The present configuration of the program generates outside surface temperature histories (TEMDAT (NPT, NEL, IDELX) vs. TIMDAT (NPT)). A bypass is provided in case no plotting is desired. A maximum of 200 points is permitted by a logical IF statement. The pressure parameters are defined for plotting tank pressure history, PDAT (NPP) vs. TIMPDT (NPP). A maximum of 400 points is permitted.

Computation now returns to the beginning of Stage A for computation during the next time increment.

3. Approximate Method for Predicting the New Mix Conditions

The method used in the original model to compute the state conditions of the new mix after a computing interval was approximate because it was based entirely upon conditions at the beginning of the interval. It has been discontinued but is presented here for the sake of completeness.

Previous to computing the new mix conditions, it is necessary to establish the heat input to the tank shell, the mass of gas, MG, and liquid, ML, in the tank, and the total heat input to the liquid as well as to the gas. These are done by the existing methods explained elsewhere.

Then, the following equations are used to compute the increase in the heat content of the liquid in the container. Two specific cases exist. The first case is that for which no mass is lost through the relief valves. The second case distinguishes between liquid or gas flow out each relief valve. The index M indicates time for purposes of this explanation and was not a computing index. The program is recycled to execute computations for a new time. Equilibrium conditions between liquid and vapor were assumed to prevail.

Case 1:

$$MR = 0$$

$$DMG(M) = \frac{(MG(M) - MG(M-1))}{DELTA} \text{, } 3600$$

$$DHF = \frac{(QLSUM(M) + QGSUM(M) - DMG(M)\ L(M)\ \frac{DELTA}{3600})}{ML(M) + (DML(M)\ \frac{DELTA}{3600}}$$

Case 2:

For liquid flow through the relief valve, if one connects with the element X,

$$MR1(M,X) = 192000\ CD\ A$$

For gas flow through the relief valve, if one connects with the element, ΔX,

$$MR2(M,X) = \frac{CD\ A\ UC(M)}{VC(M)}$$

where $$UC(M) = GAMMA \cdot 32.2 \cdot GASCON \cdot (TL(M) + 460)\exp\left(\frac{2}{GAMMA + 1}\right))^{1/2}$$

and $$VC(M) = \frac{GASCON \cdot (TL(M) + 460)^{\frac{2}{GAMMA + 1}}}{PL(M)\left(\frac{2}{GAMMA + 1}\right)^{\frac{2}{GAMMA + 1}}}$$

The total mass loss is computed:

$$MR(M) = \sum_{X=1}^{X=NX} MR1(M) + \sum_{X=1}^{X=NX} MR2(M)$$

and the remaining mass is inventoried:

$$MTOT(M) = MTOT(M - 1) - MR(M)\ \frac{DELTA}{3600}$$

Then the enthalpy of the remaining liquid is determined:

$$DHG - HG(M) - HG(M - 1)$$

$$HF = \frac{\sum_{N=1}^{N=NX} QLSUM(M,X) + \sum_{N=1}^{N=NX} QGSUMM(M,X) - (DMG + MR) \cdot L \cdot \frac{DELTA}{3600} \cdot MG \cdot DHG}{ML(M) + DML - MR) \cdot \frac{DELTA}{3600}}$$

$$HF\ (M + 1) = HF(M) + DHF$$

The value of HF(M + 1) as computed above is then used in conjunction with the liquid-vapor saturation tables to obtain values of PL(M + 1), TL(M + 1), VF(M + 1), VG(M + 1), and L(M + 1).

4. Derivation of Formula for T(N,IDELX)

The unsteady heat conduction equation in a polar coordinate system in terms of radius r, angle θ, and time t with constant thermal properties is:*

$$\rho c \frac{\partial T}{\partial t} = \frac{k}{r^2} \frac{\partial^2 T}{\partial \theta^2} + k \frac{\partial^2 T}{\partial r^2} + \frac{k}{r} \frac{\partial T}{\partial r} \tag{D-2}$$

When this is expressed as a difference equation, using central differences, and transformed to a curvilinear system in r and y using $\Delta y = r\Delta\theta$, it becomes:

$$\rho c\left(\frac{T_t - T_{t-1}}{\Delta t}\right) = k\left(\frac{T_{y+1} - 2T_y + T_{y-1}}{\Delta_y^2}\right) + k\left(\frac{T_{r+1} - 2T_r + T_{r-1}}{\Delta r^2}\right) + \frac{k}{r}\left(\frac{T_{r+1} - T_{r-1}}{\Delta r}\right) \tag{D-3}$$

where T_t is the temperature at time t, T_t-1 is the temperature at the previous time step, T_{y-1} and T_{y+1} are the temperatures at elements adjacent in the y-direction, and T_{r-1} and T_{r+1} are the temperatures at the elements adjacent in the r-direction.

Equation 1 must be accompanied by boundary conditions and initial values in order to use it to describe a problem. For example, the heat transfer rate (k dT/dr) may be specified at the boundaries and a specified uniform temperature may be given as the initial value.

Now the use of Equation (D-3) implies a quasi-steady treatment for an infinitesimal time interval. Furthermore, the size of elements of the tank shell, as measured by Δy, that would be practical for computation is considerably greater than the shell thickness. Consequently, it is practical to consider the element thickness, Δr, to be the same as shell thickness, These two considerations mean that the terms for conduction in the radial direction may be expressed in terms of the boundary conditions. Using a part of the shell in contact with the vapor as an example.

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*c.f. Carslaw, H.S., and Jaeger, J.C., Conduction of Heat in Solids, Oxford University Press, 1959.

$$k\left(\frac{T_{r+1} - 2T_r + T_{r-1}}{\Delta r^2}\right) + \frac{k}{r}\left(\frac{T_{r+1} - T_{r-1}}{\Delta r}\right) = \text{(heat conducted through the insulation minus QGT)/thickness}$$

The heat conducted through the insulation minus QGT is equal to (neglecting the effect of its mass):

$$\text{QINTO}\ (1 + \frac{\text{THK}}{\text{RTANK}}) - \text{QGT}$$

where the quantity in parenthesis corrects for the area change with radius. Putting it in terms of the grouped variables:

$$\frac{\text{QINTO}}{\text{THICK}}\left(1 + \frac{\text{THK}}{\text{RTANK}}\right) - \frac{\text{QGT}}{\text{THICK}} \simeq \frac{\text{DAO}\cdot\text{QINTO} - \text{D}\cdot\text{QGT}}{\text{THICK}\cdot\text{AEL}\cdot\text{DELTA}}$$

Substituting this into Equation (D-3) and using CRV = C RHO THICK AEL and CON = KK THICK DELTA/AEL

$$\frac{\text{CRV}\cdot(\text{T(N,IDELX)} - \text{T}'\text{(N,IDELX)}}{\text{THICK}\cdot\text{AEL}\cdot\text{DELTA}} = \frac{\text{DAO}\cdot\text{QINTO} - \text{D}\cdot\text{QGT}}{\text{THICK}\cdot\text{AEL}\cdot\text{DELTA}}$$

$$+ \frac{\text{CON}\cdot\text{AEL}\ (\text{T(N + 1, IDELX)})}{\text{THICK}\cdot\text{DELTA}\cdot(\text{AEL})^2}$$

$$-(\text{CON} \cdot \text{AEL}) - \frac{2\cdot\text{T(N,IDELX)} + \text{T(N - 1, IDELX)})}{\text{THICK DELTA (AEL)}^2}$$

Solving for T(N,IDELX), which represents here the variable T_t, the relation for T(N,IDELX) in the program is obtained.

$$\text{T(N,IDELX)} = \frac{1}{\text{CRV}}\ (\text{DAO QINTO+CON (T(N - 1, IDELX)-T}'\text{(N,IDELX))}$$

$$+ \frac{\text{CON}}{\text{CRV}} \cdot (\text{T(N + 1, IDELX) - T}'\text{(N,IDELX))}$$

$$- \frac{\text{QGT}}{\text{CRV}} \cdot \text{D} + \text{T}'\text{(N,IDELX)}$$

5. Proof for Temperature Profile Used for Shell

If a function, say f(x), is continuous on an interval $a \leq x \leq b$, then its average value, or mean value, is given by

$$\bar{f} = \frac{1}{b - a} \int_a^b f(x)\, dx$$

In particular, if $0 \leq x \leq \delta$ and the profile for T is the parabolic form:

$$T = \frac{\alpha t}{\delta^2} - \left(1 - \frac{x}{2\delta}\right) \frac{x}{\delta} + \frac{1}{3} \qquad \text{(D-4)}$$

then

$$\bar{T} = \frac{1}{\delta} \int_0^\delta \left[\frac{\alpha t}{\delta^2} - \frac{(1 - x)}{\delta\, 2\delta} \frac{x}{\delta} + \frac{1}{3}\right] dx$$

$$= \frac{1}{\delta} \left[\frac{\alpha t}{\delta^2} x - \frac{1}{\delta} \frac{x^2}{2} + \frac{1}{2\delta^2} \frac{x^3}{3} + \frac{x}{3}\right]_0^\delta$$

$$= \frac{\alpha t}{\delta^2}$$

Define $F_0 = \frac{\alpha t}{\delta^2}$, Then $\bar{T} = F_0$. Now, from the requirements that at $x = 0$. $T = T_o$, and at $x = \delta$, $T = T_i$, then substititing in Equation (D-4):

$$T_0 = \frac{\alpha t}{\delta^2} + \frac{1}{3} = \bar{T} + \frac{1}{3}$$

$$T_i = \frac{\alpha t}{\delta^2} - \frac{1}{6} = \bar{T} - \frac{1}{6}$$

Setting up $\bar{T}$ in terms of T_o and T_i gives

$$\frac{1}{2} T_o = \frac{1}{2} \bar{T} + \frac{1}{6}$$

$$T_i = \bar{T} - \frac{1}{6}$$

or $$\frac{1}{2} T_0 + T_i = \frac{3}{2} \bar{T}$$

$$\bar{T} = \frac{1}{3} (T_o + 2T_i)$$

which are the desired formulae, as used in the program.

6. Derivation of Equations for Vapor Mass Flow Rate

The basic equation for conservation of mass in one dimension states that $\partial(A\rho u)/\partial x = 0$ at any point along a flow passage, i.e., that mass flow rate, $\dot{m} = A\rho u = Au/v$, where A is flow area, u is velocity, ρ is density, and v, specific volume, when the valve on a tank car is open only for pressures of magnitude greater than about 200 psi. This insures choked flow through the valve because the ratio of atmospheric pressure to tank pressure is less than the critical value. This means that at the point along the passage where its flow area is minimum the flow velocity will be sonci. The relation for sonic velocity in a gas of constant ratio of specific heats is $u = \sqrt{g\gamma RT_s}$, where g is the acceleration of gravity, γ is the ratio of specific heats, R is the universal gas constant, and T_s is the stream static temperature. T_s can be obtained from $T_s = (T_R)^{2/(\gamma+1)}$* where T_R is a reservoir or total temperature and in the present case $T_R = T_L$, temperature of the vapor in the tank.

At this critical point where velocity is sonic, the specific volume is desired, also. From the perfect gas equation of state, $v_c = RT_s/P_s$.

In real flows the full value of Au/v is not realized, and a flow coefficient is defined as the ratio of actual to ideal flow rate of $C = \dot{m}/Au/v$. Thus, all the relations used in the program to compute MR, i.e., $\dot{m}$, are explained.

This method for computing mass flow rate of vapor is a simplified one inasmuchi as it assumes the fluid to reamin in the vapor phase during its expansion. A small amount of liquid actually forms although its effect is negligible. Flow of vapor could have been treated by the method described below for liquid.

*See any textbook on gasdynamics, e.g., Shapiro, A.H., Compressible Fluid Flow, Ronald Press, 1953.

7. Derivation of Equations for Liquid Flow Rate

Flow of fluid through a relief valve is assumed to be isentropic, at constant total enthalpy. Total enthalpy is $u^2/2g + h$ where h is static enthalpy. Therefore, $u_1^2/2g + h_1 = u2^2/2g + h_2$ where the subscripts refer to two different stations along the flow passage. Let 1 represent the inlet at tank conditions and 2 be the minimum area condition. But $u_1 = 0$ so that $u_2 = \sqrt{2g(h_1 - h_2)}$. The mass flow rate is $\dot{m} - CA_2u_2/v_2$ (See Appendix D-6). Combining these two equations.

$$\frac{\dot{m}}{CA_2} = \sqrt{\frac{2g(h_1 - h_2)}{v2}}$$

Now the enthalpy at 1, h_1, is that of saturated liquid found in the thermodynamic table for the fluid. To find h_2 and v_2, use S_1, the entropy of saturated liquid (from the table), which is equal to S_2. At any given pressure, p_2, downstream in the valve, the fraction x_2, of liquid to total fluid mass (i.e., quality) can be determined from $x = S - S_f/S_g - S_f$ where f and g denote liquid and vapor (i.e., gas), respectively. Then enthalpy, h_2, and specific volume can be determined from the relations:

$$h_2 = x(h_g - h_f) + h_f$$

$$v_2 = x(v_g - v_f) + v_f$$

Calculations for various pressures, p_2, yield curves of $\dot{m}/CA_2$ versus p_1; each with a single maximum. A curve through these maximum points appears as shown in the following figure, which is for propane. (The maximum point is analogous to the choked condition.) The case for a departure from isentropic flow by 20 percent was also computed.

In general, relatively large changes in entropy would be expected in the valve, as well as significant loss of flow energy due to the momentun exchange with liquid droplets that are formed. Consequently, a flow coefficient C_D, should be used to account for the losses.

The program uses a constant value of 3200 lb/sec-ft^2 for $m/C_D A_2$ because liquid relief of propane only occurs above 265 psia, and the curve in the figure does not vary much for higher pressures. (See Figure D-6). When this value is multiplied by 3600 to convert the units to lb/hr-ft^2, the constant, 11,520,000 is obtained. This, in turn, must be divided by TLENTH to put MR1 on a lb/hr per foot of tank length basis.

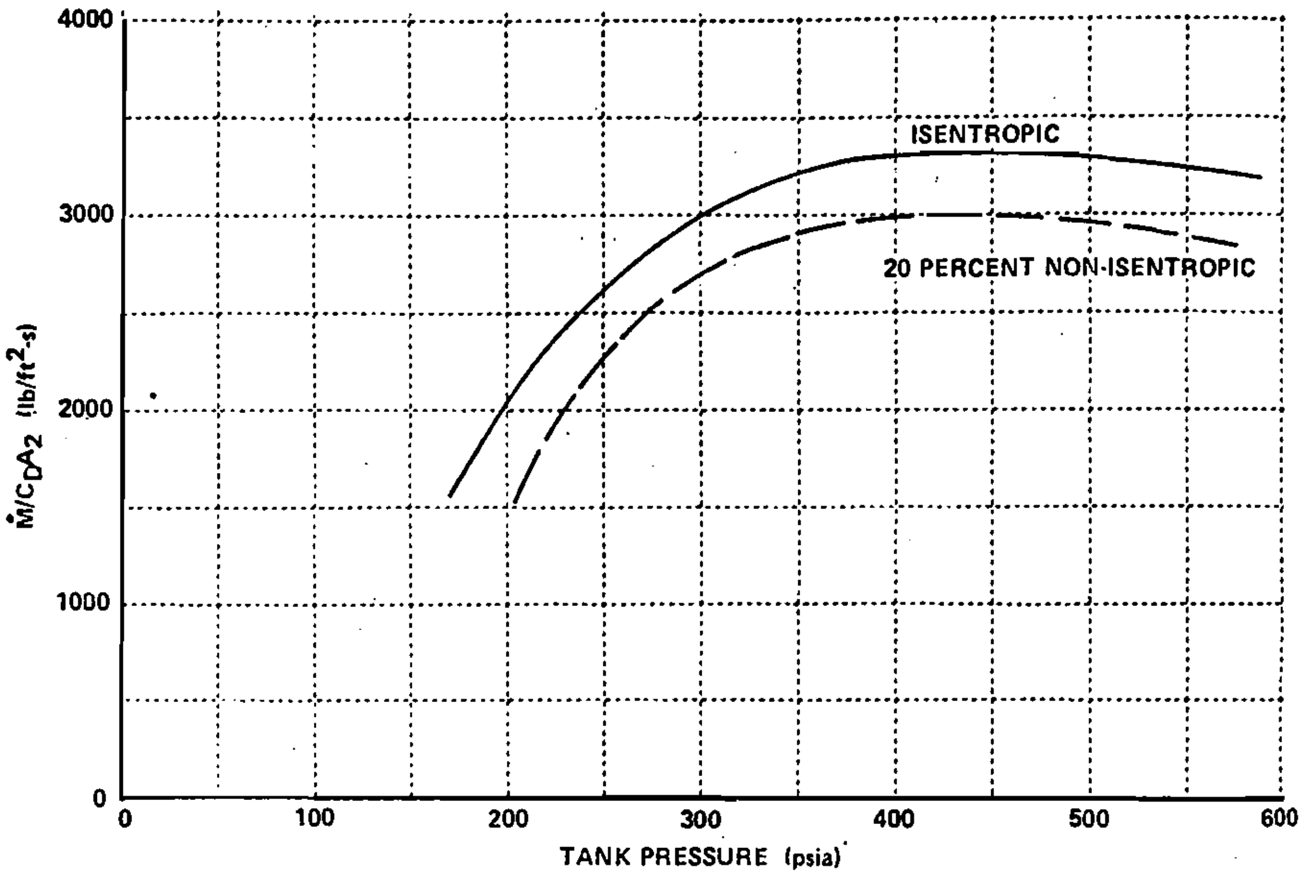

Figure D-6 MAXIMUM FLOW OF LIQUID PROPANE THROUGH AN ORIFICE

```
      C*        COMPUTATIONAL VARIABLES;
      C*
      C*    ALL     - CROSS SECTIONAL AREA OF EACH ELEMENT OF THE TANK CAR
      C*        SHELL
      C*    DANG    - INCLUDED ANGLE OF EACH TANK CAR ELEMENT
      C*    FLG  -    IF 0., THE VALVE IS CLOSED , IF 1., IT IS OPEN
      C*    FLIG    - IF ZERO, IMPLIES GAS FLOW THROUGH VALVES
      C*            - IF 1. IMPLIES LIQUID FLOW THROUGH VALVES
      C*    G       - GRAVITY
      C*    HE      - COMBINED GAS/SHELL HEAT TRANSFER COEFFICIENT . BTU/FT**2-HR-DG
      C*    HF      - SPECIFIC ENTHALPY OF LIQUID AT TIME T
      C*    HG      - SPECIFIC ENTHALPY OF VAPOR AT THE TIME T
      C*    QG      - GAS HEAT TRANSFER RATE PER UNIT AREA FOR ONE ELEMENT
      C*        OF TANK CAR SHELL   ......  BTU/FT**2-HR
      C*    QGSUM   - TOTAL HEAT INPUT TO THE INTERNAL GAS ENVIRONMENT FROM
      C*        THE TANK CAR WALL ...... BTU
      C*    QINTG   - HEAT TRANSFER RATE PER UNIT AREA APPLIED TO THE OUT-
      C*        SIDE WALL OF THE TANK CAR ...... BTU/FT**2-HR
      C*    QL      - LIQUID HEAT TRANSFER RATE PER UNIT AREA FOR ONE ELEMENT
      C*        OF THE TANK CAR SHELL ...... BTU/FT**2-HR
      C*    SIGC - CIRCUMFERENTIAL STRESS IN TANK CAR SHELL ... LB/IN**2
      C*    SIGT    - TRANSVERSE STRESS IN TANK CAR SHELL .... LB/IN**2
      C*    T       - AVERAGE TEMPERATURE OF TANK SHELL ELEMENT ... DEG. F.
      C*    SIG     - BOLTZMAN'S CONSTANT
      C*    TAU     - SHEAR STRESS AT 45-DEGREE PLANE IN TANK CAR SHELL ELE-
      C*        MENT .... LB/IN**2
      C*    TC      - FIRE TEMPERATURE .... DEG. F.
      C*    THET    - ANGLE FROM THETA =0 TO LIQUID-GAS INTERFACE AT TANK
      C*        CAR SHELL .... DEG.
      C*    THETA   - POSITION OF THE CENTROID OF EACH ELEMENT OF THE TANK
      C*        CAR SHELL ... RADIANS
      C*    TI      - TEMPERATURE OF INSIDE SURFACE OF TANK CAR SHELL ELEMENT
      C*        ...... DEG. F.
      C*    TL      - TEMPERATURE OF LIQUID IN TANK CAR ..... DEG. F.
      C*    To      - TEMPERATURE OF OUTSIDE OF TANK CAR SHELL .... DEG. F.
      C*    TS      - SONIC TEMPERATURE FOR GAS FLOW THROUGH RELIEF VALVE
      C*        ..... DEG. F.
      C*    UC      CRITICAL VELOCITY ..... FT/SEC
      C*    VF      - SPECIFIC VOLUME OF LIQUID IN TANK CAR .... LB**3/LB
      C*    VG      - SPECIFIC VOLUME OF GAS IN TANK CAR ..... FT**3/LB
      C*    VOLG    - VOLUME OF GAS IN TANK CAR PER UNIT LENGTH ... FT**3/FT
      C*    VOLL    - VOLUME OF LIQUID IN TANK CAR PER UNIT LENGTH ...FT**3/FT
      C*
      C*                                                             00000010
      C*          TANK CAR MODEL PROGRAM                             00000020
      C*                                                             00000030
ISN 0001  COMMON/OUTPUT/ TIME,HF,PL, TL,TI,VOLL, VOLG, MC,ML,MR,THE,
         *DANG,TINI(50,4),SIGC,SIGT,TG,QINTG,QGSUM,QLSUM,QTSUM
ISN 0002  COMMON/COMP/C,EC,EM,G,KK,KP,RHO,PP,SIG,CG,GAMMA,GASCON,HCT  00000050
         *,RHOSK,SKTHK                                               00000060
```

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0003            REAL KK,KP,MTOT,C,MG,ML,LT,MGG,MR                                          00000100
0004            COMMON/TEMP/ T( 30,6), TT( 30,6),TSURF( 30,6),CT(25),
               * THS(30,6), TTHS(30,6),       HFT(25),TLT(25),PLT(25),VFT(25),
               *VOL(25),        TTT(30),PBT(30) ,          X(4,5),S(5)                    00000130
0005            COMMON/MODIFY/ HEATX(30,10,6), TEMPX(30,10,6), ANG(30), TIMET(10),00000140
               1 TICT,PITCH,NX,FRAC
0006            COMMON/VALVES/ VENPOS(6), VAREA(6)                                         00000160
0007            COMMON/TO/ TO( 30,6)
0008            REAL MR1, MR2,LAGTHK                                                       00000170
0009            REAL*4 LAGTH1, LAGTH2
0010            REAL*4 KK1, KK2
0011            COMMON/ON/HOFTIM(3), TOFTIM(3)                                             00000180
0012            COMMON/PRINT/MODPNT
0013            COMMON  /GENRL/ DELX,                   CPTNK, EI, EFIRE, FKS,             00000190
               1 RHOTNK, CD, DELTA, HF1, MTOT, NEL, PR, THICK, RTANK, PRL, EMO,           00000200
               2 TLENTH,TDCMP,                 CINS1, CINS2, TEM1, TEM2, LAGTH1,
               3 LAGTH2, FK1, FK2
0014            COMMON /PLOTS/ PLOT,TPLOT,NPT, NPP, TEMDAT(200, 25,6),TIMPDT(400),
               1         PDAT(400), TIMDAT(200), NRAD, NP,NSTA1,NSTA2
0015            DATA IPRINT/0/
0016            DATA DELQ1/100./
          C*                                                                               00000230
0017            NAMELIST /BUG/ ML,PL,QFRST,QINTO,  TI2,  T23, NG,
               *  TIME, DELTA,     TEST,PREV,HF ,QIN , PL2, QLSUM, ICOUNT,
               *  MG, MG2,  HG,  ML2,  MTOT
          C*                                                                               00000220
0018            XLAGR(C0,C1,C2,CX,U0,U1,U2)=(CX-C1)*(CX-C2)/(C0-C1)/(C0-C2)*U0-           00000240
               1(CX-C0)*(CX-C2)/(C0-C1)/(C1-C2)*U1+(CX-C0)*(CX-C1)/(C0-C2)/(C1-C2)00000250
               2*U2                                                                        00000260
0019        102 FORMAT  (8F8.3,F16.12,F4.0)                                                00000280
0020        113 FORMAT  ('   BURST TABLE LIMITS ELEMENT ',I6)                              00000290
0021        114 FORMAT  (' T ',F7.2,'(',I4,')  PL',F6.2,'  PB',F6.0,' TIME',F7.0) 00000300
0022        123 FORMAT  (10F10.2)
0023            CALL INPUT                                                                 00000370
0024            GO TO 2                                                                    00000380
0025          1 CONTINUE                                                                   00000390
0026            NP = NP-1
0027            IF(NP.LE.0) NP=1
0028            NPT= NPT -1
0029            IF(NPT.LT.2) NPT=2
0030            NPP= NPP-1
0031            IF(NPP.LT.1) NPP=2
0032            IF(PLOT.EQ. 1.0) CALL PLOTR
0033            CALL INPUT
0034          2 CONTINUE
0035            TSAV = DELTA
0036            LAGTH1 = LAGTH1/12.
0037            LAGTH2 = LAGTH2/12.
```

```
0038              TIME=0.
0039            3 THICK=THICK/12.                                              00000430
0040              XLX=FLOAT(NX)
          C*      DELTNG=0.
0041              FLIQ = 0.
0042              DELQ1=0.
0043              DELQ2=0.
0044              DELV=0.
0045              DANG=3.1416/(NEL-1)
0046              EEO = SQRT(EO)
0047              EEEO = SQRT(EEO)
0048              AEL   =(RTANK+.5*THICK)*DANG                                 00000460
0049              DA    =DELTA/3600.*AEL                                       00000470
0050              CRV   =C*RHO*THICK*AEL                                       00000480
0051              VOL   =3.1416*RTANK*RTANK
0052              CON   =KK*THICK*DELTA/(AEL*3600.)                            00000500
0053              QTSUM=0.0
0054              QLSUM=0.                                                     00000510
0055              QGSUM=0.                                                     00000520
0056              FLAG=0.                                                      00000530
0057              FLG=0.                                                       00000540
0058              POP= 0.0
0059              RHOCI=TDCNP
          C*                                                                   00000550
          C*         SEARCH ENTHALPY TABLES FOR HF1                            00000560
          C*                                                                   00000570
0060              DO  6  J=2,20
0061              IF  (HFT(J)-HF1)  6,6,7                                      00000590
0062            6 CONTINUE                                                     00000600
0063            7 H1=HFT(J-2)                                                  00000610
0064              H2=HFT(J-1)                                                  00000620
0065              H3=HFT(J)                                                    00000630
0066              J=J-2                                                        00000650
0067              DO 10 I=1,4
0068              X(I,1)=TLT(J)                                                00000670
0069              X(I,2)=PLT(J)                                                00000680
0070              X(I,3)=VFT(J)                                                00000690
0071              X(I,4)=VGT(J)                                                00000700
0072              X(I,5)=LT(J)                                                 00000710
0073           10 J= J+1
          C*                                                                   00000730
          C*         INTERPOLATE FOR THE INDEPENDENT VARIABLE HF1 THE SATURATION   00000740
          C*         VALUES FOR;                                               00000750
          C*              PLT  - PRESSURE                                      00000760
          C*              TLT  - TEMPERATURE                                   00000770
          C*              VFT  - SPECIFIC VOLUME OF SATURATED LIQUID           00000780
          C*              VGT  - SPECIFIC VOLUME OF SATURATED VAPOR            00000790
          C*              LT   - HEAT OF VAPORIZATION OF LADING                00000800
          C*                                                                   00000810
```

```
                DO 15 J=1,5
                XV1=XLAGR(H1,H2,H3,HF1,X(1,J),X(2,J),X(3,J))
             15 S(J) = XV1
                TL=S(1)
                PL=S(2)
                VF=S(3)
                VG=S(4)
                VGS= VG
                VGS2 = VG
                VGS1  = VG
                L =S(5)
                HG=HF1+L
                CPG=.65
                HF=HF1
                TG = TL + 460.
                I=NEL+1
                NPRNT=40
                TK1 = TL
                TK2 = TL
                KK1=FK1
                KK2=FK2
                THK = LAGTH1 + LAGTH2
                DO 20 J=1,1
                DO 20 IDELX=-1,NX
                T(J,IDELX) = TL
                TSURF(J,IDELX)=TL+300.
                THS(J,IDELX)=TL
                TTHS(J,IDELX) = TL
                TO(J,IDELX)=TL
             20 TT(J,IDELX)= TL
                TL2=TL
          C*
          C*       INITIALIZE PLOT PARAMS
          C*
                NPT=1.
                NPP= 1
                IF(TPLOT.EQ. 0.0) TPLOT= DELT
                IF(NRAD .EQ. 0) NRAD= 10
          C                   GET MASS OF LIQUID
          C              GET MASS OF VAPOR
             30 ML=(VOL-MTOT*VGS)/(VF-VGS)
                MG = MTOT - ML
                IF(MTOT-ML) 32,32,31
             31 IF(ML.LT.0.) ML= 0.
                VOLL=ML*VF
                VOLG=MG*VGS
                GO TO 50
             32 ML= MTOT
                GSR = 0.
          C*
          C*       TANK IS SHELL FULL
          C*
```

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      VOLL=ML*VF
      IF(MG.LT.0.) MG=0.
      VOLG=0.
      VOLP=0.
      PS=PL*(2./(KP+1.))**(KP/(KP-1.))
      TS=(TG)*(2./(KP+1.))
      VC=RP*TS/(PS*144.)
      UBC=ABS(KP*G*RP*TS)
      UC=SQRT(UBC)
      DO 40 IDELX= 1,NX
      A = VAREA(IDELX)
      IF(VENPOS(IDELX) -(IDELX-1)*DELX .LT. .75*DELX .AND.
     1   VENPOS(IDELX) -(IDELX-1)*DELX .GT. .25*DELX ) GO TO 35
      A= .5*A
   35 CONTINUE
      GSR = GSR + (CG*A*UC)/VC*(3600./TLENTH)
   40 VOLP= VOLP+11520000./TLENTH*CG*A
      TOPM=VOLP*DELTA*VF/3600.
      POP= VOLL- VOL
      IF(POP.LE.TOPM) GO TO 45
      WRITE(6,171)
  171 FORMAT('  TANK RUPTURE DUE TO EXPANDING LIQUID')
      GO TO 1
   45 GASR=GSR*DELTA/3600.
      POPMS=POP/VF
      VOLL=VOL
      IF(FLIQ) 46,46,47
   46 IF(GASR .GE. POPMS .AND. PL .GE. PR) GO TO 49
      GO TO 48
   47 IF(PL .GE. PR) GO TO 49
   48 FLAG = 1.0
      VOLG=0.
      ML=VOLL/VF
      MTOT=ML
   49 CONTINUE
C*        DETERMINE THE LIQUID/GAS INTER-FACE
   50 IF(VOLG.GT.0.5*VOL) GO TO 51
      V= 6.28*VOLG/(VOLG+VOLL)
      Y=V/3.
   55 W= Y-SIN(Y)
      IF(IFIX((V-W)*1000.))  57,65,57
   57 Y=Y+(V-W)*.2
      GO TO 55
   51 V=6.28*VOLL/(VOLL+VOLG)
      Y=V/3.
   58 W=Y-SIN(Y)
      IF(IFIX((V-W)*1000.)) 59,62,59
   59 Y=Y+(V-W)*0.2
      GO TO 58
```

```
      62 THET=3.1415-0.5*Y
         THE=THET*57.2958
         GO TO 71
      65 THET= .5*Y
      70 THE=.5*57.2958*Y
      71 CONTINUE
         DAU=DANG*(RTANK+THK+THICK/2.)*DELTA/3600.
         D=RTANK*DANG*DELTA/3600.                                      00001600
         M=1
         WEHT= MTOT
         PLIN= PL
         FLUID= ML
         GASIN = MG
         VF1= VF
         VG1= VGS
         HF1= HF
         HG1= HG
         EL1= L
         TL1= TL
         TG1 = TG
      75 CONTINUE
C*
C*          THIS IS THE START OF THE LOOP THAT ITERATES FOR PRESSURE
C*          SAVE LADING CONDITIONS FROM PREVIOUS TIME INTERVAL
C*
         DAO=DAU/(1.5**(M-1))
         CON=CON/(1.5**(M-1))
         D=D/(1.5**(M-1))
         MTOT= WEHT
         QLSUM=0.
         QGSUM=0.
         PL= PLIN
         ML= FLUID
         MG= GASIN
         VF= VF1
         VGS= VG1
         HF= HF1
         HG= HG1
         TG = TG1
         L= EL1
         TL= TL1
C*
C*       START LOOP ON X STATIONS
C*
         IPLOT= 0
         TG = TG-460.
         DO 200 IDELX= 1,NX
         DO 200 N=1,NEL
```

```
0206            CALL HUNTEM(TIME, TIMET, 10, J)
          C*
          C*      J POINTS TO LEAD INDEX FOR LAGRANGIAN FIT FOR TIME
          C*
0207            ANGLE = DANG*FLOAT(N) * 57.2958
0208            CALL HUNTEM (ANGLE, ANG, 30, K)
          C*
          C*      K POINTS TO LEAD INDEX FOR LAGRANGIAN ON HEATX
          C*
          C*      NOW GENERATE THREE POINTS DEFINING HEATX AS FUNCTION OF TIME
          C*      AND TEMPX AS FUNCTION OF TIME
          C*
0209            KSAV = K
0210            DO 80 I=1,3
0211            HOFTIM(I)  = XLAGR(TIMET(J),TIMET(J+1),TIMET(J+2),TIME,
               1 HEATX(K,J,IDELX),HEATX(K,J+1,IDELX),HEATX(K,J+2,IDELX))
0212            TOFTIM(I)  = XLAGR(TIMET(J),TIMET(J+1),TIMET(J+2),TIME,
               1 TEMPX(K,J,IDELX), TEMPX(K,J+1,IDELX), TEMPX(K,J+2,IDELX))
0213            K=K+1
0214         80 CONTINUE
0215            K = KSAV
0216            HE  = XLAGR(ANG(K),ANG(K+1),ANG(K+2),ANGLE,
               1 HOFTIM(1),HOFTIM(2),HOFTIM(3))
0217            TE  = XLAGR(ANG(K),ANG(K+1),ANG(K+2),ANGLE,
               1 TOFTIM(1),TOFTIM(2),TOFTIM(3))
0218            IF(N-1) 81,81,82
          C*
          C*       COMPUTE HEATING RATES AND ELEMENT TEMPERATURES
          C*
0219         81 XX = TT(2,IDELX)
0220            YY=TTHS(2,IDELX)
0221            GO TO 85
0222         82 XX= TT(N-1,IDELX)
0223            YY=TTHS(N-1,IDELX)
          C*
          C*       NG IDENTIFIES ELEMENT AT LIQUID SURFACE
          C*
0224         85 NG= IFIX(1.5+THET/DANG)
0225            QFRST=HE*(TE-TSURF(N,IDELX))+SIG*(EO*EM*(TE+460.)**4-EM*
               * (TSURF(N,IDELX)+460.)**4)
0226            IF(N-NG) 90,100,100
0227         90 CONTINUE
0228            QG=HGT*(TT(N,IDELX)-TG)
0229            QGSUM=QGSUM+2.*D*QG/XLX
0230            QL=0.
0231            RF = 1.
0232            QGT=QG+SIG*EI*((T(N,IDELX)+460.)**4-RF*((TL+460.)**4))
0233            QINTO=QFRST
0234            IF(LAGTH1 .EQ. 0. .AND. LAGTH2 .EQ. 0.) GO TO 95
          C*
          C*      THE TANK IS INSULATED-GET ITS TEMPERATURE AND HEAT PENETRATRATION
          C*
```

```
0235          QINTO=2.0*KK*(TO(N,IDELX) - T(N,IDELX))/THICK
0236          QEX=2.*KK2*(TTHS(N,IDELX)-TO(N,IDELX))/THK
0237          IF(QINTO .LT. 0.0) QINTO=0.0
0238          IF(QEX .LT. 0.0) QEX=0.
0239          QINTO=.5*(QEX+QINTO)
0240          THS(N,IDELX)=TTHS(N,IDELX)+DELTA/(3600.*RHOCI)*(KK2*
             1 (TTHS(N+1,IDELX)-2.*TTHS(N,IDELX)+YY)/(((RTANK+THK/2.)*DANG)**2)
             2 +QFRST/THK*(1.+THK/RTANK)-QINTO/THK)
0241          IF(THS(N,IDELX) .LT. TO(N,IDELX)) THS(N,IDELX)=TO(N,IDELX)
0242          TSURF(N,IDELX)=2.*(THS(N,IDELX)-TO(N,IDELX))
0243          IF(TSURF(N,IDELX) .GT. 1.05*(EEED*(TE+460.)-460.))
             1 TSURF(N,IDELX) = 1.05*(EEED*(TE+460.)-460.)
0244          T12=THS(N,IDELX)
0245          T23=TSURF(N,IDELX)
0246          KK1=FK1
0247          TINS=ABS(THS(N,IDELX)-TEM2)
0248          KK2=FK2+CINS2*TINS**1.945
0249          GO TO 96
0250       95 CONTINUE
0251          TSURF(N,IDELX)=TO(N,IDELX)
0252       96 CONTINUE
0253          T(N,IDELX)= (DAO*QINTO+CON*(XX-TT(N,IDELX))+CON*
             1 (TT(N+1,IDELX)-TT(N,IDELX))-QGT*D+CRV*TT(N,IDELX))/CRV
0254          TI1=T(N,IDELX)
0255          TI= T(N,IDELX)-QINTO*THICK/(6.*KK)
0256          IF(TI .LT. TL) TI=TL
0257          TO(N,IDELX)=3.*T(N,IDELX)-2.*TI
0258          GO TO 180
0259      100 CONTINUE
0260          TI=TT(N,IDELX)-(.5*QINTO*THICK/KK)
0261          IF(TI .LT. TL) TI=TL
0262          HTCL=(15.+(.0642E-6)*PL**3.347)*((ABS(T(N,IDELX)-TL))**1.55)
0263          IF(HTCL .GT. 6000.) HTCL=6000.
0264          QL=HTCL*(T(N,IDELX)-TL)
0265          IF(QL .GT. 100000.)QL=100000.
0266          IF(N .EQ. NG) QL = .5*QL
0267      120 CONTINUE
0268          IF(N .EQ. NEL) QL=.5*QL
0269          QLSUM=QLSUM+2.*D*QL/XLX
0270          QG=0.
0271          QINTO = QFRST
0272          IF(LAGTH1 .EQ. 0. .AND. LAGTH2 .EQ. 0.) GO TO 130
       C*
       C* THE TANK IS INSULATED-GET ITS TEMPERATURE AND HEAT PENETRATION
       C*
0273          QINTO=2.0*KK*(TO(N,IDELX) - T(N,IDELX))/THICK
0274          IF(QINTO .LT. 0.) QINTO=0.
0275          QEX=2.*KK2*(TTHS(N,IDELX)-TO(N,IDELX))/THK
```

```
0276            IF(QEX .LT. 0.) QEX=0.
0277            QINTO=.5*(QEX+QINTO)
0278            THS(N,IDELX)=TTHS(N,IDELX)+DELTA/(3600.*RHOC1)*(KK2*
              1 (TTHS(N+1,IDELX)-2.*TTHS(N,IDELX)+YY)/(((RTANK+THK/2.)*DANG)**2)
              2 +QFRST/THK*(1.+THK/RTANK)-QINTO/THK)
0279            IF(THS(N,IDELX) .LT. TC(N,IDELX)) THS(N,IDELX)=TO(N,IDELX)
0280            TSURF(N,IDELX)=2.*(THS(N,IDELX)-TO(N,IDELX))
0281            IF(TSURF(N,IDELX) .GT. 1.05*(EEEO*(TE+460.)-460.))
              1 TSURF(N,IDELX) = 1.05*(EEEO*(TE+460.)-460.)
0282            T12=THS(N,IDELX)
0283            T23=TSURF(N,IDELX)
0284            KK1=FK1
0285            TINS=ABS(THS(N,IDELX)-TEM2)
0286            KK2=FK2+CINS2*TINS**1.945
0287            GO TO 140
0288        130 CONTINUE
0289            TSURF(N,IDELX)=TO(N,IDELX)
0290        140 CONTINUE
0291            T(N,IDELX)= (DAU*QINTO+CON*(XX-TT(N,IDELX))+CON*
              1 (TT(N+1,IDELX)-TT(N,IDELX))-QL *D+CRV*TT(N,IDELX))/CKV
0292            TO(N,IDELX)=T(N,IDELX)+(.5*QINTO*THICK/KK)
0293        180 CONTINUE
0294            TINT(N,IDELX)= TI
0295        190 CONTINUE
0296        200 CONTINUE
          C*
          C*       THIS IS THE END OF A LOOP FOR COMPUTING TANK SHELL TEMPERATURES
          C*
          C*    DELTNG = T(NG,IDELX)-TT(NG,IDELX)
0297            DAO=DAU*(1.5**(M-1))
0298            CON=CON*(1.5**(M-1))
0299            D=D*(1.5**(M-1))
0300            FLIQ=0.0                                                  00002410
0301            IF(THE-TILT .LT. 0.0) FLIQ = 1.0                          00002420
0302            DO 220 IDELX= 1,NX
0303            T(NEL+1,IDELX) = T(NEL,IDELX)
0304        220 CONTINUE
0305            REAL*4 ML2, MG2
0306            ML2=ML
0307            MR=0.
0308            MR1 = 0.0                                                 00002560
0309            MR2 = 0.0                                                 00002570
0310            ICOUNT=0
0311            PL1= PL
0312            PREV = QGSUM+QLSUM
0313            TG = TG+460.
0314            DO 300 IDELX= 1,NX
0315            A = VAREA(IDELX)                                          00002590
```

```
0316          IF(VENPOS(IDELX)-(IDELX-1)*DELX .LT. .75*DELX                 00002600
             * .AND. (VENPOS(IDELX)-(IDELX-1)*DELX .GT. .25*DELX)) GO TO 225
0317          A = .5* VAREA(IDELX)                                          00002620
        C*                                                                  00002630
        C*      USE ONE HALF VALVE AREA IF VALVE IS "NEAR" AN ELEMENT BOUNDARY 00002640
        C*                                                                  00002650
0318      225 IF(FLAG) 230,230,250
0319      230 IF(FLG) 245,245,235
0320      235 IF(PL-PRL) 240,260,260
0321      240 FLG= 0.0
        C*                                                                  00002740
        C      TEST FOR VALVE OPEN PRESSURE
        C*                                                                  00001710
0322      245 IF(PL-PR) 350,250,250
0323      250 FLG= 1.0
        C*
        C*      THE VALVE IS OPEN.
        C*
0324      260 HG=0.0
0325          MG2=0.0
0326          IF(FLIQ) 280,280,270
0327      270 MR1= MR1+ 11520000./TLENTH*CG*A
0328          GO TO 300
0329      280 CONTINUE
0330          TS=(TG1)*(2./(KP+1.))
0331          PS=PL*(2./(KP+1.))**(KP/(KP-1.))
0332          VC=RP*TS/(PS*144.)                                            00002830
0333          UBC=ABS(KP*G*RP*TS)
0334          UC=SQRT(UBC)
0335          MR2 = MR2 + (CG*A*UC)/VC*(3600./TLENTH)
0336      300 CONTINUE
0337          MR = MR1 + MR2                                                00002880
0338          MTOT= MTOT-MR*DELTA/3600.
        C*
        C*      TEST VALVE CLOSING PRESSURE
        C*
0339          IF(PL-PRL) 350, 310,310
        C*
        C*     THE VALVE IS OPEN,ITERATE ON HEAT INPUT FOR PRESSURE
        C*
0340      310 CONTINUE
0341          PL2=.95*PL
0342          CALL FPLT(PL2, PLT,HFT,TLT,VFT,VGT,LT,TL,VG,VF,HF,L )
0343          HFR = HF
0344          ELR = (L+EL1)/2.
0345          DML=ML+(QLSUM-ML*ELR+MR1*DELTA/3600.*ELR)/(ELR-HFR+HF1)
0346          ML2 = ML - DML
0347          IF(MTOT-ML2.LE. 0.0) GO TO 315
0348          MG2=MTOT-ML2
0349          HG=(QGSUM-(MR1+MR2)*L*DELTA/3600.+MG2*HG1+(ML-ML2)*L)/MG2
```

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      315 CONTINUE
          TSTHG=CPG*(.5*FRAC*(T(1,IDELX)-TL)+TL+460.)
          HGE=HF+L
          IF(TSTHG .LT. HGE)TSTHG=HGE
          HG=TSTHG
          GO TO 322
      320 CONTINUE
          PL2 = PL
          CALL FPLT(PL2, PLT,HFT,TLT,VFT,VGT,LT,TL,VG,VF,HF,L )
          L = (L+EL1)/2.
C*
C*          SOLVE HEAT BALANCE ON LIQUID FOR NEW MASS OF LIQUID
C*
          DML=ML+(QLSUM-ML*L+(MR1*DELTA/3600.)*L)/(L-HF+HF1)
      321 CONTINUE
          ML2 = ML  - DML
          MG2=MTOT-ML2
      322 CONTINUE
          CPG = .829 - .000298*HG - .00009*(PL2 - 50.)
          TG=HG/CPG
          VGS= (.245*TG+.05*PL2+1.15/VGS**2)/(PL2+23.)
          ML2=(VOL-MTOT*VGS)/(VF-VGS)
          MG2= MTOT-ML2
      325 CONTINUE
          QIN= ML2*(HF-HF1)+GASIN*(HG-HG1)+(MG2-GASIN)*(HG-HF1)+MR2*DELTA/
         13600.*(HG1-HF1+0.5*(HG-HG1))+MR1*DELTA/7200.*(HF-HF1)
          TEST= QIN/FRAC
          DELQ2= PREV-TEST
          IF(ABS(DELQ2) .LE.  .1*(ABS(PREV)+10. )) GO TO 330
          PL= PL2-DELQ2*(PL1-PL2)/(DELQ1-DELQ2)
          PL= ABS(PL)
          PL1= PL2
          PL2= PL
          DELQ1 = DELQ2
          ICOUNT = ICOUNT+1
          IF(ICOUNT .GT. 30) GO TO 400
          GO TO 320
      330 PL= PL2
          DELV=.0017*VOL*(TL-TL1)
          DELQ1=DELQ2
          IF(PL.LT. .99*PRL) GO TO 340
          FLG=0.
          IF(PL.GT. 1.02*PRL) FLG= 1.0
          GO TO 410
      340 M= M+1
          DELTA= DELTA/1.5
          GO TO 75
      350 CONTINUE
```

```
0394            PL2= 1.05*PLIN
          C*
          C*     THE VALVE IS CLOSED, ITERATE ON HEAT INPUT FOR PRESSURE
          C*
0395        360 CONTINUE
0396            CALL FPLT(PL2, PLT,HFT,TLT,VFT,VGT,LT,TL,VG,VF,HF,L )
0397            DIFFL = ELI - L
0398            DIFFH = HF - HF1
0399            DIFFV = VF - VF1
0400            IF(DIFFL .GT. 5.) L=EL1-5.
0401            IF(DIFFH .GT. 6.) HF=HF1+6.
0402            IF(DIFFV .GT. 0.0008)VF=VF1+.0008
0403            IF(PREV .LT. 1.0) GO TO 369
0404            L = (L+EL1)/2.
          C*
          C*      SOLVE HEAT BALANCE ON LIQUID FOR NEW MASS OF LIQUID
          C*
0405            ML2=(ML*L-QLSUM+ML*((HF-HF1)-.1851*(PL2*VF-PLIN*VF1)))/L
0406            IF(MTOT-ML2.LE. 0.0) GO TO 364
0407            MG2=MTOT-ML2
0408            HG=(QGSUM+(ML-ML2)*L+MG2*HGI-.1851*MG2*(PLIN*VGS1-PL2*VGS))/MG.
0409            GO TO 364
0410        362 CONTINUE
0411            PL2=PL
0412            CALL FPLT(PL2, PLT,HFT,TLT,VFT,VGT,LT,TL,VG,VF,HF,L )
0413            DIFFL = EL1 - L
0414            DIFFH = HF - HF1
0415            DIFFV = VF - VF1
0416            IF(DIFFL .GT. 5.) L=EL1-5.
0417            IF(DIFFH .GT. 6.) HF=HF1+6.
0418            IF(DIFFV .GT. 0.0008)VF=VF1+.0008
0419            L = (L+EL1)/2.
0420            ML2=(ML*L-QLSUM+ML*((HF-HF1)-.1851*(PL2*VF-PLIN*VF1)))/L
0421            MG2=MTOT-ML2
0422        364 CONTINUE
0423            TSTHG=CPG*(.5*FRAC*(T(1,I     )-TL)+TL+460.)
0424            HGE=HF+L
0425            IF(TSTHG .LT. HGE)TSTHG=HGE
0426            HG=TSTHG
0427            CPG = .829 - .000298*HG - .00009*(PL2 - 50.)
0428            TG = HG/CPG
0429            VGS= (.243*TG+.05*PL2+1.15/VGS**2)/(PL2+23.)
0430            ML2=(VOL-MTOT*VGS)/(VF-VGS)
0431            MG2= MTOT-ML2
0432        365 CONTINUE
0433            APV=0.1851*(ML2*(PL2*VF-PLIN*VF1)+GASIN*(PL2*VGS-PLIN*VGS1))
0434            QIN= ML2*(HF-HF1)+GASIN*(HG-HGI)+(MG2-GASIN)*(HG-HF1)-APV-0.1851*
               1(PL2*VGS-PLIN*VF1)
0435            TEST= QIN/FRAC
```

```
      DELQ2= PREV-TEST
      IF(ABS(DELQ2) .LE.   .1*(ABS(PREV)+10. )) GO TO 370
      PL= PL2-DELQ2*(PL1-PL2)/(DELQ1-DELQ2)
      PL= ABS(PL)
      PL1= PL2
      PL2= PL
      DELQ1 = DELQ2
      ICOUNT = ICOUNT+1
      IF(ICOUNT .GT. 20) GO TO 400
      GO TO 362
  369 CONTINUE
      PL2 = PL1
  370 CONTINUE
      ML=ML2
      IF(PL .GT. 1.05*PR .AND. VOLL .LT. VOL) GO TO 340
      IF(PL .GT. 1.05*PR .AND. ML .LT. MTOT) GO TO 340
      DELQ1=DELQ2
      VGS1= VGS2
      VGS2= VGS
      VOLL=ML*VF
      FLAG=0.
      GO TO 410
  400 PRINT 3334
 3334 FORMAT('   ITERATION EXCEEDED')
      GO TO 1
  410 CONTINUE
C*
C*     PRESSURE SATISFACTORY FOR VALVE ACTION AND HEAT IS BALANCED
C*
      DO 450 IDELX= 1,NX
      DO 450 K=1,NEL
      DO 420 J=2,25
      IF(TTT(J) -T(K,IDELX)) 420,420,430
  420 CONTINUE
      WRITE (6,113)  K                                          00003280
  430 Z= (T(K,IDELX)-TTT(J-1))/(TTT(J)-TTT(J-1))
      PB=Z*(PBT(J)-PBT(J-1))+PBT(J-1)                           00003300
      THSTRS= 120.0*(TO(I,IDELX)-TINT(I,IDELX))
      PALL=(THSTRS* THICK/RTANK+PL-14.7)*RTANK/5.0
      IF(PALL-PB) 460, 440, 440
  440 CONTINUE
      WRITE(6,114) T(K,IDELX),K,PL,PB,TIME
      WRITE(6,1000)
 1000 FORMAT (1H1)
      GO TO 1
  450 CONTINUE
  460 CONTINUE
```

```
C*      THSTRS=128.*(TO(1,IDELX)-TINT(I,IDELX))
        THSTRS=128.*(TO(1,1     )-TINT(1,IDELX))
        SIGC= (PL-14.7)*RTANK/THICK+ABS(THSTRS)
        SIGT=.5*SIGC                                    00003440
        TAU=.25*SIGC                                    00003450
        TIME=TIME+DELTA
        TSAV = TSAV + DELTA
        QTSUM=QTSUM+QGSUM+QLSUM
        TG=TG-460.
        IF(MOD(IPRINT,MODPNT) .EQ. 0) CALL OUTPUT
        TG=TG+460.
        IPRINT = IPRINT + 1
        IF(PLOT .NE. 1.0) GO TO 4000
C*        BYPASS PLOT SETUP
C*
        IF(TSAV .LT. TPLOT) GO TO 4000
        TSAV = DELTA
        IPLOT= 1
        TIMDAT(NPT) = TIME
        DO 3980 IDELX=1,NX
        NP= 1
        DO 3960 N=1,NEL,NRAD
C*
C*     PLOT ONLY ONE AXIAL STATION.... IF ALL ARE DESIRED UP THE DIMENSION
C*     ON TEMDAT AND LOOP THROUGH NX STATIONS.
C*
C*      TEMDAT(NPT,NP,1) = TO(N,1)
        TEMDAT(NPT,NP,IDELX)=TO(N,IDELX)
        NP= NP+1
        IF(NP.GT.25) NP= 25
 3960   CONTINUE
        IF(IPLOT .EQ. 1) NPT= NPT+1
        IF(NPT.GT. 200) NPT= 200
 3980   CONTINUE
        PDAT(NPP) = PL
        TIMPDT(NPP)= TIME
        NPP= NPP+1
        IF(NPP.GT.400) NPP= 400
 4000   CONTINUE
        DELTA=DELTA*(1.5**(M-1))
        IF(ML.LE.0.0) GO TO 1
        QLSUM=0.                                        00003550
        QGSUM=0.                                        00003560
        DO 4010 IDELX= 1,NX
        DO 4010 J=1,NEL
        TTHS(J,IDELX)=THS(J,IDELX)
 4010   TT(J,IDELX)= T(J,IDELX)
        GO TO 30
        END                                             00003850
```

```
0001          SUBROUTINE INPUT                                                   00000010
0002          COMMON IPRNT(5)                                                    00000040
0003          COMMON YLAB(41),XLAB(30), PLAB(41)
0004          COMMON SPACE(41,121)
0005          COMMON ISYM(21)
0006          COMMON XAXIS(13), YAXIS(41)
0007          COMMON/COMP/C,EO,EM,G,KK,KP,RHO,RP,SIG,CG,GAMMA,GASCON,HGT         00000020
             *,RHOSK,SKTHK                                                       00000030
0008          REAL KK,KP,MTOT,L,MG,ML,LT,MGG,MR                                  00000050
0009          COMMON/TEMP/ T( 30,6), TT( 30,6),TSURF( 30,6),LT(25),
             * THS(30,6), TTHS(30,6),      HFT(25),TLT(25),PLT(25),VFT(25),
             *VGT(25),        TTT(30),PBT(30) ,          X(4,5),S(5)             00000080
0010          COMMON/MODIFY/ HEATX(30,10,6), TEMPX(30,10,6), ANG(30), TIMET(10),00000090
             1 TILT,PITCH,NX,FRAC
0011          COMMON/VALVES/ VENPOS(6), VAREA(6)                                 00000110
0012          REAL MR1, MR2,LAGTHK                                               00000120
0013          COMMON/ON/HOFTIM(3), TOFTIM(3)                                     00000130
0014          COMMON/PRINT/MODPNT
0015          COMMON  /GENRL/ DELX,               CPTNK, EI, EFIRE, FKS,         00000140
             1 RHOTNK, CO, DELTA, HF1, MTOT, NEL, PR, THICK, RTANK, PRL, EMU,    00000150
             2 TLENTH,TOCMP,               CINS1, CINS2, TEM1, TEM2, LAGTH1,
             3 LAGTH2, FK1, FK2
0016          COMMON /PLOTS/ PLOT,TPLOT,NPT, NPP, TEMDAT(200, 25,6),TIMPOT(400),
             1          PDAT(400), TIMDAT(200), NRAD, NP,NSTA1,NSTA2
0017          CALL CLEAR(VENPOS(1), VAREA(6))
0018          CALL CLEAR(DELX, FK2)
0019          REAL*4 LAGTH1, LAGTH2
0020          READ(5,3000) XLAB
0021          READ(5,3050) YLAB
0022          READ(5,3050) PLAB
0023     3000 FORMAT(20A4/10A4)
0024     3050 FORMAT(41A1)
0025          NAMELIST/DATA / HEATX,TEMPX,TIMET,TILT,ANG,NX,PITCH,FRAC
0026          NSTA1=1
0027          NSTA2=1
0028          ENTRY INPUT1                                                       00000310
0029          READ(5,DATA,END=50)
        C*                                                                       00000330
        C*       CARD INPUTS:                                                    00000340
        C*                                                                       00000350
        C*    DATA SET "INPUT"                                                   00000360
        C*                                                                       00000370
        C*    HEATX   - HEAT DISTRIBUTIONON CIRCUMFERENTIAL ELEMENTS PER LENGTH 00000380
        C*    TEMPX   - FIRE TEMPS CIRCUMFERENTIALLY DISTRIBUTED PER LENGTH      00000390
        C*    ANG     - RADIAL LOCATIONS FOR HEATX AND TEMPX                     00000400
        C*    TIMET   - TIME TABLE FOR DURATION OF FIRE                          00000410
        C*    TILT    - ROLL ANGLE OF THE VENT VALVE FROM UPRIGHT POSITION       00000430
        C*    PITCH   - PITCH ANGLE OF THE TANK CAR                              00000440
        C*                                                                       00000450
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0030           NAMELIST /LADING/ HFT, LT, TLT, PLT, VGT, VFT, GAMMA, GASCON, HGT 00000460
0031           READ(5,LADING)                                                    00000470
         C*                                                                      00000480
         C*       CARD INPUTS:                                                   00000490
         C*                                                                      00000500
         C*    DATA SET "LADING"                                                 00000510
         C*                                                                      00000520
         C*    HFT      - SPECIFIC ENTHALPY OF SATURATED LIQUID LADING           00000530
         C*    LT       - HEAT OF VAPORIZATION OF LADING                         00000540
         C*    TLT      - TEMPERATURE VALUES FOR ENTHALPY AND VOLUME DATA        00000550
         C*    PLT      - PRESSURE VALUES FOR ENTHALPY AND VOLUME DATA           00000560
         C*    VGT      - SPECIFIC VOLUME OF SATURATED VAPORIZED LADING          00000570
         C*    VFT      - SPECIFIC VOLUME OF SATURATED LIQUID LADING             00000580
         C*    HGT      - GAS HEAT TRANSFER COEFFICIENT  ...... BTU/FT**2-HR     00000590
         C*    GASCON   - GAS CONSTANT                                           00000600
         C*    GAMMA    - RATIO OF SPECIFIC HEATS                                00000610
         C*                                                                      00000620
0032           NAMELIST /BURST/ TTT, PBT                                         00000630
0033           READ(5,BURST)                                                     00000640
         C*                                                                      00000650
         C*       CARD INPUTS:                                                   00000660
         C*                                                                      00000670
         C*    DATA SET "BURST"                                                  00000680
         C*                                                                      00000690
         C*    TTT      - TANK BURST TEMPERATURES                                00000700
         C*    PBT      - TANK BURST PRESSURES                                   00000710
         C*    FRAC -  FRACTION  OF THE TANK THAT IS INVOLVED IN FIRE - THIS
         C*            IS USED WITH A UNIFORM VALVE FOR HEATX
         C*                                                                      00000720
0034           WRITE(6,100)
0035       100 FORMAT(1H1,40X,'INPUT DATA FOR TANK CAR PROGRAM'///)
0036           WRITE(6,101)
0037       101 FORMAT(34X,'THERMODYNAMIC PROPERTIES OF SATURATED LADING'//)
0038           WRITE(6,102)
0039       102 FORMAT(9X,'LIQUID',9X,'VAPORIZATION',9X,'TEMPERATURE',8X,'PRESSURE
              1',11X,'SPECIFIC',11X,'SPECIFIC')
0040           WRITE(6,103)
0041       103 FORMAT(5X,'SPECIFIC',11X,'LATENT',51X,'VOLUME',13X,'VOLUME')
0042           WRITE(6,104)
0043       104 FORMAT(8X,'ENTHALPY',12X,'HEAT',54X,'GAS',14X,'LIQUID')
0044           WRITE(6,105)
0045       105 FORMAT(9X,'BTU/LB',11X,'BTU/LB',17X,'OF',15X,'PSIA',14X'FT3/LB',
              113X,'FT3/LB'/)
0046           WRITE(6,106)
0047       106 FORMAT(10X,'HFT',16X,'LT',18X,'TLT',15X,'PLT',16X,'VGT',15X,'VFT')
0048           WRITE(6,107) (HFT(K),LT(K),TLT(K),PLT(K),VGT(K),VFT(K),K=1,25)
0049       107 FORMAT(7X,E9.3,10X,E9.3,10X,E9.3,10X,E9.3,10X,E9.3,10X,E9.3)
0050           WRITE(6,108) GAMMA
0051       108 FORMAT(//10X,'LADING-RATIO OF SPECIFIC HEATS',7X,F6.4,/)
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0052            WRITE(6,109) GASCON
0053        109 FORMAT(10X,'LADING GAS CONSTANT (FT-LBF/LBM/OF)',2X,F8.4//)
0054            WRITE(6,110)
0055        110 FORMAT(//24X,'CORRELATION BETWEEN SHELL ELEMENT TEMPERATURE AND MA
               1XIMUM SHELL PRESSURE'//)
0056            WRITE(6,111)
0057        111 FORMAT(45X,'TEMPERATURE',10X,'PRESSURE',//49X,'OF',17X,'PSIG',/48X
               1,'TTT',17X,'PBT')
0058            WRITE(6,112) (TTT(K),PBT(K),K=1,30)
0059        112 FORMAT(30(46X,E9.3,11X,E9.3/))
0060            WRITE(6,113)
0061        113 FORMAT(//32X,'EXTERNAL HEAT TRANSFER COEFFICIENT AND FIRE TEMPERAT
               1URE'//)
0062            WRITE(6,114)
0063        114 FORMAT(31X,'HEAT TRANSFER COEFFICIENT',14X,'FIRE TEMPERATURE'/31X,
               1'FOR EXTERNAL SURFACE OF',12X,'CORRESPONDING TO ADJACENT'/29X,'TAN
               2K CAR',24X,'HEATX VALUE'/37X,'BTU/FT2/HR/OF',27X,'OF'/)
0064            WRITE(6,115)
0065        115 FORMAT(41X,'HEATX',29X,'TEMPX')
0066            WRITE(6,116) (((HEATX(I,J,K),TEMPX(I,J,K),I=1,30),J=1,10),K=1,NX)
0067        116 FORMAT(39X,1PE9.3,25X,1PE9.3 )
0068            WRITE(6,117) ANG
0069        117 FORMAT(//'  RADIAL ANGLE DISTRIBUTION OF SHELL ELEMENT CENTROIDS-D
               1EGREES-DEFINES ANGLE INCREMENTS USED IN HEATX-TEMPX TABLE'//(10(F1
               22.2)))
0070            WRITE(6,118) TIMET.
0071        118 FORMAT(//'  HEATING TIME-SECONDS-DEFINES TIME INCREMENTS USED IN H
               1EATX-TEMPX TABLE'//(10(F12.2))///)
0072            NAMELIST/GENRL/ DELX, VENPOS, VAREA, CPTNK, EI, EFIRE, FKS,       00000900
               1 RHOTNK, CD, DELTA, HFI, MTOT, NEL, PR, THICK, RTANK, PRL, EMO,  00000910
               2 TLENTH,TDCMP,          CINS1, CINS2, TEM1, TEM2, LAGTH1,
               3 LAGTH2, FK1, FK2, PLOT, TPLOT, NRAD, MODPNT,NSTA1,NSTA2
0073            READ(5,GENRL)                                                     00000940
          C*                                                                      00000950
          C*       CARD INPUTS:                                                   00000960
          C*                                                                      00000970
          C*    DATA SET "GENRL"                                                  00000980
          C*                                                                      00000990
          C*    CPTNK    - SPECIFIC HEAT OF TANK CAR SHELL MATERIAL .... BTU/LB-DEG00001000
          C*    RTANK    - RADIUS OF TANK CAR ...  FT                             00001010
          C*    EI       EMISSIVITY OF INSIDE OF TANK SHELL                       00001020
          C*    EFIRE    - EMISSIVITY OF FIRE                                     00001030
          C*    FKS      - THERMAL CONDUCTIVITY OF TANK SHELL                     00001040
          C*    RHOTNK   - DENSITY OF TANK CAR SHELL                              00001050
          C*    VAREA    - AREA OF RELIEF VALVE     ..... FT**2                   00001060
          C*    CD       - RELIEF VALVE FLOW COEFFICIENT                          00001070
          C*    DELX     - LENGTH OF EACH TANK CAR ELEMENT                        00001080
          C*    DELTA    - TIME INCREMENT IN CALCULATION .... SECONDS             00001090
          C*    HFI      - INITIAL SPECIFIC HEAT OF LADING                        00001100
          C*    MTOT     - INITIAL TOTAL MASS OF LIQUID AT LADING    .... LB/FT   00001110
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```
              C*      NEL      - NUMBER OF TANK CAR RADIAL STATIONS                  00001120
              C*      PR       - PRESSURE TO OPEN RELIEF VALVE  ..... PSF            00001130
              C*      THICK    - THICKNESS OF TANK CAR SHELL  .... INCHES            00001140
              C*      PRL      - LOW PRESSURE VALVE LIMIT                            00001150
              C*      EMO      - EMISSIVITY OF TANK CAR OUTSIDE                      00001160
              C*      VENPOS   - LOCATIONS OF VALVES                                 00001170
0074                1 CONTINUE
              C*      LAGTH1   - THICKNESS OF OUTER LAYER OF INSULATION.... INCH
              C*      LAGTH2   - THICKNESS OF INNER LAYER OF INSULATION.... INCH
              C*      FK1      - THERMAL CONDUCTIVITY OF OUTER INSULATION AT REFERENCE
              C*                 TEMPERATURE .... BTU/FT-DEGF-HR
              C*      FK2      - THERMAL CONDUCTIVITY OF INNER INSULATION AT REFERENCE
              C*                 TEMPERATURE .... BTU/FT-DEGF-HR
              C*      CINS1    - SLOPE OF LINEAR TEMPATURE VARIATION OF THERMAL
              C*                 CONDUCTIVITY FOR OUTER INSULATION
              C*      CINS2    - SLOPE OF LINEAR TEMPERATURE VARATION OF THERMAL
              C*                 CONDUCTIVITY FOR INNER INSULATION
              C*      TEM1     - REFERENCE TEMPERATURE FOR FK1 .... DEG F
              C*      TEM2     - REFERENCE TEMPERATURE FOR FK2 ....DEG F
              C*      PLOT     - IF 1.0, PLOTS REQUESTED
              C*      IPLOT    - TIME INTERVAL AT WHICH TO PRODUCE TEMPERATURE PLOT
              C*      NRAD     - RADIAL STATION INCREMENT FOR TEMPERATURE PLOT
              C*                 (25 DISTINCT PLOT SYMBOLS AVAILABLE)
              C*                                                                     00001180
0075                2 G = 32.16                                                      00001190
0076                  CE = CD                                                        00001200
0077                  RHO = RHOTNK                                                   00001210
0078                  KP= GAMMA                                                      00001220
0079                  RP= GASCON                                                     00001230
0080                  KK = FKS                                                       00001240
0081                  C = CPTNK                                                      00001250
0082                  SIG = .173E-8                                                  00001260
0083                  EM= EMO                                                        00001270
0084                  EO= EFIRE                                                      00001280
0085                  QCN=HEATX(1,1,1)*(TEMPX(1,1,1)-70.)+SIG*(EO*EM*(TEMPX(1,1,1)+460.),
                     1**4.-EM*530.**4.)
0086                  WRITE(6,119)
0087              119 FORMAT(51X,'GENERAL INPUT DATA'//)
0088                  WRITE(6,120)
0089              120 FORMAT(10X,'TANK CAR'/)
0090                  WRITE(6,121) TLENTH
0091              121 FORMAT(10X,'LENGTH (FEET)',43X,F7.3)
0092                  WRITE(6,122) RTANK
0093              122 FORMAT(10X,'INSIDE RADIUS (FEET)',36X,F7.3)
0094                  WRITE(6,123) DELX
0095              123 FORMAT(10X,'LONGITUDINAL ELEMENT SIZE (FEET)',24X,F7.3)
0096                  WRITE(6,124) THICK
0097              124 FORMAT(10X,'SHELL THICKNESS (INCHES)',32X,F7.3)
0098                  WRITE(6,125) PITCH
0099              125 FORMAT(10X,'PITCH ANGLE OF TANK CAR (DEGREES)',23X,F7.3)
```

```
0100          WRITE(6,126) FKS
0101      126 FORMAT(10X,'THERMAL CONDUCTIVITY OF SHELL MATERIAL (BTU/FT/HR/OF)'
             1,3X,F7.3)
0102          WRITE(6,127) CPTNK
0103      127 FORMAT(10X,'SPECIFIC HEAT OF SHELL MATERIAL (BTU/LBM/OF)',12X,F7.3
             1)
0104          WRITE(6,128) RHOTNK
0105      128 FORMAT(10X,'DENSITY OF SHELL MATERIAL (LBM/FT3)',21X,F7.3)
0106          WRITE(6,129) NEL
0107      129 FORMAT(10X,'NUMBER OF SHELL ELEMENTS AROUND CIRCUMFERENCE',11X,I3)
0108          WRITE(6,130) NX
0109      130 FORMAT(10X,'NUMBER OF SHELL LENGTH ELEMENTS',25X,I3)
0110          WRITE(6,156) HGT
0111      156 FORMAT(10X,'GAS HEAT TRANSFER COEFFICIENT FOR INTERNAL SHELL SURFA
             1CE (BTU/FT2/HR/OF)',F7.3//)
0112          WRITE(6,131)
0113      131 FORMAT(10X,'RELIEF VALVE'/)
0114          WRITE(6,132) (IDELX,VENPOS(IDELX),VAREA(IDELX),IDELX=1, 6)
0115      132 FORMAT(40X,'LENGTH ELEMENT',10X,'VALVE POSITION',9X,'MAXIMUM VALVE
             *',/43X,'NUMBER',14X,
             1'FROM END OF CAR',11X,'FLOW AREA'/70X,'FT',20X,'FT2'//(45X,I2,21X,
             2F6.3,17X,F6.4)//)
0116          WRITE(6,133) TILT
0117      133 FORMAT(10X,'ROLL ANGLE TO CENTERLINE OF RELIEF VALVES (DEGREES)',3
             1X,F8.3)
0118          WRITE(6,134) PR
0119      134 FORMAT(10X,'TANK PRESSURE TO OPEN RELIEF VALVE (PSIA)',13X,F8.3)
0120          WRITE(6,135) PRL
0121      135 FORMAT(10X,'LOWER LIMIT FOR RELIEF VALVE OPEN (PSIA)',14X,F8.3)
0122          WRITE(6,136) CD
0123      136 FORMAT(10X,'VALVE FLOW COEFFICIENT',32X,F8.3//)
0124          WRITE(6,137)
0125      137 FORMAT(10X,'TANK CAR SHELL INSULATION'/)
0126          WRITE(6,138) LAGTH1
0127      138 FORMAT(10X,'THICKNESS-OUTER INSULATION LAYER (INCHES)',26X,F9.4)
0128          WRITE(6,139) LAGTH2
0129      139 FORMAT(10X,'THICKNESS-INNER INSULATION LAYER (INCHES)',26X,F9.4)
0130          WRITE(6,140) FK1
0131      140 FORMAT(10X,'CONDUCTIVITY-OUTER LAYER OF INSULATION AT REF. TEMP. (
             1BTU/FT/HR/OF)',F9.4)
0132          WRITE(6,141) FK2
0133      141 FORMAT(10X,'CONDUCTIVITY-INNER LAYER OF INSULATION AT REF. TEMP. (
             1BTU/FT/HR/OF)',F9.4)
0134          WRITE(6,142) CINS1
0135      142 FORMAT(10X,'SLOPE OF VARIATION OF CONDUCTIVITY-OUTER INSULATION',1
             16X,F9.4)
0136          WRITE(6,143) CINS2
0137      143 FORMAT(10X,'SLOPE OF VARIATION OF CONDUCTIVITY-INNER INSULATION',1
             16X,F9.4)
0138          WRITE(6,144) TEM1
```

```
  144 FORMAT(10X,'REFERENCE TEMPERATURE-OUTER INSULATION (OF)',24X,F9.4)
      WRITE(6,145) TEM2
  145 FORMAT(10X,'REFERENCE TEMPERATURE-INNER INSULATION (OF)',24X,F9.4)
      WRITE(6,146) TDCMP
  146 FORMAT(10X,'DENSITY-SPECIFIC HEAT PRODUCT-INSULATION (BTU/FT3/OF)'
     1,14X,F9.4//)
      WRITE(6,147)
  147 FORMAT(10X,'INFORMATION FOR PLOTTING OF GRAPHS'/)
      WRITE(6,148) PLOT
  148 FORMAT(10X,'GRAPHS REQUESTED (1.0=YES,0.0=NO)',13X,F9.2)
      WRITE(6,149) TPLOT
  149 FORMAT(10X,'TIME INTERVAL FOR TEMPERATURE GRAPHS (SECONDS)',F9.2)
      WRITE(6,150) NRAD
  150 FORMAT(10X,'RADIAL STATION INCREMENT FOR TEMP. GRAPHS',5X,I6)
      WRITE(6,151) NSTA1,NSTA2
  151 FORMAT(10X,'GRAPHS TO BE PLOTTED AT LENGTH STATIONS',11X,I2,' AND'
     1,I2//)
      WRITE(6,152)
  152 FORMAT(10X,'INITIAL CONDITIONS'/)
      WRITE(6,153) HFI
  153 FORMAT(10X,'ENTHALPY OF LIQUID LADING (BTU/LBM)',19X,F10.2)
      WRITE(6,154) MTOT
  154 FORMAT(10X,'TOTAL MASS OF LADING (LBM/FT)',25X,F10.2)
      WRITE(6,155) QCW
  155 FORMAT(10X,'COLD WALL HEAT TRANSFER TO TANK CAR SHELL (BTU/FT2/HR)
     1',F10.2//)
      WRITE(6,157)
  157 FORMAT(10X,'EMISSIVITIES'/)
      WRITE(6,158) EI
  158 FORMAT(10X,'INSIDE SURFACE OF TANK CAR SHELL',1X,F8.3)
      WRITE(6,159) EMO
  159 FORMAT(10X,'OUTSIDE SURFACE OF TANK CAR SHELL',F8.3)
      WRITE(6,160) EFIRE
  160 FORMAT(10X,'FIRE',29X,F8.3//)
      WRITE(6,161) DELTA
  161 FORMAT(10X,'THE TIME STEP DURING COMPUTATIONS IS',F6.2,' SECONDS')
      WRITE(6,165) FRAC
  165 FORMAT(10X,'FRACTION OF THE TANK THAT IS INVOLVED IN FIRE',F6.3)
      WRITE(6,162)
  162 FORMAT(1H1)
      RETURN                                                           00001610
   50 CONTINUE
      STOP
      END                                                              00001620
```

```
FORTRAN IV G LEVEL  21                  HUNTEM              DATE = 73190          09/34/01              PAGE 0001

0001               SUBROUTINE HUNTEM (V,X,N,J)
0002               DIMENSION X(N)
            C*        V     - THE INDEPENDENT VARIABLE
            C*        X     - TABLE OF INDEPENDENT VARIABLES
            C*
0003               L = N-1
0004               DO 20 I=2,L
0005               IF(X(I-1) .LE. V .AND. V.LE. X(I) .OR.
                  * X(I-1) .GE. V .AND. V .GE. X(I)) GO TO 30
0006            20 CONTINUE
            C*
            C*        ARRIVAL HERE IMPLIES THE USE OF THE LAST VARIABLE
            C*
0007               I=L
0008            30 CONTINUE
0009               J = I-1
            C*
            C*      J POINTS TO X ARRAY FOR 3 POINT FIT INDEXING
0010               RETURN
0011               END
```

```
0001          SUBROUTINE FPLT(PL, PLT,HFT,TLT,VFT,VGT,LT, TL,VG, VF,HF, L)
0002          DIMENSION PLT(1),HFT(1),TLT(1),VFT(1),VGT(1),LT(1)
0003          REAL*4 LT,L
0004          XLAGR(C0,C1,C2,CX,U0,U1,U2)=(CX-C1)*(CX-C2)/(C0-C1)/(C0-C2)*U0-
             1(CX-C0)*(CX-C2)/(C0-C1)/(C1-C2)*U1+(CX-C0)*(CX-C1)/(C0-C2)/(C1-C2)
             2*U2
0005          DO 6 J=2,20
0006          IF(PLT(J)-PL) 6,6,7
0007        6 CONTINUE
0008          J= 20
0009        7 H1=PLT(J-2)
0010          H2= PLT(J-1)
0011           H3=PLT(J)
0012          H4= PLT(J+1)
0013          TL= XLAGR(H2,H3,H4,PL, TLT(J-1), TLT(J), TLT(J+1))
0014          VG= XLAGR(H2,H3,H4,PL, VGT(J-1), VGT(J), VGT(J+1))
0015          VF= XLAGR(H2,H3,H4,PL, VFT(J-1), VFT(J), VFT(J+1))
0016          HF= XLAGR(H2,H3,H4,PL, HFT(J-1), HFT(J), HFT(J+1))
0017          L = XLAGK(H2,H3,H4,PL,  LT(J-1),  LT(J),  LT(J+1))
0018          RETURN
0019          END
```

```
0001            SUBROUTINE OUTPUT                                                  00000010
0002            COMMON /PLOTS/ PLOT,TPLOT,NPT, NPP, TEMDAT(200, 25,6),TIMPDT(400),
               1        PDAT(400), TIMDAT(200), NRAD, NP,NSTA1,NSTA2
0003            COMMON/MODIFY/ HEATX(30,10,6), TEMPX(30,10,6), ANG(30), TIMET(10),00000140
               1 TILT,PITCH,NX,FRAC
0004            COMMON/OUTPU1/ TIME,HF,PL, TL,TI,VOLL, VOLG, MG,ML,MR,THE,
               *DANG,TINT(30,6),SIGC,SIGT,TG,QINTO,QGSUM,QLSUM,QTSUM
0005            COMMON/TEMP/ T( 30,6), TT( 30,6),TSURF( 30,6),LT(25),
               * THS(30,6), TTHS(30,6),       HFT(25),TLT(25),PLT(25),VFT(25),     00000120
               *VGT(25),       TTT(30),PBT(30) ,           X(4,5),S(5)             00000060
0006            COMMON/VALVES/ VENPOS(6), VAREA(6)                                 00000090
0007            COMMON  /GENRL/ DELX,                  CPTNK, EI, EFIRE, FKS,      00000120
               1 RHOTNK, CD, DELTA, HF1, MTOT, NEL, PR, THICK, RTANK, PRL, EMO,    00000130
               2 TLENTH,TDCMP,                CINS1, CINS2, TEM1, TEM2, LAGTH1,
               3 LAGTH2, FK1, FK2
0008            COMMON/TO/ TO( 30,6)
0009            REAL KK,KP,MTOT,L,MG,ML,LT,MGG,MR                                  00000030
0010            DIMENSION  LSET(6), STA(10)
0011            DO 30 K=NSTA1,NSTA2
0012            IF(TIME.EQ.DELTA) LSET(K)=0
0013            IUNIT =  K+7
         C**********************************************************************
         C*
         C*        IF NX = 1, IUNIT MAY BE SET TO 6 FOR SYSOUT
         C*
         C**********************************************************************
0014            IF( MOD (LSET(K),25).EQ. 0 ) WRITE(IUNIT,1000) K
0015       1000 FORMAT(1H1,50X,'AXIAL STATION NO. ', I3/2X,'TIME',2X,'TANK',3X, 'S
               1HELL',3X,'TOTAL',3X,'ANGLE',2X,'BURST',3X,'LIQUID',1X,'VAPOR',2X,'
               2RELIEF',2X,'HEAT',3X,'HEAT',3X,'ACCUMULATIVE',1X,'LIQUID',1X,'LIQU
               3ID',/8X,'PRESS',2X,'ELEMENT',1X,'MASS',4X,'TO',5X,'PRESS.',2X,'TEM
               4P.',2X,'TEMP.',2X,'VALVE',3X,'INPUT',2X,'INPUT',2X,'HEAT',1X,'INPU
               5T',3X,'VOLUME',1X,'ENTHALPY',/15X,'TEMP.-',2X,'OF',6X,'LIQUID AT',
               620X,'MASS',4X,'GAS/',3X,'LIQUID TO TANK CAR  IN',/15X,'VAPOR',3X,'
               7LADING',9X,'ELEMENT',15X,'FLOW',4X,'TIME   /TIME',15X,'TANK',/15X,
               8'SPACE',18X,'TEMP.',17X,'LBM/HR',2X,'STEP',3X,'STEP',16X,'CAR',
               9/2X,'SEC',3X,'PSIA',3X,'OF',6X,'LBM/FT',2X,'DEG.',3X,'PSIA',4X,
               *'OF',5X,'OF',5X,'/FT',5X, 'BTU/FT',1X,'BTU/FT',1X,'BTU/FT',7X,
               *'FT3/FT',1X,'BTU/LBM'/)
0016            DO 20 J=2,25
0017            IF(TTT(J)-T(1,K)) 20,20,21
0018         20 CONTINUE
0019         21 Z=(T(1,K)-TTT(J-1))/(TTT(J)-TTT(J-1))
0020            PB=Z*(PBT(J)-PBT(J-1))+PBT(J-1)+14.7
0021            WRITE(IUNIT,1010) TIME,PL,T(1,K),MTOT,THE,PB,TL,TG,MR,QGSUM,QLSUM,
               1QTSUM,VOLL,HF
0022       1010 FORMAT(   F6.0,2X,F6.1,1X,F7.2,1X,F6.1,2X,F6.2,1X,F7.2,1X,F6.1,1X,
               1F6.1,1X,F7.0,1X,F6.1,1X,F6.1,1X,F12.1,1X,F6.2,1X,F7.2/)
0023            LSET(K) = LSET(K) +1
0024         30 CONTINUE
0025            RETURN
0026            END
```

```
0001            SUBROUTINE PLOTTR (XDATA,YDATA,NPT)
          C
          C     REQUIRED INPUTS;
          C      Y-LABEL CARD (1)
          C      X-LABEL CARDS (2)
          C
          C     INPUT PARAMETERS REQUIRED;
          C      XDATA --- X
          C      YDATA --- Y
          C      NPT - NUMBER OF X,Y PAIRS IN ENITE PLOT INCLUDING OVERLAYS
          C      ISYM(I) - ENTER NUMBER OF X,Y PAIRS FOR WHICH SYMBOL I APPLIES
          C      ISYM(I+1) =0 TERMINATES PLOTS
          C
          C
          C      SUBROUTINES REQUIRED;
          C      GRID
          C      NORMAL
          C      AXSCAL SCALES TO 2,5, OR 10 DELX
          C      CALLING SEQUENCE;
          C        CALL PLOTTR(XDATA,YDATA,NPTS)
          C
          C
0002            COMMON IPRNT(5)                                          00000070
0003            COMMON YLAB(41),XLAB(30), PLAB(41)
0004            COMMON PLOT(41,121)
0005            COMMON ISYM(21)
0006            COMMON XAXIS(13),YAXIS(41)
0007            DIMENSION  SYMBOL(20)
0008            DIMENSION XDATA (1), YDATA(1)
0009            DATA PLUS/1H+/
0010            DATA SYMBOL/1H*,1H-,1H.,1H,,1HY,1H#,1H$,1H@,1H%,1H&,1H=,1H>,
               1 1H<,1H?,1H(,1H),1HA,1HX,1HZ,1HO/
0011            CALL GRID
0012            XMIN = XDATA(1)
0013            YMIN = YDATA(1)
0014            XMAX = XDATA(1)
0015            YMAX = YDATA(1)
0016            DO 5 I = 1,NPT
0017            IF ( YDATA(I) .LT. YMIN)  YMIN = YDATA(I)
0018            IF ( XDATA(I) .LT. XMIN) XMIN = XDATA(I)
0019            IF (XDATA(I) .GT. XMAX) XMAX = XDATA(I)
0020            IF (YDATA(I) .GT. YMAX) YMAX = YDATA(I)
0021          5 CONTINUE
0022            DATA SYSMAX/Z7F7F7F7F/
0023            IF(XMIN.EQ. SYSMAX .OR. YMIN.EQ. SYSMAX) GO TO 200
          C
          C       CALCULATE SCALE FACTORS BASED ON 35 SKIP INTERVALS IN THE Y-
          C     DIRECTION AND 115 TYPE BAR INTERVALS IN THE X- DIRECTION.
          C     NOTE THAT THERE ARE ACTUALLY 5 MORE INTERVALS SERVUNG AS "BUFFER"
          C     SPACE.
          C
```

```
0024              DELX = (XMAX-XMIN)/116.
0025              DELY   = (YMAX-YMIN)/35.
0026              IF(DELX.EQ. 0.0) DELX=1.0
0027              IF(DELY.EQ. 0.0) DELY= 1.0
0028              CALL NORMAL (DELX,IXPNT)
0029              CALL NORMAL(DELY,IYPNT)
            C   C PRINT 9999, DELX, DELY, IXPNT, IYPNT
0030         9999 FORMAT('  DELX, DELY, IXPNT, IYPNT ',2F10.2, 2I10)
            C
            C        "NORMAL" RETURNS DELX,DELY NORMALIZED TO X.XXXXXX
            C        "NORMAL" RETURNS A "NORMALIZED NUMBER IN POWER OF 10 NOTATION.
            C     NOW SELECT A SCALE FOR THE NORMALIZED NUMBER SUCH THAT IT IS
            C     NEAREST TO 2, 4, 5, 6, 8, OR 10.
            C
            C
0031              CALL AXSCAL(DELX)
0032              CALL AXSCAL(DELY)
            C     PRINT 1234, XMIN,YMIN
0033              XSCALE = DELX*(10.**IXPNT)
0034              YSCALE= DELY*(10.**IYPNT)
0035              IDELX = DELX
0036              IDELY = DELY
            C*
            C*   ESTABLISH PLOTTER ORIGIN AT IXIN, IYMIN
            C*
0037              XK = 0.0
0038              IF(XMIN .GT. 0.) GO TO 106
0039          100 IF(XSCALE *XK .LE. XMIN) GO TO 105
0040              XK = XK - 10.
0041              GO TO 100
0042          105 IXMIN = DELX*XK - .5
0043              GO TO 109
0044          106 IF(XSCALE*XK.GT. XMIN) GO TO 107
0045              XK = XK + 10.
0046              GO TO 106
0047          107 IXMIN = DELX*(XK-10.0)
0048          109 XK = 0.0
0049              IF(YMIN .GT. 0.) GO TO 121
0050          110 IF(YSCALE*XK .LE. YMIN) GO TO 120
0051              XK = XK - 5.0
0052              GO TO 110
0053          120 IYMIN = DELY*XK - .5
0054              GO TO 125
0055          121 IF(YSCALE*XK .GT. YMIN) GO TO 122
0056              XK = XK+ 5.0
0057              GO TO 121
0058          122 IYMIN = DELY*(XK-5.0)
```

```
      125 CONTINUE
C
C
C        NOW RE-ESTABLISH XMIN, YMIN AS THE LEFT HAND CORNER OF THE
C     GRID.
C
C
C
          XMIN = IXMIN*10.**IXPNT
          YMIN = IYMIN*10.0**IYPNT
C*    PRINT 1234, XMIN, YMIN, DELX, DELY, IDELX, IDELY, IXMIN, IYMIN
     1234 FORMAT(1X, 4F15.4,4I10)
C        NOW PREPARE TO DRAW A YAXIS AT JXX ON THE X- AXIS
C
         JXX = (-IXMIN/IDELX)+1
         IF(JXX .LE. 1 .OR. JXX .GE. 121) GO TO 7
         DO 6 K = 1,41
         PLOT(K,JXX) = PLUS
        6 CONTINUE
        7 CONTINUE
         JYY = 42-(-IYMIN/IDELY +1)
         IF(JYY .LE. 1 .OR. JYY .GE. 41) GO TO 8
         DO 70 K =1,121
         PLOT(JYY,K) = PLUS
       70 CONTINUE
        8 CONTINUE
         XSCAL = IDELX*10.**IXPNT
         YSCAL = IDELY*10.**IYPNT
         LO = 1
         LHI = 0
         DO 15 K = 1,20
         IF(ISYM(K) .EQ. 0) GO TO 16
         LHI = ISYM(K) + LHI
         DO 10 J = LO,LHI
         JXX = (XDATA(J)-XMIN)/XSCAL +  1.5
         JYY = (YDATA(J) - YMIN)/YSCAL +  1.5
         IYY = 42 - JYY
         IF(IYY .LT. 1 .OR. IYY .GT. 41) GO TO 10
         IF (JXX .LT. 1 .OR. JXX .GT. 121) GO TO 10
         PLOT(IYY,JXX) = SYMBOL(K)
       10 CONTINUE
         LO = LHI + 1
       15 CONTINUE
       16 CONTINUE
         DO 21 I=1,13
         XAXIS(I) = IXMIN +(I-1)*10*IDELX
       21 CONTINUE
         DO 22 J = 1,41,5
         K = 42-J
         YAXIS(K) = IYMIN+(J-1)*IDELY
```

```
   22 CONTINUE
   25 WRITE(6,2000) IYPNT,IXPNT
 2000 FORMAT(1H1,  50X,11HYSCALE=10**,I2,5X,11HXSCALE=10**,I2,///)
      DO 30 J = 1,41,5
      K = J+1
      L = J+4
      WRITE(6,2005) YLAB(J),YAXIS(J),(PLOT(J,I), I=1,121)
 2005 FORMAT(1X,A1,F10.0,121A1)
      IF(J .EQ. 41) GO TO 30
      DO 29 M = K,L
      WRITE(6,2010) YLAB(M),(PLOT(M,I),I=1,121)
 2010 FORMAT(1X,A1,10X,121A1)
   29 CONTINUE
   30 CONTINUE
      WRITE(6,1010) XAXIS
 1010 FORMAT(/3X,13(F10.0)/)
      WRITE(6,1015) XLAB
 1015 FORMAT (30A4)
      RETURN
  200 WRITE(6,1050)
 1050 FORMAT('   PLOT DIAGNOSTIC **** PLOTTER ORIGIN AT SYSTEM MAXIMUM
     * *** CHECK INPUT TO PLOT.')
      RETURN
      END
```

```
FORTRAN IV G LEVEL  21              AXSCAL              DATE = 73190          C9/34/01              PAGE 0001

      SUBROUTINE AXSCAL(X)
      IF( 8.0 .LT. X .AND. X.LT.10.0) X=10.0
      IF(6.0.LT.X.AND. X.LT. 8.0) X= 8.0
      IF(5.0 .LT. X .AND. X.LT. 6.0) X=6.0
      IF(4.0 .LT.X .AND. X .LT. 5.0) X=5.0
      IF(2.0 .LT. X .AND. X.LT.4.0) X= 4.0
      IF(X.LT. 2.0) X=2.0
      RETURN
      END
```

```
FORTRAN IV G LEVEL  21                  NORMAL            DATE = 73190          09/34/01                PAGE 0001

            SUBROUTINE NORMAL (XN,IPNT)
            K = 0
            XIN = XN
          1 IF(ABS(XN) .LT. 1.0) GO TO 10
            IF(ABS(XN) .GE. 10.0) GO TO 20
      C
      C
      C   FALL THROUGH IMPLIES 1.0 .LE. ABS(XN) .LT. 10.0
            IPNT = -K
            IF(K .EQ. -0) IPNT=0
            RETURN
         10 K = K+1
            XN = 10.0**K*XIN
            GO TO 1
         20 K = K-1
            XN = 10.0**K*XIN
            GO TO 1
            END
```

```
FORTRAN IV G LEVEL  21                GRID               DATE = 73190          09/34/01              PAGE 0001

                 SUBROUTINE GRID
                 COMMON IPRNT(5)                                                  00000070
                 COMMON YLAB(41),XLAB(30), PLAB(41)
                 COMMON PLOT(41,121)
                 COMMON ISYM(21)
                 COMMON XAXIS(13),YAXIS(41)
                 DATA BLANK/1H /
                 DATA PLUS/1H+/
                 DO 5 I = 1,41
                 DO 5 J = 2,120
                 PLOT(I,J)= BLANK
               5 CONTINUE
                 DO 16 J = 1,41,5
                 K = 42 - J
                 DO 15 I = 1,121,5
                 PLOT(K,I) = PLUS
              15 CONTINUE
              16 CONTINUE
                 DO 18 I = 1,41
                 PLOT(I,1) = PLUS
                 PLOT(I,121) = PLUS
              18 CONTINUE
C                INSERT GRID
                 RETURN
                 END
```

```
      SUBROUTINE PLOTR
      COMMON IPRNT(5)                                                   00000070
      COMMON YLAB(41),XLAB(30), PLAB(41)
      COMMON PLOT(41,121)
      COMMON ISYM(21)
      COMMON XAXIS(13),YAXIS(41)
      COMMON /PLOTS/DUMMY,TPLOT,NPT, NPP, TEMDAT(200, 25,6),TIMPDT(400),
     1        PDAT(400), TIMDAT(200), NRAD, NP,NSTA1,NSTA2
      COMMON/MODIFY/ HEATX(30,10,6), TEMPX(30,10,6), ANG(30), TIMET(10),00000140
     1 TILT,PITCH,NX,FRAC
      COMMON  /GENRL/ DELX,                 CPTNK, EI, EFIRE, FKS,      00000190
     1 RHOTNK, CD, DELTA, HF1, MTOT, NEL, PR, THICK, RTANK, PRL, EMD,   00000200
     2 TLENTH,TOCMP,              CINS1, CINS2, TEM1, TEM2, LAGTH1,
     3 LAGTH2, FK1, FK2
C*
C*     NOTE THAT NX MUST BE 1 FOR COMPATIBILITY WITH SIZE OF TEMDAT
C*
      NX= 1
      DO 20 M= NSTA1, NSTA2
      ISYM(1) = NPT
      ISYM(2) = 0
C*
C*
C*        SINGLE PLOTS ONLY REQUIRED
      DO 10 L=1,NP
      CALL PLOTTR(TIMDAT,TEMDAT(1,L,M),NPT)
      LEVEL= 1+ NRAD*(L-1)
      WRITE(6,1000) M, LEVEL
 1000 FORMAT(//10X,'AXIAL STATION NO. ',I5,10X,'CIRCUMFERENTIAL STATION
     *NO.',I5)
   10 CONTINUE
   20 CONTINUE
C*
C*     NOW PLOT PRESSURES
C*
      ISYM(1) = NPP
      DO 30 I=1,41
   30 YLAB(I)= PLAB(I)
      CALL PLOTTR(TIMPDT,PDAT,NPP)
      RETURN
      END
```