A STUDY TO REDUCE THE HAZARDS OF TANK CAR TRANSPORTATION

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16. Abstract

The report details the findings of a 4-month study contract directed at reducing the hazards of tank car transportation. Principal objectives were to (1) define thermal inputs and associated vapor generation rates for hazardous materials transported in tank cars when subjected to fire exposure, (2) develop performance specifications and conceptual design and application requirements for safety devices preventing catastrophic car failures, and (3) formulate a research program for the design and test verification of recommended safety devices. Prime effort was directed toward the prevention of catastrophic rupture of large-capacity pressure-type cars.

A number of shortcomings with existing safety-relief specifications were indicated. A key finding was that the controlling condition in sizing for propane relief should be the liquid feed, or "upset" car condition, and not vapor feed per the current criterion. The net result is a significant undersizing of relief area considering the existing heat flux criterion to be accurate. Analytical studies and review of test data indicate the existing heat flux criterion to be significantly low--further increasing the possibilities of overpressure.

A staged safety relief system was recommended for cars with liquefied compressed gas ladings. The primary relief element would be a pressure-maintaining system sized for handling abnormal operating conditions other than severe fire exposure. The secondary relief system would be a "dump" type to drop system pressures to levels preventing catastrophic rupture and "rocketing" under severe fire exposure conditions.

Both model and full scale test programs are recommended.

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TANK CAR OF LIQUEFIED PETROLEUM GAS RUPTURING VIOLENTLY AT CRESCENT CITY, ILLINOIS ON 21 JUNE 1970.

Section 1
BACKGROUND AND OBJECTIVES

A large portion, estimated by one source as 50%, of all chemical cargoes in the United States are transported by rail [1]. A significant portion of this total is transported in the approximately 177,000 tank cars in service on the nations railroads [2]. In 1968, the probability of a given shipment of hazardous material handled by rail being involved in an accident resulting in unintentional release of contents was reported to be slightly greater than 1 in 1000, a probability which was less than the approximate figures of 3 per 1000 and 19 per 1000 for barges and trucks, respectively [3]. However, derailment trends have been increasing.

On 27 June 1970, a four-month study contract (DOT-FR-00028) was awarded by the Federal Railroad Administration (FRA) to Cornell Aeronautical Laboratory, Inc. (CAL) as part of an overall program directed toward reducing the hazards of tank car transportation. The contract announcement on 9 July 1970 included a statement by FRA administrator R.N. Whitman that

> "in view of increasing number of serious rail accidents in the past few years involving hazardous materials, it is clear that we must direct our efforts at minimizing the chances for catastrophe when tank cars are exposed to fire and/or derailment conditions."

Formal contract activity began 13 July 1970. On 7 August 1970, CAL personnel met with FRA representatives in Washington to review the original CAL proposal and to obtain additional clarification and definition regarding the distribution of CAL effort among the several tasks of the program. Prime effort was to be directed toward the prevention of catastrophic rupture of large-capacity pressure-type cars exposed to derailment and fire.

The principal objectives of the program were to:

- define thermal inputs and associated vapor generation rates for hazardous materials transported in tank cars when subjected to fire exposure,
- develop performance specifications and conceptual design and application requirements for safety devices which will prevent catastrophic tank car failures, and
- formulate a research program for the design and test verification of the recommended safety devices.

Section 2
INTRODUCTION

2.1 DESIGN PHILOSOPHY

When undertaking the design or redesign of some functional entity, be it a tank car or a concrete block, a certain basic underlying design philosophy applies. It is fundamental and noncontroversial that the design meet all normal operational conditions for a useful period of time. Additionally, provision should be made to account for design or material uncertainties, degradation during a useful life span, and to safely respond to any reasonable combination of extreme conditions which might exist during normal usage.

Consideration of extreme conditions must include an analysis of potential "threats" and their probability of occurrence. Concomitant with this analysis, the mode and consequences of failure should be studied to arrive at a design providing the desired level of protection. Note that "design" refers to the overall system design--not simply a specific safety device. Seldom will an "add-on" philosophy to provide safety protection result in a satisfactory design.

The ultimate design goal from the safety standpoint is to provide a system which will survive, or fail in a safe manner regardless of the severity of the environment. This goal is seldom practical from an economic standpoint, even if it should be practical from a technical viewpoint. It is difficult, therefore, to arrive at a consensus in selecting a level of protection. Since the question of expense versus safety is involved, the argument essentially evolves to an "it depends on whose ox is gored" theme.

The greater emotional response of persons toward infrequent, but spectacular high loss incidents, versus more frequent but lower loss incidents cannot be ignored. Attempts to "place in perspective" by comparing total losses over long periods of time will likely be received unsympathetically. Therefore, design for prevention of catastrophic loss must be taken into

account and the decision cannot be based on simple loss per unit time statistical comparison with other activities.

A number of codes have been established in an attempt to provide for reasonable margins of safety in design. Such codes should be used as a guide, not a "crutch". Technological change may justify designing to other than specific code values, but not to the intent of the code provision for adequate margins of safety. Encroachment on this margin should be avoided. Codes which use "rules-of-thumb" or are otherwise strongly empirical should be used cautiously, particularly if materials, types of construction or sizes are markedly changed from the period from which the code was derived. Following the code to the letter without consideration of a new set of conditions subvert the intent of the code completely.

No code is a substitute for sound engineering.

2.2 OVERALL TANK CAR SAFETY PROBLEM

In this section we will briefly discuss basic causative elements which can lead to catastrophic rupture of tank car shells employed in the transportation of hazardous materials, given the derailment environment. The current program deals principally with safety-relief systems. However, it is appropriate to discuss overall design problems related to tank car safety--not from the standpoint of discovery--but simply to place the current study in context with the overall system problem.

2.2.1 Problem Areas:

Research conducted during this program, and information obtained from other sources suggest a set of parameters influencing the failure potential of tank cars loaded with certain hazardous materials when exposed to the derailment environment. Some of these parameters are:

- Degree of resistance of the shell to intense thermal loading
- Safety relief valve capacity
- Filling density of the lading
- Hazardous nature of the lading
- Interrelationship of the above parameters leading to a paradox of adjusting one factor to improve a situation causing aggravation of another (e.g., increasing chance of vapor space overheating by lowering filling density to reduce chance of introducing regenerative heating mode in a fire).
- Resistance of the tank shell to impact, including potential brittle behavior of the shell material at low temperatures [4] (within the anticipated ambient temperature range in areas served by these cars).
- Fatigue strength...resistance of tank structure/materials to the effects of continued vibration.
- Quality control (example problem area being the possible existence of "stratification" in the steel plate material from which the tanks are fabricated [5]).

In a number of situations, several factors could combine to cause a catastrophic failure, with the degree of involvement of the individual factors varying widely. Probably the single most important element, from a consideration of the possibilities of catastrophic rupture, is the presence of a large thermal load due to fire exposure. Under certain conditions, mechanical impact can also produce immediate, complete shell rupture.

2.2.2 Approach to Solution:

The present study was directed primarily at the prevention of catastrophic failure of large, stub sill tank cars transporting hazardous materials. The development of safety-relief flow capacity requirements are general, however, and are not restricted to stub sill design. It is essential to consider all factors involved in a total system analysis, if a truly meaningful solution is to be generated.

Postulated solution inputs to be evaluated would include the following:

- Adding thermal protection to the tank shell
- Changing the setpoint and/or flow capacity of the safety relief devices
- Changing the number and/or type and location of safety relief devices
- Increasing impact resistance or adding protection to areas of tank shell which experience a significant puncture rate during mishaps
- Reviewing appropriateness of allowable filling densities
- Reviewing the metallurgy of shell materials for appropriateness to tank car application ("Metallurgy" in its broadest sense, including fabricating procedures)

A number of the above considerations have been explored in this, and other studies. However, additional effort will be required to more fully define appropriate solutions.

2.3 SUMMARY CONDENSATION

The following sections of this report delve into a number of the aspects of tank car safety in rather elaborate technical detail, with supporting equations and other mathematical manipulations. This section is designed to present the reader with the essence of the report without his being burdened with complete mathematical formulations or, on the other hand, being restricted to a skeletal abstract. It is essentially a narrative with certain conclusions and key points drawn from the technical text.

The sequence of events that typically may occur in a Liquefied Petroleum Gas (LPG) tank car being heated, and equipped with a properly operating safety relief valve is as follows: As the temperature of the liquid increases, the liquid volume increases due to thermal expansion until the car fills.

Continued expansion causes the internal pressure to increase and the safety relief valve begins to discharge the lading at the start-to-discharge setpoint. If the flow capacity is sufficient, the internal pressure of the car will remain within the range bounded by the flow rating pressure (maximum) and the vapor-tight pressure (minimum). If the valve capacity is not sufficient, the pressure in the car will continue to increase. (Note: Similar behavior occurs with partially liquid-filled cars, except there is no compressed liquid state and discharge begins when the vapor pressure reaches the relief valve setpoint.)

The existing specifications governing safety-relief flow requirements for cars containing liquefied compressed gases are considered inadequate. A fundamental difficulty arises from the fact that the existing specified safety-relief valve capacity tests are based on gas phase flow. The values thus obtained can be grossly different from the actual relieving capacity under certain operating conditions, e.g. where a change of state from liquid to vapor occurs in the passage of material through the valve. Additionally, the flow capacity requirement formulas are based upon maintenance of a balance between the heat input into the liquid lading and the heat withdrawn from it by vapor generation within the car -- and are therefore only applicable if the relief orifice is communicating with a vapor space. In the derailment environment, tank cars are frequently overturned. This causes liquid feed conditions to occur at the safety valve which can exist for a substantial period of time after discharge through the valve begins. We will refer to this condition as the "liquid" case, although it must be recognized that at the exit port of the valve, flow may be entirely gaseous, mixed gas and liquid, or entirely liquid, depending upon the properties of the lading at flowing conditions.

If it could be shown that the assumption of all gas phase flow resulted in conservative values to the side of safety, use of gas phase formulas could still be justified. However, such is not the case. A specific calculation for propane indicates that a valve sized for the gas feed case will be significantly undersized for the liquid feed case, even though the mass flow (assuming equal discharge coefficients for both gas flow and liquid flow) is greater for the liquid case for any given orifice size. There are several common assumptions which, on the basis of a casual analysis, could lead

one to the conclusion that the liquid case is not controlling for a propane feed, or for that matter, any material. These are not discussed elsewhere in the report, so we will elaborate on them at this time. An example is the assumption that equality of flow on a mass basis will assure safety given either liquid flow or gas flow. This is not true.

To extract heat on a constant-temperature (hence pressure at boiling conditions) basis, vaporization must occur. To accommodate internally the high specific volume requirements for a gas, a large weight quantity of low specific volume liquid must be discharged to provide "free" volume for a small weight quantity of vaporizing gas. If this fact is recognized, the question then becomes: "Can the valve pass the required additional flow of liquid?". One could proceed through the rationale that with a liquid, the mass flow will be greater than with gas flow conditions due to the greater pressure drop (sonic flow conditions no longer prevailing) and increased density of the material. "Plugging in" appropriate quantities in certain standard orifice flow equations can apparently assure that the liquid flow though a given orifice is sufficient to offset the increased requirement. However, it is necessary to assure that the stated or tacit assumptions built into these equations are valid for the case at hand. Some equations are deceptively general, being described as satisfactory for gas and liquid flow calculations. An example of such an equation is the ASME Research Committee on Fluid Meters weight rate of discharge equation for "use with either gases or liquids" (5.a.). In operating regions near the critical conditions of a particular fluid, or where changes of state may occur, formulas not taking into close consideration the thermodynamic properties of the flowing fluid are inaccurate. Assurance of applicability can be determined from the fundamental Bernoulli equation and the equation of continuity (a mathematical statement of the conservation of mass).

For propane at saturation conditions (or compressed liquid in tank car operating ranges), a larger relief orifice is dictated by the liquid case. This would not necessarily be the case with all materials. Therefore, both liquid and gas feed cases should be determined in sizing safety-relief systems.

An additional shortcoming of the existing specifications is that they fail to relate high-temperature performance characteristics of shell constructional materials, and insulation (if used) to safety-relief requirements in order to establish consistent levels of protection. The safety-relief system is not necessarily the controlling factor in assuring vessel integrity under abnormal conditions, e.g. the high temperature allowable stress of the vessel may be less than the potential stress at the relief system set-point.

In reviewing the theoretical and empirical bases for the A8.01 release capacity formula in the AAR Specifications for Tank Cars, particular attention was placed on the $34,500\ A^{0.82}$ expression* for determining the total heat input to a car exposed to fire. It was noted that this formula predicts effective flux rates of 8000 to 10,000 Btu/h/ft^2 over the range of the tank car sizes, and not 34,500 Btu/h/ft^2 -- the local peak flux rate. Analysis of the historical bases for estimating area exposed to fire as a function of the vessel size indicated that the $A^{0.82}$ factor was the product of a misleading correlation technique and is unsupported by theory or actual test data. The findings indicate that the effective flux level determined by the existing relief formula is significantly low, probably by a factor of at least two. It is important to note that the previously stated conclusion regarding inadequate relief capacity for propane under given liquid feed conditions is in no way predicated on a presumption of greater flux levels than considered in the existing flow capacity formulas. Higher flux levels represent an additional burden, further increasing the probability of a car failure. The consequences of underestimation of the peak thermal load are not necessarily as critical as the magnitude of underestimation might imply due to inherent thermal capacitance of the car and lading. However, underestimation of thermal load can be particularly critical with liquefied compressed gas ladings. The effect of inadequate relief capacity -- overpressure -- as a contributor to car failure

*Basis for this formula may be found in Section 5.3.

can be effectively masked by the existence of fire and mechanical damage. Common post-accident testing, such as determination that safety relief valves were operable, will not reveal this condition.

Several potential car failure modes exist, given fire exposure conditions. A predominant mode would be failure of the shell under internal pressure loading due to loss of strength from overheating of metal over the vapor space. This type of failure has been observed in a number of incidents. Failure could occur below or above relief system set pressure, depending on fire intensity and the area of involvement.

There is a possible failure mode that is particularly serious, because very low thermal fluxes would result in car failure. Safety-relief "pop" valves designed for vapor relief may operate in a proportioning mode in flashing liquid service due to a change in back pressure in the discharge channel. Failure would occur while the valve is discharging. The occurrence of failure could be many hours after initial fire exposure, even though the sustaining heat load from initial exposure to time of failure would appear inconsequential. This type of failure mode is indicated by some observations of accidents involving tank cars (see page 141).

Existing valve flow capacity tests, including those which have been performed in post-accident testing, do not cover conditions suitable for proving or disproving the possibility of altered functioning mode in liquid or "flashing" liquid service. Therefore, actual relief capacity for a condition likely to occur in a derailment is unknown.

A third mode of possible failure is due to compressed liquid. Cars loaded with liquefied compressed gas to authorized filling density reach shell-full conditions due to thermal expansion and condensation at pressures below the safety-relief setpoint. Once shell-full condition is reached, and as long as it is maintained, metal overheating is generally no longer a threat. Under continued thermal loading, the liquid becomes slightly compressed (subcooled)

as the pressure rises to the safety-relief valve setpoint and discharge begins. If the thermal flux to the car is high enough, the relief valve cannot pass sufficient fluid to accommodate the liquid thermal expansion and the car will fail hydrostatically. In the case of propane, very high fluxes are required for failure to occur, but the period of high level flux required for failure is very short. A momentary flare-up in a fire exposing a shell-full car may produce failure. Again, if the relief valve is operating in a proportioning liquid relief mode rather than "popping" to full opening upon actuation, failure may occur even at low heating rates if heating continues for a long enough time.

The ultimate safety goal is to prevent derailment, fire and loss. Prevention of violent rupture and the phenomenon of "rocketing" is a step toward that objective. Rocketing, which is the excursion of multi-ton portions of tank cars over extended distances after violent rupture under fire exposure conditions, has been observed in a number of accidents.

In the case of liquefied compressed gases, rocketing may be expected to occur in pressurized cars regardless of the fill state at the time of rupture: all liquid, all vapor, or any intermediate mixture. It should be further recognized that the thrust developed will generally not be derived from a combustion process, but will occur because of the violent expansion of the lading*. To prevent rocketing in cars which have no intrinsic ability to

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* In the case of LPG, for example, it is not possible that combustion will add to the thrust. The reason for this fact is the lack of sufficient air for any combustion prior to the escape of fuel from the effective "nozzle" (the open end of the tank car). Combustion of fuel exterior to the nozzle provides no thrust -- although the plume may be spectacular. Little comfort can be derived from knowledge of the absence of combustion derived thrust since the impulse arising from phase change is extremely large. There are auto-oxidizable materials which can produce thrust from a combustion process. Spontaneous polymerization could also produce "rocketing", but frequently the speed and intensity of the reaction would completely shatter the car. Ethylene oxide is an example of a compound shipped by rail which is capable of undergoing either auto-oxidation, or spontaneous polymerization.

inhibit fracture propagation, internal pressure in the car must be sharply lowered before the shell integrity is threatened due to overheating.

It is recommended that a staged safety-relief system be adopted for liquefied compressed gas service, with an exception for certain highly toxic materials. The primary stage would consist of a pressure-maintaining device (e.g., like a safety-relief valve) sized to protect the car under abnormal operating conditions other than severe fire. High pressures resulting from faulty purging, overfilling (at moderate rates), and high local ambient temperatures would be examples of this condition. The secondary stage, with a higher setpoint than the primary device, would be a pressure activated nonshut-off relief device (e.g., like a rupture disc). The intent of the secondary relief device is to reduce the internal pressure in the car to safe levels before shell integrity is threatened from severe fire exposure. In the context of present car construction, the recommended capacity of the relief systems is a compromise. There are certain combinations of potential fire intensity and envelopment that could still fail the car. For a higher probability of survival, the car should have high temperature thermal protection, though not necessarily insulation in the sense of past construction exemplified by 105 series cars.

No safety-relief system without a sophisticated intelligence system can detect all forms of car damage which may cause the car to rupture below the system setpoint. It would seem that a fruitful area for further research would be pressure shell design which would inherently inhibit runaway propagation of fractures.

Present safety valves must be tested under conditions which will realistically indicate their effectiveness as safety devices on tank cars. Test conditions that must duplicate, or properly simulate, those of a valve on a tank car in a fire include: pressure and temperature of the lading, liquid and gas phases at the valve entrance, rate of increase of pressure, exit flow conditions, and, most importantly, the internal flow geometry of the test valves.

Scaled-model studies are suggested for design and test verification of recommended safety devices. In conjunction with these tests, it is recommended that the computer simulation studies begun on this project be expanded as a prelude to eventual full-scale testing.

Section 3
SAFETY-RELIEF REQUIREMENTS

3.1 INTRODUCTION

Current Department of Transportation (DOT) regulations in Title 49, Code of Federal Regulations, Parts 170-189, also published as T.C. George's Tariff No. 23 [6] state that safety-relief valve sizing should be accomplished per instructions of Appendix "A" of the Association of American Railroads Specifications for Tank Cars [7].

Review and analysis of the current safety-valve flow capacity formulas are presented in subsequent paragraphs, along with a summary of their historical development. The existing formulas and associated specifications are considered inappropriate in a number of important respects. The underlying fundamental principles required for establishing satisfactory relief requirements are developed in Section 5 of this report and a candidate specification is presented in Section 7.

Although the earliest effort to quantify safety-relief requirements for tank car tanks was that of the AAR Bureau of Explosives, the existing safety-relief formulas arose from requirements to protect refinery process and storage vessels. The requirements for a tank car tanks and refinery process and storage vessels are not necessarily the same. Any formulas for such general application should be very conservative in nature. A review of the tank car transport environment will bring into perspective the similarities and differences between this environment and those of process and storage vessels. First and foremost among the differences is that the tank car is not a fixed object. The mechanical damage potential from a derailment has no counterpart in fixed vessel service. Vessels in a fixed place may be protected from kinetic fragments by shielding--the size and weight of which are not the economic factor they are in a transport vehicle. The orientation of the relief valve may be presumed to be fixed in a stationary vessel. A refinery process vessel

may receive a particular degree of protection in proportion to the danger it poses when it fails. Unlike a pressure vessel in a fixed facility, we must presume a tank car is in close proximity to human habitation and poses a life hazard. In terms of fire control, it must be presumed that effective fire-fighting measures will not be taken in the event of a tank car fire. Fixed spray systems and monitor nozzles available in the refinery are not available in the field, and during severe fires, fire-fighting personnel frequently have to abandon the fire ground. On a fixed setup, drainage may be arranged to insure that fuel will not tend to pool around an unfired pressure vessel. No such control is available for tank cars. In summary then, the tank car environment is an uncontrollable variable and is potentially very severe. Therefore, it requires stringent protection measures.

At this point, it is appropriate to quote Mr. Frank Heller, a member of the AAR Tank Car Committee, on the basic considerations for protecting pressure containers [8].

"Safety relief devices are used on containers to prevent rupture of the containers under certain abnormal conditions of exposure and use, such as external sources of heat, improper charging, or internal reactions. In considering sources of heat, one must allow not only for solar heat and radiation from sources of heat in close proximity, but also for exposure to and even complete envelopment in fire...

While all of the factors enumerated must be considered in selecting and sizing safety relief devices, probably the most serious hazard to which a container may be subjected is that which accompanies exposure to external fire. Therefore, from the standpoint of protecting a container from excessive internal pressure, external fire conditions are used in determining the required relieving capacity for safety relief devices...

In the sizing of the safety relief device one must make allowance for any or all of the previous conditions of exposure or use of the container and, in addition, choose the type of safety device to be used."

The safety-relief valve is the primary safety device used on currently operating tank cars. The safety-relief valve is a pressure-operated device opening at a preset pressure and should have a full capacity to prevent an excessive pressure accumulation in the open position. An advantage of this device is that it will not release the entire contents of the container if

the pressure decreases. However, the retained pressure can be a limitation when the application of heat weakens the vessel to the point where its rupture pressure is less than the operating pressure of the device. Frangible discs, commonly called rupture discs, are also pressure-operated devices. The discs are designed to rupture at a specific pressure level, and once ruptured, will continue to relieve until ambient pressure exists.

The fusible plug is a thermally operated device, and like the frangible disc, is a "go, no-go" device. Sufficient heat input to melt the fusible metal is necessary for proper functioning of this device. Therefore, the location and distribution of the devices are important considerations.

Certain combinations of devices are also available.

The requirements for any safety-relief device to operate properly are:

- the physical integrity of the surrounding vessel must be sound at the system operating levels,
- relief devices must be orientation insensitive or properly distributed,
- the system must have the ability to relieve at peak, or near-peak vapor generation rates, with due consideration given to liquid as well as vapor venting situations, and
- untimely failure potential must be close to nil.

3.2 HISTORICAL DEVELOPMENT OF RELIEF REQUIREMENTS

In 1928, John H. Fetterly of the Bureau of Explosives conducted tests with a 300-gallon propane-filled tank. The heat source consisted of kerosene-soaked wood. An analysis of Fetterly's work [9] indicated the experiments yielded heat fluxes on the order of 20,000 to 23,000 Btu/h/ft^2 and

effective flame temperatures of 1400°F. As a result of his experiments, Mr. Fetterly developed an orifice flow formula for any gas, a basis for estimating the probable heat input to a storage vessel, and flow requirements such that the safety valve would maintain a predetermined pressure--a balance between the heat input into the gas, and the heat withdrawn from it by vaporization. A 1200°F exterior surface temperature was assumed for the determination. Further description of Fetterly's work may be found in References 8 and 9.

In 1943, Messrs. Duggan, Gilmour and Fisher [9] presented a paper to the American Society of Mechanical Engineers (ASME) which reviewed previous work in determining requirements for relief-valve capacity upon exposure to fire. A summary of their findings follows:

- In the 1930's, the American Petroleum Institute (API) utilized an arbitrary value of 6000 Btu/h/ft^2 for protection of a 1000 ft^2 vessel. As a result of actual experience by the authors, it was determined that designing to that flux level provided totally inadequate protection.

- In addition to the Fetterly test previously described, the authors analyzed an Underwriters Laboratory (UL) test in 1938 where an 8-ft x 3-ft x 1/8-in. steel plate was exposed to a gasoline fire produced in a 3-ft^2 pan. Flux density from this test was 32,300 Btu/h/ft^2. The authors also described tests by Alcoa in 1930 with 150-gallon aluminum tanks exposed to a hydrocarbon fire. The flux level was 20,100 Btu/h/ft^2 using an estimate of the exposed area.

- During the years 1938 to 1940, fire exposure tests were conducted by the authors at Carbide & Carbon Chemicals Company. Three thousand gallon test tanks, 7 ft in diameter x 11 ft 6 in. high, placed 5 ft above the fire pan were used. The test tanks were surrounded by an asbestos sheet which tended to reduce the wind and act to increase radiation. The authors felt that any added radiation would be counteracted by the cold-draft action of the arrangement. The fuel used was liquefied hydrocarbon released through a nozzle network. (Note: This system gave good area coverage, but a very shallow flame depth.)

- The authors stated that they had observed effective flame temperatures of 1600°F with hydrocarbons, and that other sources have determined point temperatures up to 3000°F in open gasoline fires. They further noted that in fires of less than one-hour duration, spilled liquids in the open have melted brass (melting point approximately 1600°F).

- The summary recommendation of the authors was that a thermal flux level of 20,000 Btu/h/ft^2 of wetted surface should be used in establishing relief requirements.

Comments on the Duggan, Gilmour and Fisher paper in the ASME Transactions included some of the following. A.B. Guise stated that a Standard of New Jersey test on 1000-gallon tanks yielded a thermal flux of 24,000 Btu/h/ft^2. Mr. F.L. Maker stated that Standard of California obtained 25,000 Btu/h/ft^2 in a test of gasoline fire surrounding a tank of water. It was his belief that this rate was applicable only for complete exposure, and he felt that larger vessels would have less exposure. He cited the Stroop (API) formula, "admittedly arbitrary," where $q = 48{,}000\ A^{2/3}$. Other critics also suggested that the constant flux factor suggested by the authors was inaccurate and that a relationship to tank size should exist. The authors replied that codes should be specific about size and shape of the vessel in relation to heat input from fire, or be designed for the worst condition.

Table 1 summarizes the Duggan, Gilmour and Fisher tests.

Table No. 1

RESULTS OF DUGGAN, GILMOUR AND FISHER TESTS

RUN NO.	FLAME TEMPERATURE	q	REMARKS
1	1500°F	25,900 Btu/h/ft^2	---
2	1274°F	17,300 Btu/h/ft^2	RAIN
3	1317°F	18,700 Btu/h/ft^2	RAIN
4	1330°F	16,850 Btu/h/ft^2	UNPROTECTED TANK

Presumably as a result of the 1943 paper of Duggan, Gilmour and Fisher, the National Fire Protection Association (NFPA) committee on gases set up a subcommittee to review the matter of pressure vessel relief area. In 1944, the chairman of that committee proposed a set of relief areas which he worked out using a 20,000-Btu/h/ft^2 heat input rate to a fire-exposed vessel. As a result, the relief areas were much greater than those contained in the then current Pamphlet 58 of NFPA. One reason for this difference is that Chairman James did not make any allowance for relatively reduced fire intensity for large tanks. Reference 8 has additional information on subsequent modifications of the 1944 proposal to the NFPA, and the 1947 conversion to flow ratings rather than orifice size requirement for safety relief.

In November 1944, the Rubber Reserve Company [10, 11 and 12] performed fire exposure tests on water-filled 5000-gallon horizontal tanks. These tanks were 5 ft in diameter and 33 ft 7 in. in length. Gasoline was used as the fuel. The results of Test No. 17 are summarized below:

- Thermal flux rate was 21,200 Btu/h/ft^2 to the wetted surface. Maximum hot gas temperature recorded was 2210°F: flame enveloped shielded thermocouples recorded a temperature of 1213°F.

- Vapor space metal temperature exceeded 1100°F, and the vapor space buckled in 28 minutes.

- Steel structures 45 ft from the fire and 28 ft in elevation collapsed from heat 6 minutes after ignition.

- Flame exposure of the tank was only 65 percent of total wetted area "because of the wind."

In November 1950, Dr. L.W.T. Cummings of the Sun Oil Company presented a paper at an API Subcommittee Meeting on Pressure Relieving Systems. Based on the interpretation of fire tests from various sources, this paper gave a method of evaluating the heat input as related to exposed area. The result

of this method can be expressed: Hourly heat input equals 34,500 $A^{0.82}$ Btu, where "A" is the tank area (wetted surface) in square feet. This is the expression contained in the current tank car specification flow capacity formulas. In subsequent paragraphs, we will be exploring this paper and associated documents in some detail.

At this point, let us define "wetted surface" and examine the validity of using total shell area for the tank car case. Wetted surface refers to the vessel surface physically in contact with liquid contents, and is the area over which effective heat transfer takes place between the vessel and the contents. Tank cars are not loaded full, but have specified outages to allow for liquid expansion due to possible temperature rise in transit. Typical outages range from two to over ten volume percent. However, the outage is based on ambient temperature rise and not on fire exposure. Full-car conditions can prevail at approximately 115°F. Therefore, because the car may indeed be full the use of total shell area is justified when calculating the maximum effective heat transfer area for tank cars.

In subsequent paragraphs discussing effective heat transfer, a $\frac{1}{A^{0.18}}$ factor and an $A^{0.82}$ factor appear frequently. A short explanation relating the two factors is in order. Dr. Cummings suggested that the fraction of total vessel area exposed to flame can be expressed as

$$E = \frac{1}{A^{0.18}}, \text{ where E equals fraction exposed}$$
$$\text{A equals wetted area.}$$

This expression may also be stated as $E = A^{-0.18}$. The effective heat transfer area is the product of total wetted area times the fraction of wetted area exposed, or Effective Area equals A x E. Substituting for E, the expression becomes

$$\text{Effective Area} = A^{1.0} \times A^{-0.18}, \text{ or } A^{0.82}.$$

Because of its pivotal importance to the current safety relief specification for tank cars, significant portions of the paper are reproduced

verbatim. The following quotes are from the second revision, dated 31 October 1951, of the original paper [13] as they appeared in Appendix III of the minutes, Subcommittee on Pressure Relieving Systems of the API.

"General

In addition to operational sources of overpressure, accidental fire exterior to a vessel will generate vapor resulting in overpressure unless relieved. The heat input to vessels containing volatile hydrocarbon liquids exposed to exterior fire is estimated by employing the appropriate values of the fuel, exposure, and environment factors in the formula provided. The actual volume of vapor to be relieved is then determined by the latent heat of the contained liquid at the pressure and temperature obtaining. This will establish the minimum relieving requirement.

The relieving capacity follows directly once the conditions are given. Sound engineering judgment and experience on the part of the designer, however, are required to attain a safe and economical solution to the fire protection problem. For example, adequate relief capacity could be installed to take care of a catastrophic fire, but this would be neither a safe nor in the long run an economical solution for vessels located within a processing area.

Relief valves do not protect a vessel against fire. They only protect against overpressure. An exterior fire may so weaken the metal at the operating pressure that shear failure may occur which in turn may result in cleavage failure extending randomly throughout the vessel to unheated areas. It should be the objective of the designer not only to provide adequate relief, but also to provide conditions which limit with certainty heat transmission to the vessel and particularly to its unwetted surface.

Scope

The method is applicable to all open free-burning fires outside a vessel, but does not include those cases where fuel under pressure is jetted as a torch against the vessel. Protection of a vessel against the high temperatures and heat generated by torch action requires interposition of a body between the torch and the vessel or some means of extinguishing the torch, such as, depressuring the fuel source.

Fire Classification

Fires surrounding vessels may be classified as catastrophic, uncontrolled, and controlled. The fuel source is the hydrocarbon contained in the vessel or adjacent vessels which has inadvertently flowed from the vessel and become ignited. The liquid fuel lies in a pool around the vessel and the fire is said to be open or free burning, as contrasted with the

mechanically controlled combustion in a furnace. The flames are luminous and consequently have a higher intensity of radiation than flames encountered in furnace design.

Catastrophic Fire

The catastrophic fire is one in which the vessel is practically completely surrounded by fire. An example would be a vessel located inside a building or enclosure where the absence of air currents permits the flame to surround the vessel to considerable depth.

Uncontrolled Fire

An uncontrolled fire is one in which the only favorable factor limiting the heat input to the vessel is the wind which tends to carry the flame off its target. All other factors are unfavorable. Nothing has been done in advance to reduce the fuel supply and no attempt is made to extinguish the fire.

Controlled Fire

Fires in which the interior environment is such as to withdraw the fuel away from the vessel and where prompt effective means to extinguish the flame are employed are considered controlled.

Unit Heat Input

The average unit heat input rate or heat flux to a vessel exposed to open fire is expressed by the following general formula:

$$\frac{Q}{HA} = Q_F E F_1 F_2 \text{ (Btu/hr/sq ft of wetted surface)} \qquad [4]$$

where $\frac{Q}{H}$ is the total heat input to the wetted surface of the vessel, expressed as Btu/hr, A is the total wetted surface in square feet. The symbols of the right hand member of the equation are respectively fuel, exposure and environmental factors to be defined below.

Fuel Factor

The fuel factor, Q_F, is defined as the actual unit heat flux to the outside of the wetted surface of a vessel completely exposed to the open flame expressed as Btu/hr for one square foot, when the receiver temperature is low so as to make reradiation insignificant with respect to the radiating power of the flame. The results of numerous investigations indicate the value of the fuel factor for liquid kerosine, gasoline, and butane to be 34,500 Btu/hr/sq ft. Comparison of small liquid propane and gasoline fires in the laboratory and large fires in the field indicate the same intensity of luminosity for both liquid fuels. Accordingly, the fuel factor, Q_F, for all hydrocarbon fuels burning from an open liquid pool is set at 34,500 Btu/hr/sq ft."

NOTE: The original paper suggested that lighter fuels, when vaporizing under release to atmospheric pressure, may have a lower radiating power, noting that gaseous propane was found to have a fuel factor of 24,600 Btu/h/ft^2 by one group of investigators.

Continuing with excerpts from the third revision.

"Exposure Factor

Controlled and Uncontrolled Fires

The exposure factor, E, is defined as the fraction of the wetted surface of the vessel exposed to open fire. The data from many tests indicate that as the size of a vessel increases the fraction of the wetted surface exposed decreases in accordance with the expression:

$$E = \frac{1}{A^{0.18}} \text{ (a ratio, no units)}$$

This relationship applies only to controlled and uncontrolled fires.

Exposure factors less than 1.0 decrease the average heat flux calculated by the general formula, Equation 4, but this is not interpreted to mean that the actual unit flux is less for larger vessels. The part of the wetted surface of the bare vessel enveloped with fire receives heat at the rate set by the fuel factor, and the result obtained by the formula is the total flux averaged over the wetted surface. The actual flux intensity to the metal surface may be lowered only by insulation on the vessel or by deluge equipment.

Catastrophic Fires

The exposure factor for catastrophic fires is 1.0.

Environment Factors

Environment factors apply to all classes of fires. They are divided into two groups, designated as the exterior environment factor, F_1, and the adjacent environment factor, F_2.

Exterior Environment Factor

(sic) Absence of drainage of the fuel away from the vessel and prompt effective fire fighting, the exterior environment factor, F_1, is 1.0. In the event both of these measures are employed, the exterior environment

(Quotation Continued)

factor has been shown in the examination of actual fires and by tests to become 0.6. Improved drainage methods and snuffing ditches in which the fuel is trapped out of contact with air have not been evaluated, but may be worthy of the designer's consideration. The exterior environment factor, F_1, is 1.0 for uncontrolled fires by definition. It is also 1.0 for catastrophic fires, because of possible inaccessibility of the vessel.

Adjacent Environment Factor

The adjacent environment factor, F_2, is concerned with the degree of limitation of the influx of heat to the wetted surface and consequently, the generation of vapor to be relieved. It has to do with the environment immediately adjacent to the vessel, such as insulation, and deluge equipment.

For a bare vessel, the value assigned to F_2 is 1.0. The effect of various methods of limiting vapor generation is reflected in the values of F_2, shown in Table I for typical conditions.

Section VI points out that the dry metal surfaces in contact with an open fire will rise rapidly in temperature so that the allowable metal stress may be exceeded at the designed operating pressure. The metal temperature below the liquid surface may also rise rapidly if film boiling is encountered as a result of high fluxes from torch flames or vapor blanketing of surfaces as the vapor generated rises along the metal surface. In addition, high metal temperatures will be encountered below the contained liquid surface when coke or other material is deposited as an insulating layer on the inner surface. These considerations indicate the desirability of providing, in advance, an environment which will limit the heat flux to the vessel.

Unit Heat Input Summary

The factors to be used in the average unit heat input formula, Equation 4, are summarized in Table II for various classes of fires.

Table II
FACTORS FOR HEAT INPUT FORMULA

FIRE CLASSIFICATION	FUEL FACTOR QF	EXPOSURE FACTOR E	ENVIRONMENT FACTORS	
			EXTERIOR F_1	ADJACENT F_2 (SEE TABLE I)
CATASTROPHIC	34,500	1.0	1.0	0.0 TO 1.0
UNCONTROLLED	34,500	$E = \frac{1}{A^{0.18}}$	1.0	0.0 TO 1.0
CONTROLLED	34,500	$E = \frac{1}{A^{0.18}}$	0.6	0.0 TO 1.0

(Quotation Continued)

The general formula, Equation 4, may also be expressed as follows for uncontrolled and controlled fires:

$$\frac{Q}{HA} = \frac{34{,}500\ F_1F_2}{A^{0.18}} \quad \text{(Btu/hr/sq ft wetted surface)}$$

Surface Area Exposed to Fire

Only the surface which is wetted is considered to be effective in generating vapor to be relieved and not the total surface. Little heat is transferred by metal conduction from the dry to the wetted surface. If the dry surface temperature becomes sufficiently high to contribute heat by radiation to the liquid, it may rupture at the operating pressure. This position is borne out by the data of several investigators.

The wetted surface of process surge tanks is to be taken as 50% of their normal capacity, and that of pressure storage is 100%.

Vapor Volume to be Relieved

The total heat input rate to the vessel resulting from exposure to open fire is the product of the wetted surface and the average unit heat flux determined by the conditions using Equation 4 or 5. The rate of vapor relief required is then calculated from the total heat flux and the latent heat of the contained fluid at the operating temperature and pressure. The relieving capacity so determined is considered to be the minimum requirement.

Methods Employed for Limiting Heat Flux (tentative)

Large tanks in storage areas are to be provided with appropriate spacing and grading within the dyke area to slope away from tank with or without snuffing ditches.

Pressure storage is to be protected by one of the following methods:

Insulating the entire vessel with protected insulation of thermal conductance not to exceed 1.0 Btu/hr/sq ft/F determined at 100F.

Burying the vessel underground.

Covering vessels at or above grade with earth with specified overburden.

Isolating individual bare units at appropriate spacing.

In alternates 1 through 3 the exterior of the vessel surface is to be adequately protected against corrosion.

All vessels in processing areas are to have protected insulation.

All classes of storage may use underground caverns, either man-made or natural.

As a matter of record, Propane and Butane tank cars ICC classification 105 have been in service approximately 20 years and have been subjected to approximately 50 fires of varying intensity without a single failure. ICC 105 specifies an approved insulation with a maximum thermal conductance of 0.075 Btu/hr/sq ft/F covered with a metal jacket."

(End of Quote)

3.3 ANALYSIS OF CUMMINGS' PAPER

We are in general agreement with the bulk of observations made in this work [13] and subsequent letters amplifying on the findings [14], although there are certain reservations which we will discuss. Particularly appropriate was the statement that the objective of the relief-system designer should be to not only provide adequate relief capacity, but to provide conditions which limit heat transmission to the vessel, and particularly to its unwetted surface. In the case of tank cars, this could mean providing a protective cover such as insulation, if the basic shell could not survive in "uncontrolled" fire. It was Cummings' opinion that protection against "catastrophic" fires--fires involving total envelopment and zero wind such as could occur with a vessel inside a building--was not economical.

In considering the category of "uncontrolled" fires, Dr. Cummings' suggested that the average thermal flux to large vessels would be less than that to small vessels, because a smaller fraction of the wetted surface would be exposed to open fire. He stated that test data indicated the exposed fraction could be expressed as $E = \frac{1}{A^{0.18}}$, where E = fraction of vessel area exposed and A = total wetted area. Before examining this expression in detail, let us consider the initial premise of dependency of heat flux on vessel size.

Given a fixed sized pool of a given burning fuel, it is reasonable to assume that the trend of area fraction exposed, hence effective heat flux, is smaller with an increase in vessel size. A vessel geometry factor could affect, but probably not reverse this trend. We would further agree that a peak level, or "saturation" flux condition exists (the author suggested 34,500 Btu/h/ft^2 for hydrocarbon fuels). However, we also submit that saturation is not a point function, and that a decrease in effective flux from the peak level is a function of vessel size relative to fire size and not to vessel size alone. In other words, effective unit heat flux to an object the size of a tennis ball or that of a 55-gallon drum engulfed in a

very large fire of a given fuel will be essentially at saturation, despite their orders of magnitude area difference. Also, at equal conditions of exposure, effective unit flux levels will tend to be independent of vessel area.

Table 2 and Figure 1 are taken from supporting documents to References 13 and 14 for the API Subcommittee. Note in Table 2 that heat flux to exposed wetted surface does not vary widely for the various tests. Figure 1 is a plot of unit heat flux versus percentage exposure for the tests summarized in Table No. 1, showing an apparently linear relationship. A linear relationship would be appropriate, if

- flux is insensitive to area of test vessel for a given exposure area, such as we have postulated above; or
- a particular percentage of exposure is associated with a specific vessel size as postulated in Reference 13.

Unfortunately, Figure 1 cannot really be used to support or disprove either postulation. Note that the ordinate (heat flux) is the expression $\frac{Q}{HA}$, where A is total wetted area. Further, note that the abscissa (percent exposure) may be defined as $\frac{A\ \text{exposed}}{A} \times 100$, where A again is total wetted area. By virtue of having the same variable in both axes, the correlation can look very good when in fact, it may not exist. We will discuss this subject further in subsequent paragraphs.

Returning to Table 2, let us compare Duggan, Gilmour and Fisher Test No. 1, and API Project Test No. 1. Note that the 242-ft^2 (wetted area) vessel was 100 percent exposed and the 6-ft^2 vessel was 46 percent exposed. This result is in apparent contradiction to the assumption that the greater the vessel area, the smaller the fraction of area involved. A reason, of course, is that the percent exposure was a function of the particular test (e.g., size of fire) and not the vessel size. Review of the other tests leads to a similar conclusion. It would appear inappropriate to use these data in an attempt to

Table 2
SUMMARY OF FIRE EXPOSURE TESTS: REFERENCE 13

AUTHORITY	UNDERWRITERS LABORATORY	DUGGAN GILMOUR AND FISHER TEST NO. 1	DUGGAN GILMOUR AND FISHER TEST NO. 4	RUBBER RESERVE TEST NO. 17	RUBBER RESERVE TEST NO. 17	RUBBER RESERVE TEST NO. 17	API PROJECT TEST NO. 1	API PROJECT TEST NO. 2	STANDARD OIL OF CALIFORNIA
REFERENCE NUMBER *	1	2	2	3	3	3	4	4	10
TYPE OF EXPOSURE	WATER FLOWING OVER PLATE	WATER FLOWING OVER TANK	WATER HEATED IN TANK	HEATING WATER IN TANK	GENERATING STEAM IN TANK	WATER FLOWING INSIDE 3/4" STANDARD PIPE	HEATING WATER IN TANK	HEATING WATER IN TANK	HEATING WATER IN TANK
FUEL	GASOLINE	PROPANE GAS	PROPANE GAS	GASOLINE	GASOLINE	GASOLINE	KEROSENE	KEROSENE	NAPHTHA
HEAT FLUX TO WETTED SURFACE ($Btu/h/ft^2$)	32,500	25,900	12,800	23,200	21,000	30,400	15,700	16,800	32,000
OBSERVED EXPOSURE % WETTED SURFACE	100	100	48	60 TO 70	60 TO 70	80 TO 90	46	46	100
TEMPERATURE RECEIVING SURFACE (°F)	76	136	120	300	-	-	300 (APPROX.)	320 (APPROX.)	70 TO 212
HEAT FLUX TO EXPOSED WETTED SURFACE ($Btu/h/ft^2$)	32,500	25,900	26,700	35,700	32,300	35,800	34,200	36,500	32,000
TOTAL SURFACE (ft^2)	24	242	330	568	568	9.0	16.2	16.2	206
WETTED SURFACE (ft^2)	24	242	176	400	400	9.0	6.13	6.13	105
NOMINAL VESSEL CAPACITY (gallon)	-	3024	3024	5000	5000	-	37	37	1386

*REFERS TO CITED REFERENCES IN CUMMINGS PAPER

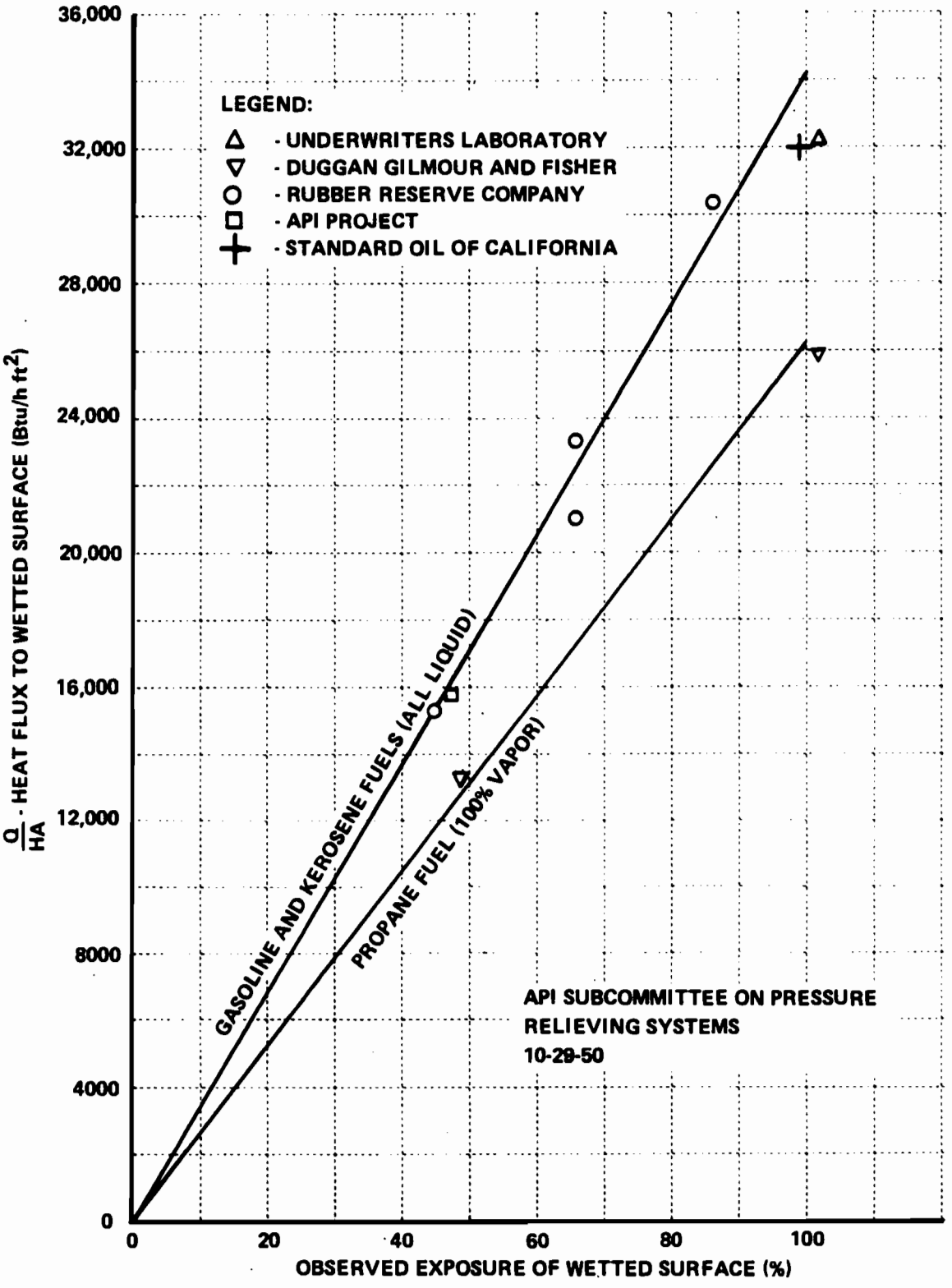

Figure 1 HEAT FLUX TO WATER-WETTED SURFACES EXPOSED TO OPEN FIRE

show a general relation between fractional area exposed and total wetted area--even if the underlying premise were sound. Figure 2 is a plot of percent surface exposed to fire versus wetted surface, including the above data points, as it appeared in an attachment to Reference 14, with the exception that the curve was extrapolated to 10,000 ft^2 in the original. The equation of this curve is the $E = \frac{1}{A^{0.18}}$ exposure factor used in the Appendix "A" tank car specifications. As in Figure 1, this plot contains the same variable (wetted surface) in both axes. Quoting from Mickley, Sherwood and Reed [15, paragraph 1-4], "Misleading Methods of Correlation."

> "Any correlation of experimental data based on a graph in which the same variable appears in both ordinate and abscissa should be viewed with suspicion. When one of the less important variables is placed in both quantities plotted, it is possible to extend the scale and make the correlation appear to be much better than it really is. Such correlations are occasionally presented in the literature. The investigator, trying various methods of plotting his results, hits upon a method of plotting that brings his data together and presents a correlation that is unintentionally deceiving as to its generality. Such methods of plotting may be arrived at by fairly sound analysis of the physical problem involved and may be defended as being rational, although a poor test of the data."

We would further add that although the use of a log-log plot is a legitimate technique for linearizing data, the visual effect in this case tended to mask the heavy emphasis on small vessel areas and the extent of extrapolation. To bring this point into perspective, we have plotted (Figure 3) the $34{,}500\ A^{0.82}$ function on a linear scale with area, and pinpointed the thermal flux data from Table 2. Remember that this plot is still tainted by the area variable in both axes, so no excessive concern over the scatter is warranted. However, the extent of extrapolation to values of interest with respect to tank cars is presented fairly. This fact, coupled with the extreme deviation to the "high" side of reported test values of thermal flux versus the supposedly conservative values on the side of safety predicted by the $34{,}500\ A^{0.82}$ expression is deeply disturbing.

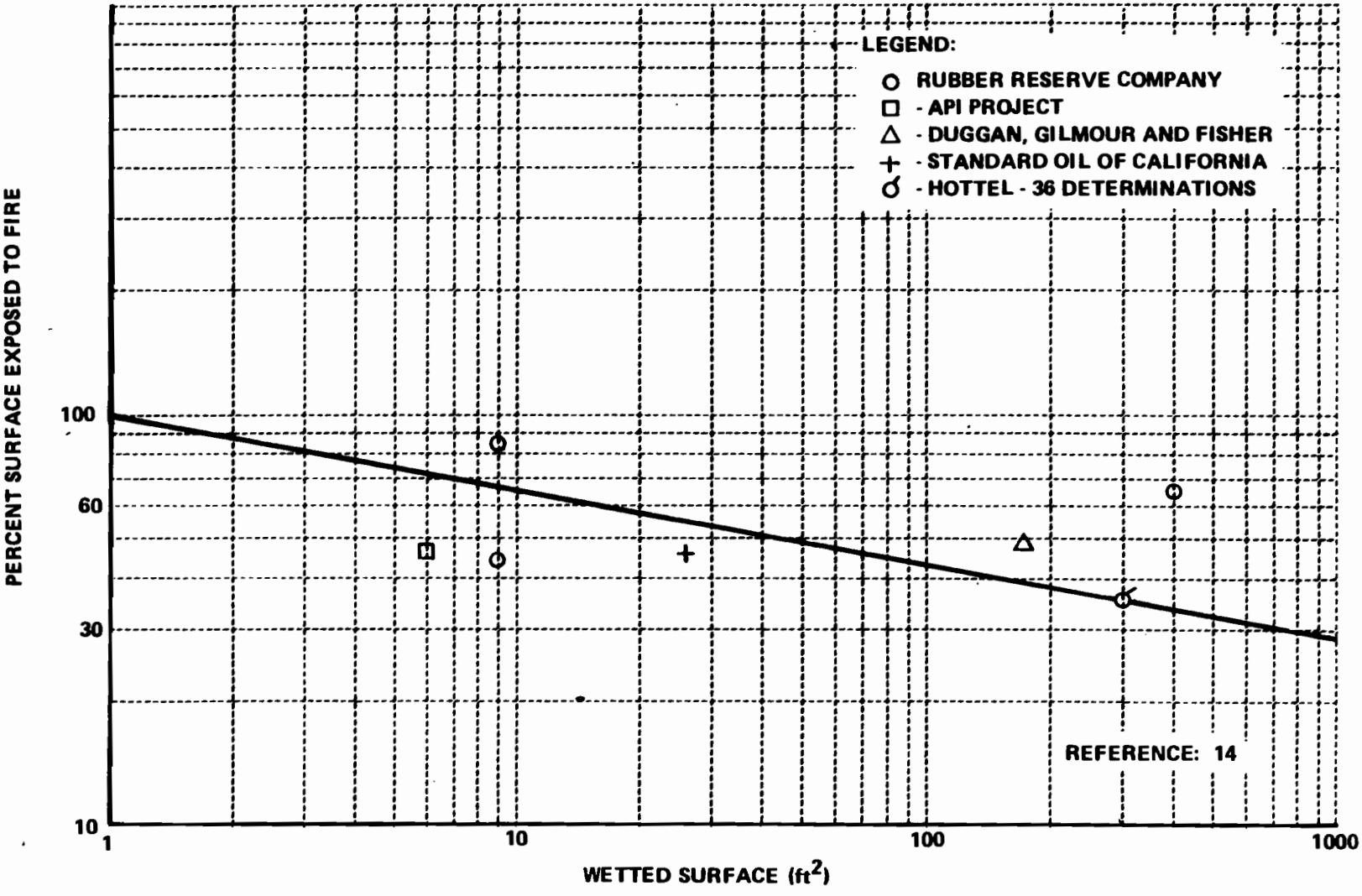

Figure 2 EXPOSURE OF WETTED SURFACE TO OPEN FIRE (UNCONTROLLED FIRES)

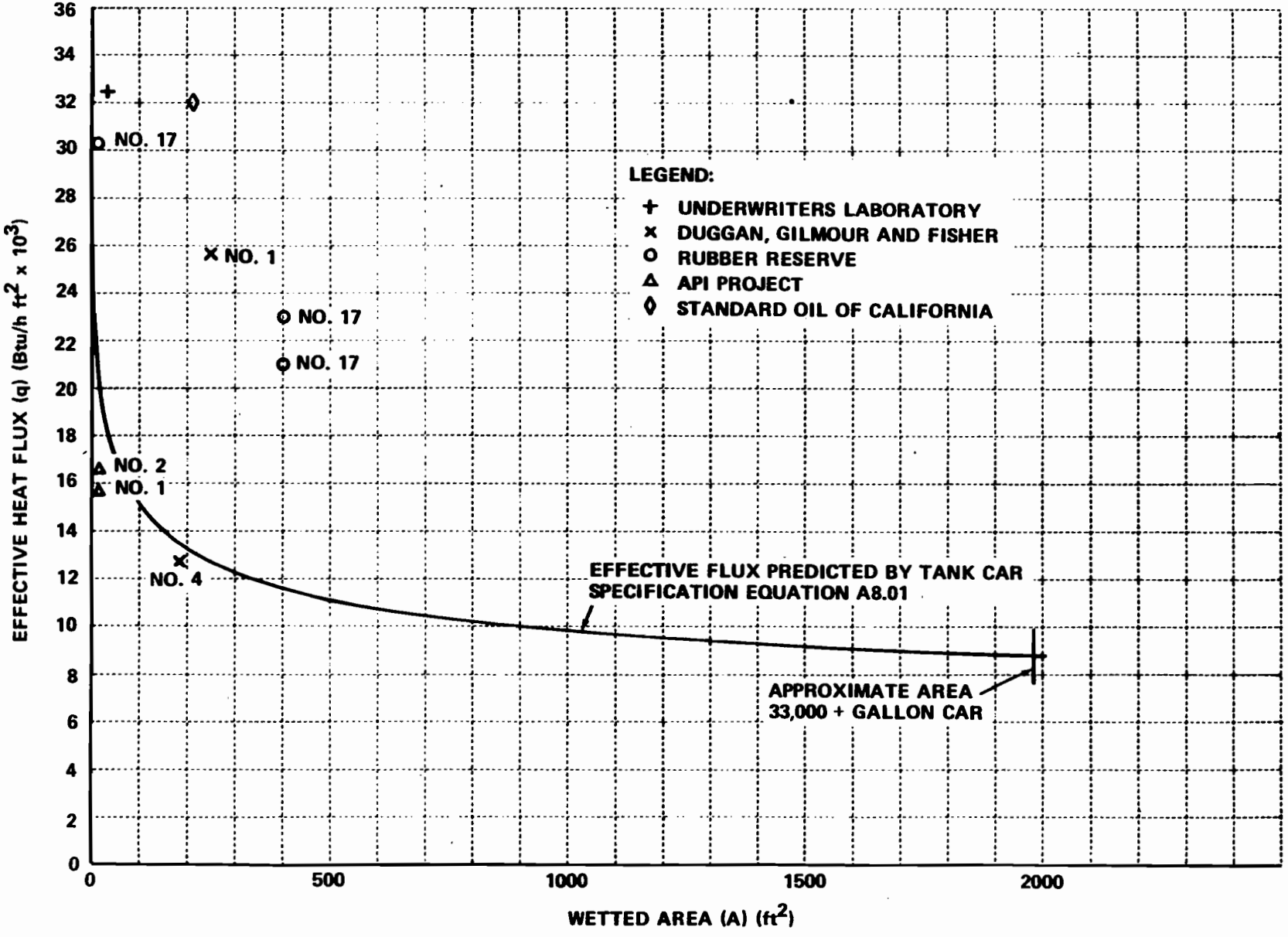

Figure 3 COMPARISON OF FIRE EXPOSURE TEST DATA REPORTED IN REFERENCE 13 VERSUS PREDICTED VALUE USING $A^{0.82}$ EXPOSURE FACTOR

Our search for quantified fire exposure data uncovered a single test involving a large vessel. As a part of a 1954 evaluation [16] by Carbide & Carbon Chemicals (Union Carbide) of the protective characteristics of a proprietary foam formulation, a "control" test was performed on an unprotected 55,000-gallon tank containing water. The 20-ft-diameter tank was filled to a depth of 23 ft 6 3/4 in., giving a wetted area of 1480 ft^2. Kerosene fuel was supplied to a 50-ft^2 diked area around the tank. The test duration was 19.5 min, with a fuel combustion rate of 8500 gal/h. Maximum wind was 3 mi/h and the average wind 0.2 mi/h. No estimate of percent area exposure by the flames was made due to lack of a standard technique, and widely divergent opinions of authorities on evaluation. The thermal flux rate was 28,800 Btu/h/ft^2, a value "higher than expected" by Carbide personnel who had been using a 20,000 Btu/h/ft^2 flux as a standard for a safety design. This test was performed under the general supervision of J.J. Duggan, who was part of the team that had proposed the 20,000 Btu/h/ft^2 guideline in 1943 [9] and as we noted in our earlier discussion, had been criticized in some quarters for not reducing his guideline value for larger vessels.

Figure 4 presents the same information as Figure 3 with the Carbide test added, along with data points from other tests of earlier years. Clearly, it would be desirable to have further data with larger area tanks. Nevertheless, little confidence can be placed in a safety formula that underestimates the heat flux to a tank exposed to an open fire by a factor of three. Eliminating any bias caused by having the same variable in both axes, we have plotted area versus total heat input in Figure 5. The "uncontrolled" and "catastrophic" fire total input levels as defined in Reference 13 are shown, as well as a 25,000 Btu/h/ft^2 constant flux reference (dashed line).

From the preceding, it can be seen that existing data from actual open-fire exposure tests, uncorrected to higher values to account for percentage area exposed due to test setup, or meteorological conditions, have frequently exceeded the predicted value for "uncontrolled" fires by a factor of at least two. Hydrocarbons of equivalent fuel values to those used in

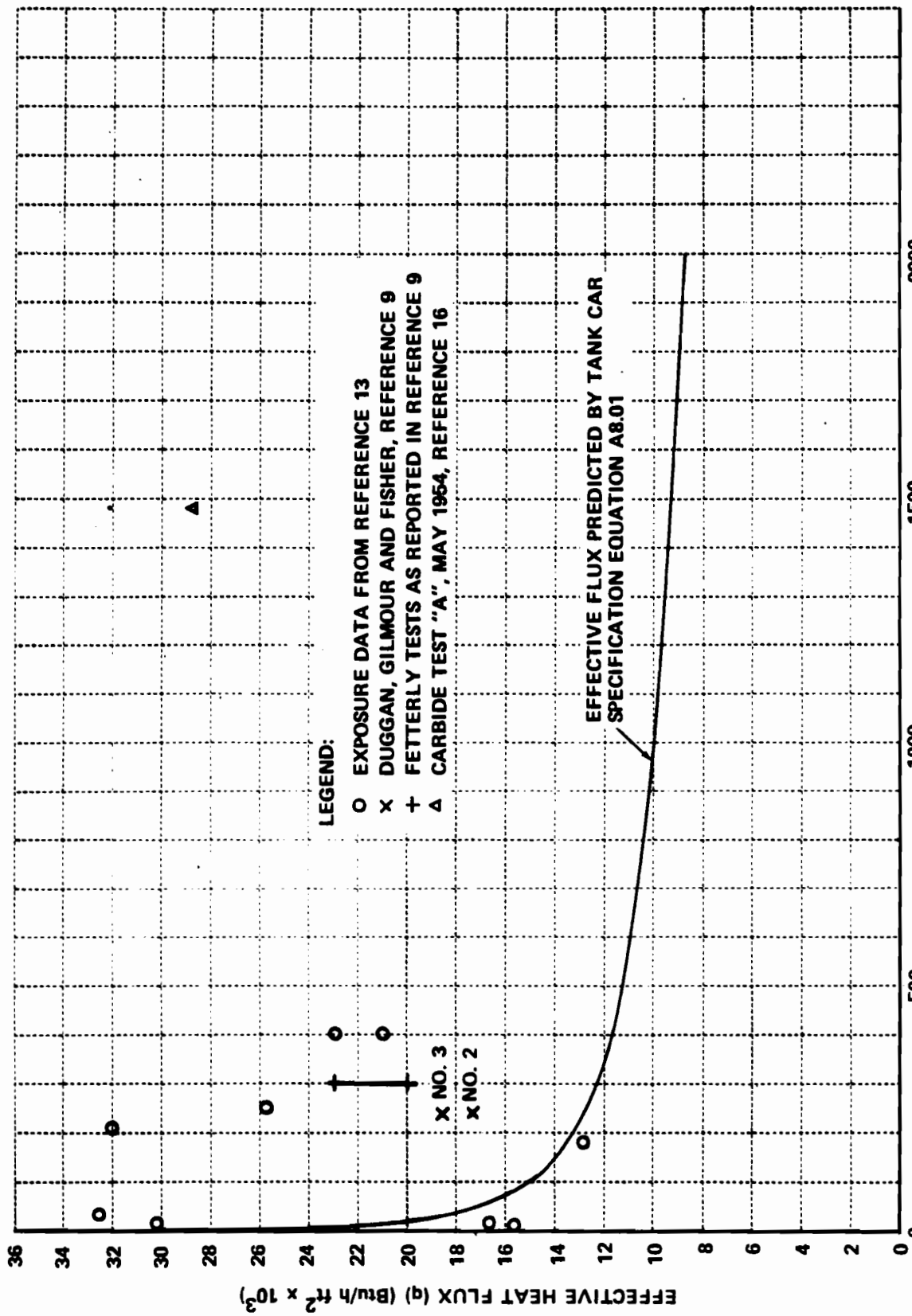

Figure 4 FIRE EXPOSURE TEST DATA VERSUS PREDICTED VALUES USING $A^{0.82}$ EXPOSURE FACTOR

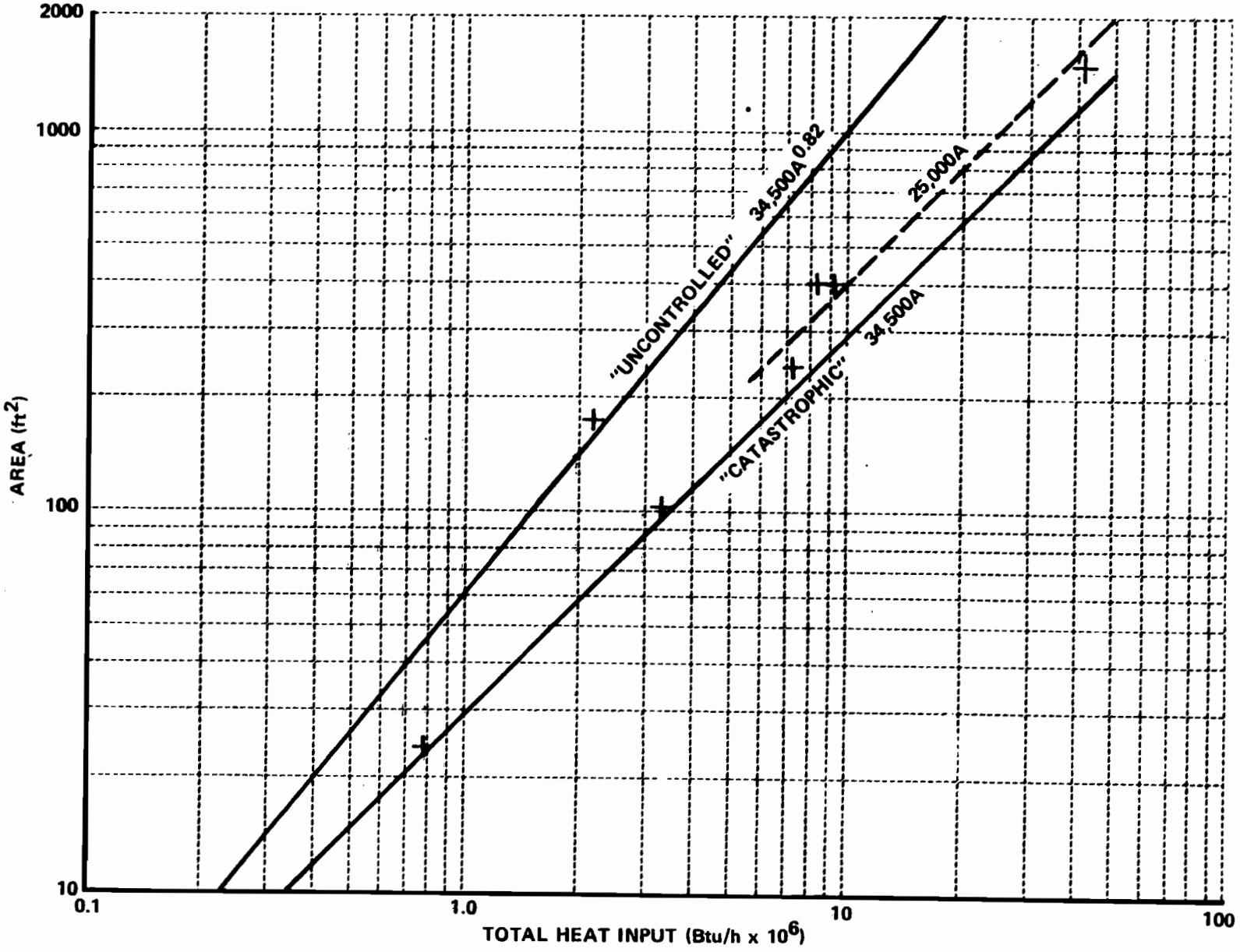

Figure 5 PLOT OF HEAT INPUT DATA FROM ACTUAL FIRE EXPOSURE TESTS VERSUS PREDICTED VALUES FOR "UNCONTROLLED" AND "CATASTROPHIC" FIRES AS DEFINED IN REFERENCE 13

the tests described are commodities handled by rail. Fuel rates to the fires producing the flux levels described in the tests were within potential rates from a ruptured tank car. They were also within the rates for the regenerative heating situation where tank car contents from an upset car are venting through a normally operating safety-relief valve.

Summarizing our findings with regard to the 34,500 $A^{0.82}$ flux level now a part of the tank car specifications for safety-relief flow requirements:

(1) The 34,500 Btu/h/ft^2 local unit flux level may be reasonable, though it is not conservative.*

(2) The effective flux level determined by using the $A^{0.82}$ exposure factor is significantly low for unprotected tank car shells--probably by a factor of at least two.

(3) We feel the exposure factor, hence flux rate, to be erroneous for the following reasons:

(a) There is no theoretical basis for assuming a larger vessel to have a lesser area exposed to fire without consideration of geometry or potential fire size.

(b) Exposure data used for establishing the relationship was a function of the particular test arrangements and conditions--not vessel size alone.

(c) A misleading correlation technique was used.

(d) Extensive extrapolation was used from a curve fitted to scattered data.

(e) It is unsupported by actual test results.

*Local flux rates to 90,000 Btu/h/ft^2 have been measured in 18-foot-diameter free-burning liquid hydrocarbon fires at CAL, [16A].

Finally, it is frequently mentioned as a point of support that the exposure factor has been adopted by the NFPA, API, Compressed Gas Association, Coast Guard, and others. This is true; however, the evidence suggests that adoption was derived from the same source, and was not the result of independent investigation.

3.4 ANALYSIS OF SPECIFIC FLOW-CAPACITY FORMULAS FROM APPENDIX A, AAR SPECIFICATIONS FOR TANK CARS

Section A8.01 of the Appendix contains the formula for compressed gas in uninsulated tanks which would be used for calculating relief capacity for a 112A340W tank for LPG/NH_3 service.

The formula is

$$Q_a = \frac{633,000}{LC}\sqrt{\frac{ZT}{M}}\,A^{0.82} \qquad (1)$$

where

A = Total outside surface area of tank in square feet

C = Gas constant which is a function of the ratio of specific heats (k). $C = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}$

L = Latent heat of gas at flowing conditions (Btu/lb)

Q_a = Required flow capacity in ft^3/min of air at standard conditions defined as 14.7 lb/in^2 absolute and 60°F (520°R)

M = Molecular weight of gas

T = Temperature in degrees Rankine (°R) of gas at flowing conditions

Z = Compressibility factor at flowing conditions

Examining the theoretical basis for this formula, the equation for weight flow of vapor through an orifice given sonic flow conditions--predominant conditions during the period of interest--is

$$W_g = CKaP\sqrt{\frac{M}{ZT}} \qquad (2)$$

where

W_g = Weight flow in lb/h of gas

K = Coefficient of discharge (dimensionless)

a = Discharge area in in^2

P = Upstream pressure in lb/in^2 absolute

Other variables as previously defined.

Vaporization rate of a liquid in response to a thermal load may be described by Equation (3).

$$W_g = \frac{q A'}{L} \tag{3}$$

where

q = Unit heat flux in Btu/h ft^2

A' = Effective heat transfer area in ft^2

Equating (2) and (3) gives the relationship for orifice size at P&T to relieve at the vapor generation rate. Following through conversion steps, one arrives at the equivalent flow capacity of air (Q_a) at S.T.P.

Conditions: constant flow area , weight flow of any gas at P&T

Convert to weight flow of air at P and standard T

$$\frac{W_{air}}{W_g} = \frac{C_{air} K a P \sqrt{\frac{M_{air}}{Z_{air} T_{STD}}}}{C \quad K a P \sqrt{\frac{M}{ZT}}}$$

$$M_{air} = 28.97 \quad C_{air} = 356 (k = 1.4) \quad Z_{air} \cong 1$$

$$W_{a.r} = W_g \times \frac{356}{C} \times \frac{\sqrt{28.97}}{\sqrt{M}} \times \frac{\sqrt{Z_g}}{1} \times \frac{\sqrt{T}}{\sqrt{520°R}}$$

$$W_{a.r} = 84 \frac{W_g}{C} \sqrt{\frac{ZT}{M}} \qquad (4)$$

convert to volumetric flow of air

$$Q_a \left(\frac{ft^3}{min}\right) = W_{air} \left(\frac{lbs}{hr}\right) \times \frac{1}{28.97} \left(\frac{lb\ mole}{lb}\right) \times 379.4 \frac{ft^3\ air\ @\ 14.7\ psia}{lb\ mole} \times \frac{1}{60} \left(\frac{hr}{min}\right)$$

& 60°F

or

$$Q_a = 0.2183\ W_{air}$$

substituting in (4)

$$Q_a = 18.34 \frac{W_g}{C} \sqrt{\frac{ZT}{M}} \qquad (5)$$

Substituting equation (3) in (5) gives

$$Q_a = \frac{18.34 q}{LC} \sqrt{\frac{ZT}{M}}\ A' \qquad (6)$$

general capacity formula for any thermal flux (vapor)

Comparing again with Equation (1), we may determine the empirical value utilized in the sizing formula, namely the value of the total thermal flux (H)--the product of qA',

$$q = \frac{633{,}000}{18.34} = 34{,}500\ \frac{Btu}{hr\ ft^2}$$

$$A' = A^{0.82}\ ft^2$$

and

$$H = 34{,}500\, A^{0.82} \quad \text{Btu/hr}$$

There can be a tendency to become overly involved in the validity of the 34,500-Btu/h/ft^2 coefficient without realizing the full implication of the area fractional exponent. Convenient charts of A versus $A^{0.82}$ can bury the significance in routine calculation. Let us present the same information in a different manner by calculating the effective unit flux for given area of tank.

$$q'_{\substack{\text{effective}\\ \text{unit flux}}} = \frac{H}{A} \tag{7}$$

Table 3
VARIATION OF FLUX WITH AREA

LET	H	q
A = 1 ft^2	34,500 Btu/h	34,500 Btu/h ft^2
A = 1000 ft^2	9.8325 x 10^6 Btu/h	9,833 Btu/h ft^2
A = 1800 ft^2	16.1115 x 10^6 Btu/h	8,951 Btu/h ft^2

Referring to Table 3, we see that the formula is really predicting fluxes of only 8000 to 10,000 Btu-/h ft^2 in our area of interest--tank cars. The question then becomes: Can fluxes in the 8000- to 10,000-Btu/h ft^2 region, not 34,500-Btu/h ft^2, be supported by test data or theory? The evidence presented in the preceding paragraphs indicated that this flux level is significantly low.

From our review of fire tests, the expression 25,000 A for total heat flux would more closely approximate expected flux values than 34,500 $A^{0.82}$. Note that we are not recommending adoption of this value as a specification, but are using this value, which has been realized in large-scale tests, to establish some order of magnitude comparisons between expected peak flows and the existing capacity requirements. Using 25,000 A as the expected flux for compressed gases in uninsulated cars, formula A8.01 becomes

$$Q_a = \frac{458,000}{LC}\sqrt{\frac{ZT}{M}}\ A \qquad (8)$$

rather than

$$Q_a = \frac{633,000}{LC}\sqrt{\frac{ZT}{M}}\ A^{0.82} \qquad (9)$$

$A^{0.82}$ represents roughly 25% of the total area for a typical tank car shell size (Figure 6). On this basis, peak flow by Equation (8) would be approximately 2.9 times the rate computed from the existing formula (Equation (9)). Since the additional formulas in Appendix "A" of the AAR Specifications are derived from the basic A8.01 formula, a review of their adequacy in this report from the vapor-flow standpoint is in order.

Later in the report, we will treat the overturned car case where liquid flow is also possible. It will be shown that the two-phase flashing flow condition for a liquefied compressed gas such as propane can be the controlling case with regard to valve sizing--a factor not considered in the present specifications.

Equation A8.02(a) for compressed gases in insulated tanks is shown below.

$$Q_a = \frac{146.8\,U(1200-t)}{LC}\sqrt{\frac{ZT}{M}}\ A^{0.82} \qquad (10)$$

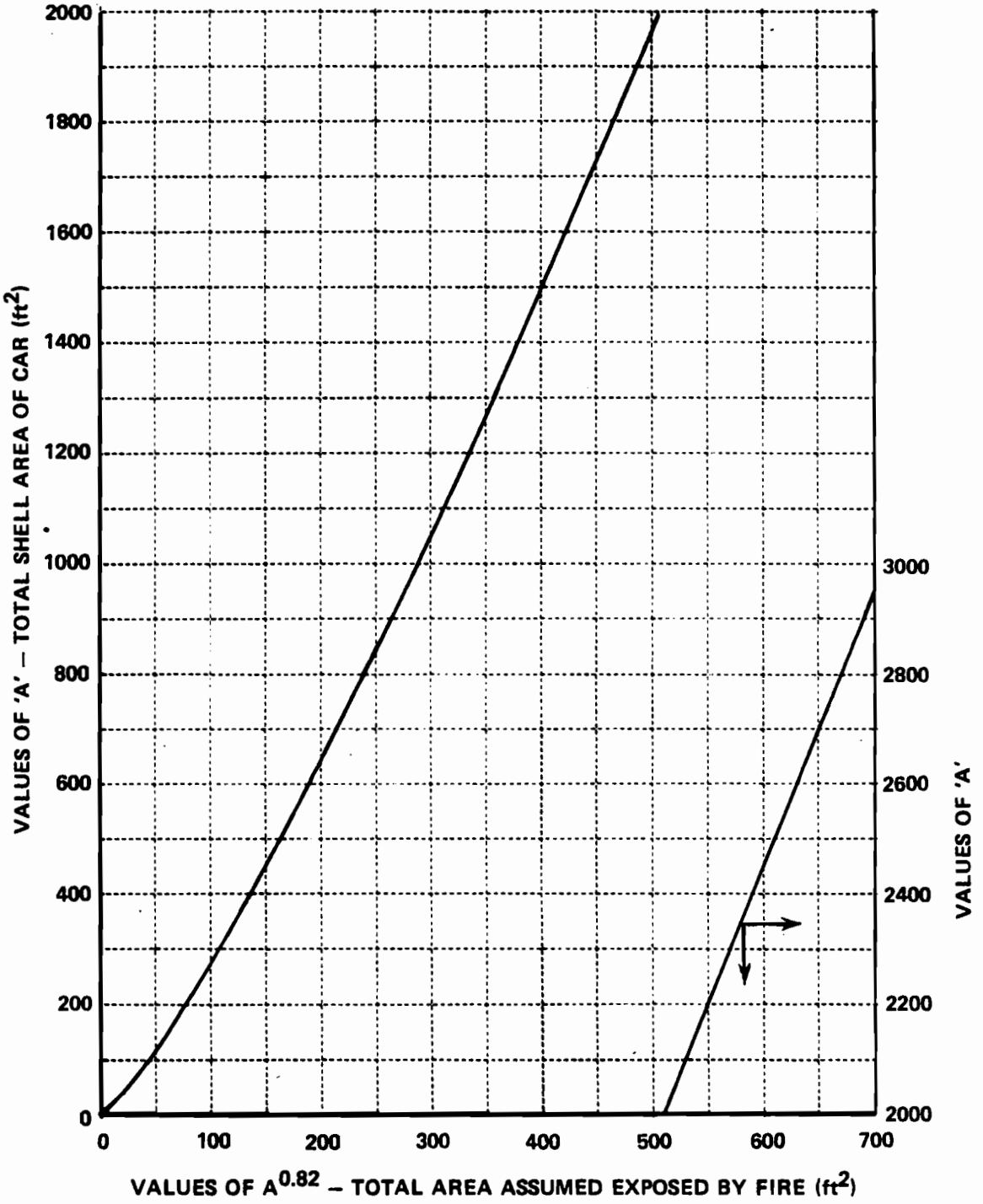

Figure 6 VALUES OF $A^{0.82}$ RELATIVE TO 'A'

where

U = thermal conductance of insulating material at 100°F where conductance equals thermal conductivity divided by thickness of insulation in inches (Btu/h/ft^2/°F/in)

t = temperature in °F of gas at flowing conditions.

The following derivation of the above equation is excerpted from the AAR Specifications.

"A9.02 DERIVATION OF FORMULA A8.02

The heat input into a bare tank has been measured at 34,500 Btu/h/ft^2. For an insulated vessel, the heat input may be expressed as U(1200-t), where:

1200 = Assumed ambient temperature, °F
Thermodynamically, U should be at the mean temperature or at $(\frac{1200-t}{2})$*

Thermal conductivity data are difficult to find at elevated temperatures, but are readily available at 100F. U is therefore defined as conductivity at 100F and to compensate for this temperature, it is multiplied by two. Then, assuming that the insulation is rendered 50 percent ineffective in a fire, the result is again multiplied by two. The heat transfer through the metal connections and fittings, projecting through the insulation is approximately equal to the transfer through the insulated area. Therefore, another factor of two is used. The product of the three factors for U is thus equal to eight. For an insulated container, formula A8.01 is multiplied by a factor F where:

$$F = \frac{8U(1200-t)}{34,500}$$

Multiplying and rearranging:

$$Q_a = \frac{146.8U(1200-t)}{LC}\sqrt{\frac{ZT}{M}}\,A^{0.82}$$

(Formula A8.02(a))."

(End of Quote)

*Typographical error as it appears in the original; mean temperature should be $(\frac{1200+t}{2})$.

Whenever possible, it would seem advisable to eliminate "rules-of-thumb" from safety specifications. Where rigorous analysis cannot be supplied, values should be indisputably conservative on the side of safety. Significant changes in materials of construction or fabrication techniques should prompt a review of existing specifications to check their validity.

Examining the above derivation, we would agree that thermal conductivity data at elevated temperatures are not universally available, but a great deal of information on the commonly used insulations is published, and moreover, determination of conductivity by test is possible. It is fruitful to examine the "times two" rule-of-thumb compensation for increase in conductivity at high temperature. A now obsolete construction practice for tank cars used rock-wool insulation.

Thermal conductivity data for rock wool [17] follows.

t MEAN TEMPERATURE (°F)	k THERMAL CONDUCTIVITY (Btu/h/ft^2) (°F/ft)
100	0.03
500	0.05

More recently, bonded glass-wool has frequently been used for insulating purposes. Nominal fiber diameter and bulk density have the most significant effect upon insulating properties. Typically, those glass-wools having the best conductivity values at normal atmospheric temperatures have the sharpest rate of rise in conductivity with temperature.

Thermal conductivity data for a 2-lb/ft^3 fine fiber wool [18] follows.

MEAN TEMPERATURE (°F)	THERMAL CONDUCTIVITY Btu/h (ft^2) (°F/ft)
100	0.02
500	0.07

It may be concluded that while the "times two" factor perhaps was at one time adequate, it can lead to serious underestimates with the adoption of insulating materials with larger $\frac{\Delta k}{\Delta t}$ factors, such as the example glass-wool. The assumption that 50% of insulation is rendered ineffective in fire without concomitant high-temperature performance standards for insulation and explanation of the terms "50% ineffective"* can be a potential trouble spot.

For example, polyurethane foam and fiberglass would be assumed equally fire resistant by the specification, where in reality the glass would be providing some protection at temperatures where the urethane foam would no longer exist as an effective insulation. The net result of the above two factors is that cars could be built to the same specification with a vast difference existing in the true degree of protection provided.

Even if the insulation is totally ineffective, there is some reduction in heat input due to the presence of the steel insulation jacket acting as a radiation barrier. This reduction can be shown to be small in comparison to the reduction provided by an effective insulation. Reradiation from the fire-exposed jacket increases the effective value of the thermal conductance U as follows. Using the wall temperature as given in the Specification, the conductance due to radiation is defined as

$$U_R = \frac{q_R}{(1200 - t)} \tag{11}$$

where q_R is the radiation heat flux between the outer jacket at 1200°F and the shell at the flowing gas temperature. The radiation from the outer jacket to the inner tank car shell is given by

$$q_R = \frac{\mathcal{E}_1 \mathcal{E}_2 (1200 + 460)^4 - (t + 460)^4}{\mathcal{E}_1 + \mathcal{E}_2 - \mathcal{E}_1 \mathcal{E}_2} \tag{12}$$

where $\mathcal{E}_2$ is the emissivity of the inside of the outer shell and $\mathcal{E}_1$ the emissivity of the outside of the inner shell.

*It makes considerable difference whether 50% of insulation is removed from 100% of the area or whether 100% is removed from 50% of the area.

The lowest emissivities result in the lowest value of q_{fR} and therefore of U_R. An emissivity of 0.6 would be low for steel, especially in the presence of decomposing insulation. However, even with this low value of emissivity, U_R would be 5.5 Btu/h ft^2 °F, whereas the conductance with the insulation in place is restricted to a maximum of two times 0.075 Btu/h ft^2 °F in the tank car specification. That is, the conductance, if the insulation is rendered ineffective, is more than 35 times as great as if the insulation remains effective. A "times two" factor would not begin to account for this type of insulation failure.

Although the A8.01 and A8.02 titles in Appendix A refer to compressed gases in general, the equations are useful for liquefied compressed gases only. The definitions of liquefied and nonliquefied gases in subpart "F" of T.C. George's Tariff No. 23 referring to the presence or absence of a liquid state under charged pressure at 70°F need to be redefined for purposes of application of the A8.01 and A8.02 flow rating formulas. Formulas A8.01 and A8.02 are applicable for vapor flow for liquefied compressed gases, redefined as compressed gases for which a saturated liquid phase exists at the flow rating pressure of the safety-relief system. Depending on the values and trends of the thermodynamic properties of the liquefied gas near the relief-system setting, the vapor flow case may not be controlling -- a subject we will discuss later. If the lading is entirely gaseous at the flow rating pressure, an entirely different criterion should apply for establishing relieving capacity. Totally gas-phase ladings represent a very small portion of shipping volume and are commonly shipped in multiple unit or 107 series cars -- outside of the purview of this study.

Formula A8.03 for liquids other than compressed gases in uninsulated tanks is given as

$$Q_a = 45A^{0.82} \qquad \text{[A8.03]}$$

The derivation of this formula as described in Appendix A follows:

"DERIVATION OF FORMULA A8.03

In Equations A8.03 and A8.04, the properties of 26 pound gasoline have been used as it was found to be the product with the greatest expansion factor shipped in tank car specifications DOT-103, DOT-103W, DOT-104, DOT-104W, DOT-103ALW, DOT-103EW, DOT-111A100W1 or 3, and AAR-203W. These cars may be used for a large group of products and, for this reason, the 26 pound gasoline was selected so that the equation could be presented in simplified form. Properties used in equations are as follows:

Flow Rating Pressure (psia)	P	100	60
Corresponding Temperature (°F)	t	190	150
Specific Heat Ratio	k	1.088	1.09
Compressibility Factor	Z	0.864	0.90
Molecular Weight-Vapor	M	63	63
Latent Heat (Btu/lb)	L	130	144
Gas Constant	C	325	327

$$Q_a = \frac{633{,}000\,A^{0.82}}{LC}\sqrt{\frac{ZI}{M}}$$

For 26 pound gasoline at 60 psia:

$$Q_a = \frac{633\,000\,A^{0.82}}{144 \times 327}\sqrt{\frac{(0.9)(610)}{63}} = 40A^{0.82}$$

For 26 pound gasoline at 100 psia:

$$Q_a = \frac{633{,}000A^{0.82}}{130 \times 325}\sqrt{\frac{(0.864)(650)}{63}} = 44.7A^{0.82}$$

$$Q_a = 45A^{0.82}$$ (Formula A8.03)."

(End of Quote)

A number of products do not require the relieving capacity of "26-pound gasoline," thereby partially offsetting the effects of underestimation

of thermal flux. Also, as we have stated in previous reports, time to reach peak vapor generation rates may be significantly longer for many materials which are in a liquid state at atmospheric conditions. Given these factors, coupled with the more favorable accident experience to date with ladings handled in nonpressure* equipment, priority remedial measures should be directed toward pressure service cars.

Nevertheless, potential problems with liquid ladings exist given the current flow capacity specifications. Acetone, for example, is a solvent with a DOT flammable liquid classification produced and shipped in tonnage quantities. Calculating safety-relief flow capacity requirements when shipped in a 111A100W1 car without insulation:

Flow rating pressure (psia)	P	100
Corresponding Temperature (°Rankine)	T	718 (250°F)
Gas Constant	c	328
Compressibility Factor	Z	0.9
Molecular Weight	M	58
Latent Heat (Btu/lb)	L	200

$$Q_a = 0.938 \times 10^{-3} q A' \quad \text{(acetone)} \qquad (13)$$

substituting $qA' = 34{,}500\ A^{0.82}$

$$Q_a \cong 32 A^{0.82} \qquad (14)$$

Therefore, if $34{,}500\ A^{0.82}$ accurately reflected the thermal flux level, Equation A8.03 would provide satisfactory vapor relief. Applying a more realistic 25,000 A flux rate, the equation for acetone becomes

*"Nonpressure" service as defined in T.C. George's Tariff No. 23.

$$Q_a \cong 23.45A \quad \text{(acetone)} \qquad (15)$$

Given a typical tank car shell area, the peak flow would be approximately twice that estimated by the existing A8.03 formula.

Formula A8.04 for liquids other than compressed gases in insulated tanks is derived in Appendix A as follows:

"A9.05 DERIVATION OF FORMULA A8.04

$$Q_a = \frac{(146.8)(1200-t)}{LC}\sqrt{\frac{ZT}{M}}\,UA^{0.82}$$

For 26 pound gasoline at 60 psia:

$$Q_a = \frac{(146.8)(1200-150)}{144 \times 327}\sqrt{\frac{(0.09)^{*}(610)}{63}}\,UA^{0.82}$$

For 26 pound gasoline at 100 psia:

$$Q_a = \frac{(146.8)(1200-190)}{(130)(325)}\sqrt{\frac{(0.664)(650)}{63}}\,UA^{0.82}$$

$$Q_a = 10.5\,UA^{0.82} \quad \text{(Formula A8.04)."}$$

(End of Quote)

Our earlier comments on the insulation factors also apply with regard to this formula. Given the lack of high-temperature performance specifications for the insulation to assure the validity of the assumptions in the relief formula, no generalized estimate can be made of the adequacy of relief capacity computed by this formula.

*Typographical error in original; should be 0.9.

3.5 EFFECTS OF UNDERESTIMATION OF THERMAL LOAD

It would be an error to make the immediate presumption that underestimation of the thermal load leads to disaster. It certainly has that potential, but further examination of the effects for various situations is required. First, the existing capacity formulas are based on peak vapor generation rates. These rates are not immediately realized due to the thermal capacitance of the tank car-cargo system. The time lag in vapor generation rate, and attendant pressure rise if unrelieved, are strongly dependent on the physical characteristics of the lading.

Consider a material which has an atmospheric boiling point above the ambient temperature range. The immediate effect of the heat load is to raise the temperature of the car and its contents. Initially, the shell temperature will rise with some thermal energy being transferred to the contents by convection in the liquid. Because this convection will at first be insufficient to extract the entire external heat load, the shell wall temperature will increase. As the shell wall temperature approaches the vaporization temperature of the liquid at the existing internal pressure, some surface or nucleate boiling will begin. At this time, the convection coefficient in the liquid at the inner wall will increase greatly, and the temperature gradient in the liquid will begin to reduce until only a small differential between the inner shell wall temperature and the bulk fluid exists.

During the time period from initial application of heat until the liquid vapor pressure is in equilibrium with the internal pressure, the external heat load has generated insignificant vapor. Some pressure increase in the vapor space will occur due to liquid expansion if non-condensable gases are present. Once saturation conditions are reached, further application of heat will result in the generation of vapor. The vapor generated in the closed tank will tend to increase the pressure. Heat from partial condensation of existing vapor and continuing application of the external heat load is added as

sensible heat to the liquid, driving it toward saturation conditions at the new temperature and pressure. This process continues until the relief valve opens. (Note that if the tank becomes liquid full below the relief setting, a compressed liquid (subcooled) condition slightly below saturation is present before valve opening.)

During the time period from initial saturation conditions to relief valve opening, vaporization does occur, controlled by the external heat load, sensible heat requirements to increase the liquid temperature to the equilibrium boiling point with rise in internal pressure, and the heat of vaporization. As the temperature rises during this period, the latent heat of vaporization reduces; therefore, more vapor is generated per unit heat input. The rate of reduction of latent heat of vaporization with temperature becomes more pronounced as the temperature approaches the critical point* of the material.

For convenience, we have separated the concurrent vaporization-condensation functions in our discussion. It must be remembered that for our closed vessel system, we physically have what might be described as "incipient" boiling. Given the added effects of liquid expansion with temperature rise, a net reduction in weight of vapor present in the system may occur with temperature rise. This does not detract from the fact that we are operating at saturated conditions and the pressure in the shell will increase exponentially with temperature rise, a prime mechanism for pressure generation being the vaporization "piece" of the overall equilibrium phase change with increase in system enthalpy.

Once the relief valve is opened, and presuming it has sufficient capacity to maintain a constant pressure in the shell, the vaporization rate is controlled only by the latent heat of vaporization and the applied external

*Critical temperature is that temperature above which the compound cannot exist in a liquid state; therefore, heat of vaporization equals zero.

heat load. It is this peak vaporization rate, occurring some time after initial exposure, that the existing capacity formulas were intended to accommodate.

The time lag before peak vaporization, as we stated earlier, is strongly dependent on the physical characteristics of the lading. Consider a liquefied compressed gas such as propane. The liquid is always at saturation (boiling) conditions during transport; therefore, incipient vaporization, hence pressure build-up, begins immediately upon the application of external heat, proceeding as previously described for saturation conditions. The net effect is that the time from initial exposure to heat to peak vapor generation can be much shorter than that for materials which are liquid at atmospheric pressure.

Presume, for the moment, that the tank shell can maintain its integrity at the designed operating pressure of the relief mechanism. Peak vapor rate capacity based on peak thermal input would not be required, if we could assure ourselves that, because of limited total fuel supply, or effective fire-fighting measures, the thermal capacitance represented by the tank and cargo could absorb the peak thermal input over the time interval of severe fire. Examination of the time-vapor rate question in more detail later in this report indicates that in the case of liquefied gases, it is doubtful that we should design for less than the potential peak vapor rate. The vessel integrity question, as it affects design, is discussed in subsequent paragraphs of this report. Assuming valves should be required to relieve vapor rates based on peak thermal input, Figure 7 illustrates the flow capacity requirements for assumed thermal flux levels. The flow capacity of safety-relief valves commonly applied to 33,000+ gallon 112A340W tank cars of approximately 2000-ft^2 shell area is also shown (dashed line). Given the 20,000+ Btu/h/ft^2 flux levels indicated by the literature search, valve flow capacity would be deficient--though the valves exceed the requirements of the current specification.

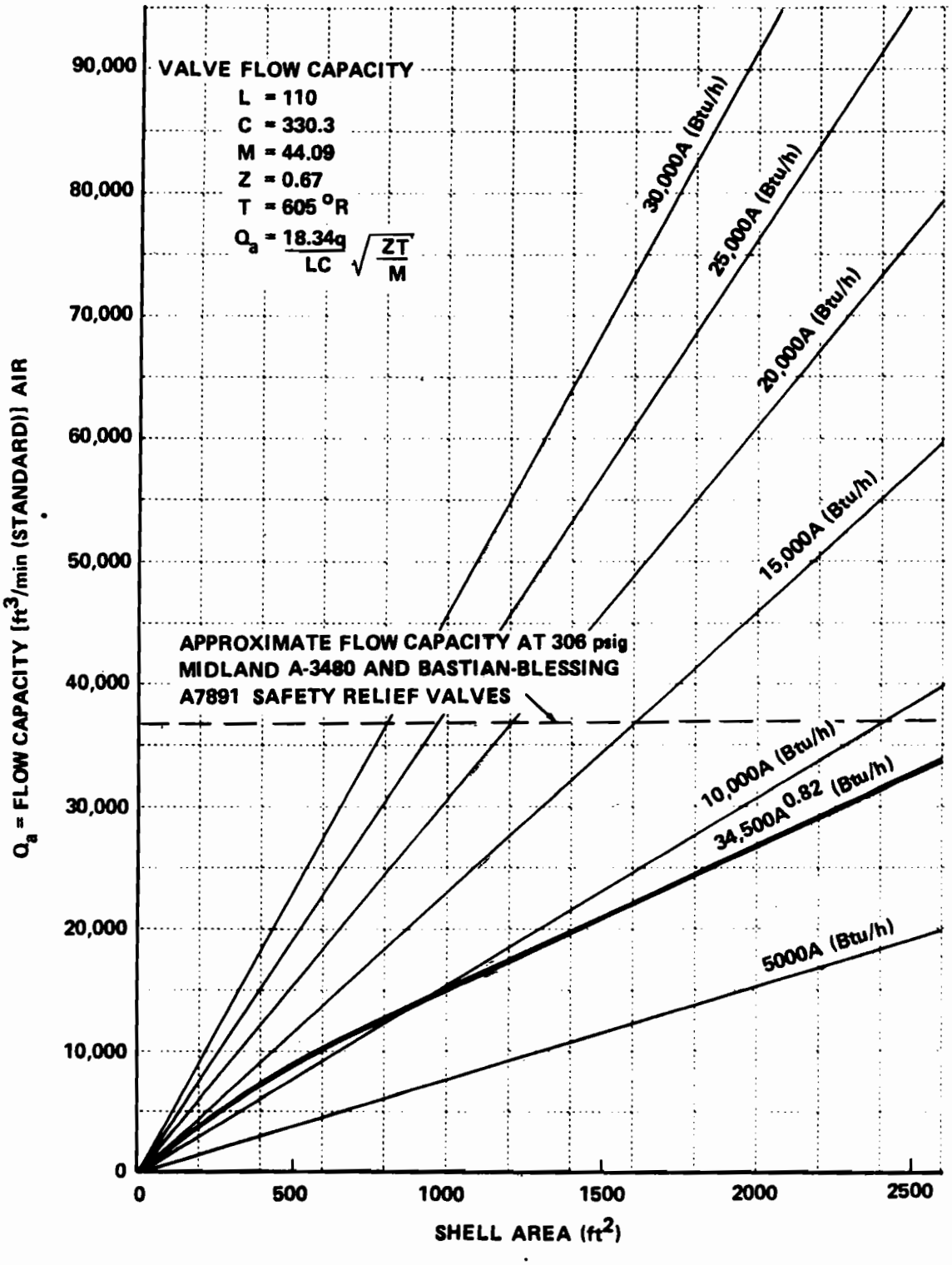

Figure 7 FLOW CAPACITY REQUIREMENTS AT ASSUMED FLUX LEVELS (PROPANE GAS): VAPOR RELIEF ONLY

The effect of vapor generation at a rate faster than it can be relieved results in an internal pressure "accumulation" above design limits, and if unchecked, can burst the shell. Such an accumulation could be particularly insidious in the derailment environment. Given the possibilities of mechanical damage and metal failure due to fire exposure, the manifestations could be well hidden. Did a weakened car rupture at normal operating pressure, or did an overpressure condition furnish the coup de grace? Physical evidence from wreckage would not necessarily supply a clue to the extent, if any, of contribution from this source. Testing of recovered safety valves for set-point and flow capacity to existing specifications would not supply the answer. Neither will evidence of metal oxidation and "thinning." On the former subject, a number of valves recovered from accidents involving fire have been determined to be in functioning order, attesting to the inherent ruggedness of their construction. In some cases, seals and "O" rings were intact, although the exterior was fire scarred. This would indicate these particular valves had opened and the heat sensitive parts were protected by the cooling effects of the escaping fluid (vapor or liquid). These findings would indicate internal pressure of the car had at least reached setpoint. The pressure could have been much greater than design levels with precisely the same result.

A safety-relief system is useless, regardless of flow capacity, if the pressure vessel fails below the operating pressure of the relief system. Outside of mechanical damage, fire exposure to the vapor space represents the principal threat to vessel integrity. Since fire exposure is a prime reason for requiring a safety valve, the effects of fire on the vessel itself are part and parcel of the overall safety design problem. Either relief flow capacity or inherent resistance of the vessel to fire can be the limiting safety factor. The designer must balance these factors at the desired level of protection. The vessel should survive, or fail in a safe manner, when exposed to the fire environment likely to be produced. Within the limits of economic considerations, it should be designed to survive the worst possible exposure condition.

Figure 8 relates tank car shell burst strength with temperature. The TC-128 steel strength was assumed to be equivalent to that of a sample tested for the AAR [19], and the wall thickness and shell diameter were chosen to approximate what might be found in a 112A340W car. For the particular example shown, the design safety margin above approximately 750°F becomes eroded at a rate on the order of 130 psi/100°F rise in temperature. Assuming an otherwise sound shell, we would not expect the car to survive, if the vapor space metal temperature reached approximately 1200°F, or correspondingly lower temperatures, if inadequate relief capacity permitted a pressure build up. The temperature factor is clearly quite critical. The ultimate temperature the metal could reach for a significant number of potential fires is probably only a few hundred degrees above the possible survival point for existing steels used in car construction. If a means of protection could be found to lower and/or delay the temperature rise to critical levels, it "buys" a great deal in terms of strength and potential survivability. Other avenues, such as increasing metal thickness, do not have much protection potential. For metal temperatures exceeding the transformation range of about 1300°F, increased thickness provides no significant gain. Below this level, a linear increase in strength with thickness is obtained, plus increase in heat capacity proportional to the added weight--the weight being a significant economic penalty.

Section 4 delves further into structural considerations.

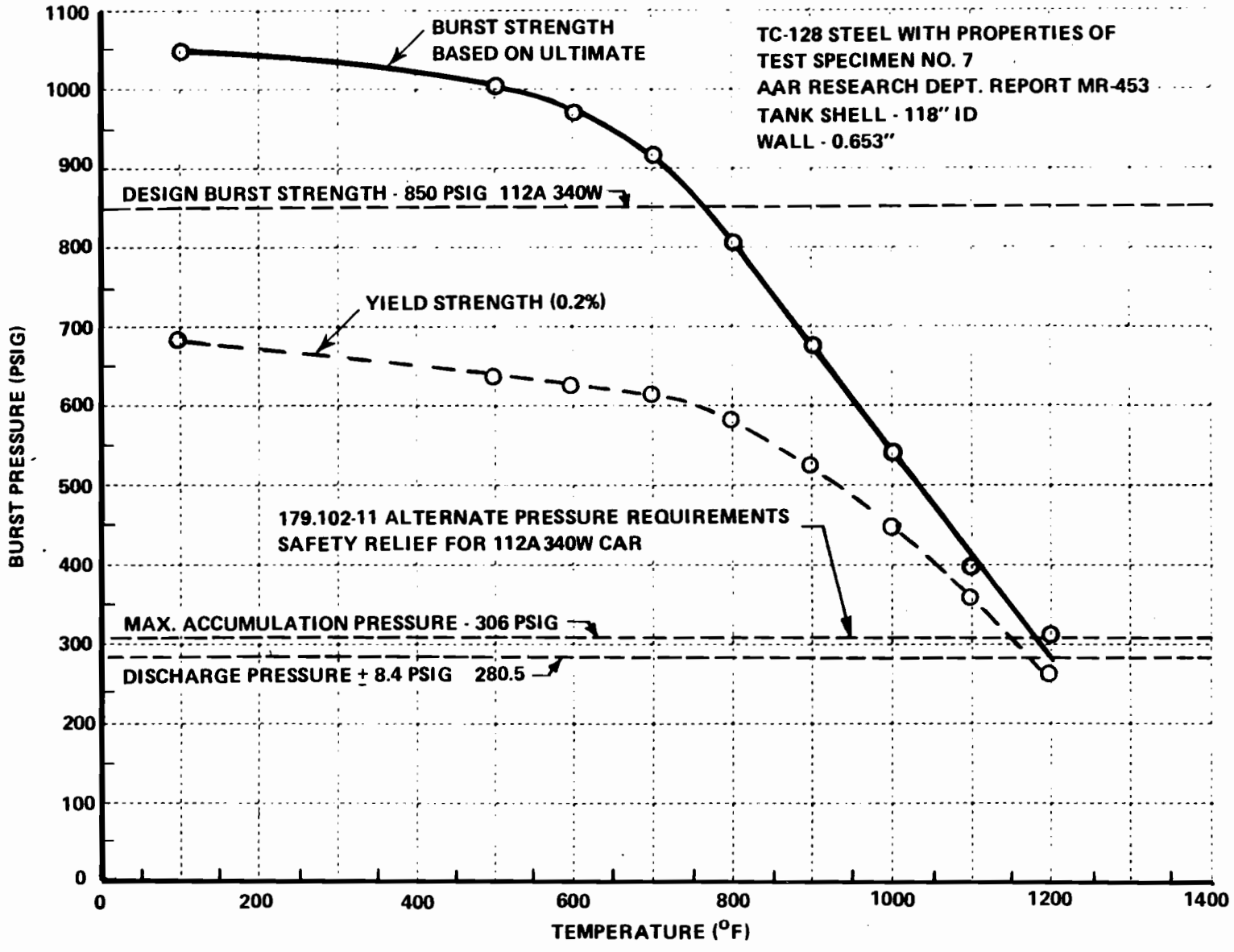

Figure 8 TANK CAR SHELL BURST STRENGTH VARIATION WITH TEMPERATURE

Section 4
PRESSURE TANK CAR DESIGN

4.1 STRUCTURAL CONSIDERATIONS—TANK CAR DESCRIPTION

The 112A340W-series tank car selected as an example for study is configured as a cylindrically shaped steel shell, 116 inches in outside diameter, employing 2:1 elliptical heads and having an overall tank length of approximately 65 feet. The tank itself is noninsulated, of frameless or monocoque construction (the tank shell forms the structural member), has a nominal capacity of 33,900 gallons and is mounted at each end upon a conventional four-wheel truck.

The tank shell (body plus heads) is of welded steel construction having a minimum specified thickness of 0.603 inch. The alloy largely favored for production is designated as TC-128-70 Grade-B steel, which has a tensile (ultimate) strength of 81,000 to 101,000 psi, a minimum yield point of 50,000 psi, and a minimum elongation (in 2 inches) of 19.00% as specified in AAR Appendix M of the AAR Specifications for Tank Cars.

The tank car is equipped with a single manway, approximately 20 inches in diameter, mounted on the upper surface of the tank at the center of the car. The manway cover, which forms the structural element across the opening, contains the several components of service and safety equipment, including inlet and outlet ports, temperature and liquid volume measuring devices and a single safety pressure relief valve. The latter unit is ordinarily either a Midland Manufacturing Company A-3480 or a Bastian-Blessing Company A7891 valve. The Midland A-3480 is rated by the manufacturer at 36,640 ft^3 of air per minute at a rating pressure of 306 psig. The corresponding flow rate for the Bastian-Blessing design is given as 37,040 ft^3/minute. The approximate area of the discharge orifice for these valves is 0.06 ft^2.

4.2 DESIGN BACKGROUND

The design of the 112A340W-series pressurized tank car is governed by the applicable specifications contained in the Hazardous Materials Regulations of the Department of Transportation (49 CFR, Part 179) republished as T.C. George's Tariff No. 23, and the AAR Specifications for Tank Cars--Standard, the current issue bearing an effective date of January 1, 1970. The latter document contains specifications in three general categories bearing the designations proposed, suspended and adopted. These designations are possibly not as descriptive as they imply, but rather represent the successive stages in the final adoption process of a given specification and are not necessarily in conformance with existing DOT specifications found in Title 49 Code of Federal Regulations, Part 179. Special permits may be requested for cars built to AAR specifications which have not received formal DOT approval and published, or cited, in the Federal Register, Code of Federal Regulations, or republished in tariff form.

Many of the present tank cars of interest have been constructed under the special permit provisions of DOT regulations. We, therefore, will continue the discussion referring to the AAR specifications as representative of current practice.

The specifications applying to the 112A340W-series tank cars are typical of the several other series of pressurized tank cars embraced by the Standard.

The AAR TC-128-70 plate from which the tank cars are fabricated is a relatively high-strength carbon-manganese steel having the mechanical properties noted above, possessing good elasticity for this type of material. It is readily joined by the fusion-welding process and does not rely on heat treating cycles to obtain its strength. It does, however, require a post-weld elevated temperature stress-relieving cycle to obtain uniform properties in the weld and weld-affected areas. Elevated temperature tests reveal no significant drop-off in tensile strength until approximately 600 to 700°F, when a steady rapid decline is indicated.

The heads of the tank shell are formed in a 2:1 elliptical shape to permit employment of the same thickness head as the shell, resulting in the same maximum membrane stress (at a given internal gas pressure) in the respective sections. This somewhat simplifies fabrication procedures. A welded-joint efficiency factor of 1.0 is permitted in the AAR specification for all joints except for seams in heads (if used), where a factor of 0.9 is required.

No welding of external supports, ancillary gear, jacking points, etc. is permitted directly to the tank surface. Instead, suitable reinforcing plates appropriately shaped and located are welded to the shell prior to the stress relieving cycle. All required attachments necessary to complete the tank car assembly are in turn attached (welded or otherwise) to these reinforcing pads. The specification requires that the detail design be executed such that the strength of items fastened to the pad attachment be only 85% of the pad-to-shell weld strength, thereby tending to preserve tank car shell integrity under overdesign loading conditions.

4.3 CURRENT DESIGN CRITERIA

The present design criteria for the tank car shell, the component of greatest interest here, are based primarily upon a structurally sound, undamaged car oriented in an upright position on the railroad track. As such, the structural design of the shell is governed by several sets of design loads and combination of loadings as specified in AAR Specification for Tank Car Subpart B--General Design and Test Requirements. These, in turn, reflect the normal anticipated loadings, including coupler and jacking loads as well as combinations of lading load, draft load and internal pressure load in the case of pressure tank cars such as the 112A340W under discussion.

The physical size and thickness of the tank shell yields section properties of considerable magnitude, such that despite the very substantial static loadings specified, the resultant maximum stresses are quite small

compared with the ambient temperature material allowable stress. For example, the critical combination of loads specified in AAR.23-3, Draft and Compressive End Loads--(b) of stub sill design, yields a maximum longitudinal stress of about 19,600 psi (tension in lower fibers) compared with an ultimate tensile allowable (minimum) of 81,000 psi and a yield point of 50,000 psi.

The specification further stipulates that the minimum tensile strength (81,000 psi) is to be divided by a factor of 3.7 for comparison with the critical stress calculated in accordance with the specification. On this basis, of course, the "allowable" stress is reduced to approximately 21,900 psi, much closer to the maximum computed value, but still yielding a positive margin over and above the 3.7 factor.

For all practical purposes, the limiting condition in the tank car shell is designated by the specified bursting pressure of (for the 112A340W car) 850 psi. The corresponding hydrostatic test pressure is 340 psi, per AAR Subpart C 179.101 Specification. The thickness required for the tank shell is determined by the maximum hoop or membrane stress from $S_t = \frac{pr}{t}$, where S_t = 81,000 psi, r (tank mean radius) is approximately 58 inches and the internal pressure p = 850 psi. This results in a required thickness of approximately 0.603 inch for the shell. Employing 2:1 seamless elliptical heads results in the same required thickness for that component. Hence, there exists a pressure vessel uniformly capable of withstanding an internal pressure of 850 psi as a minimum without bursting at ambient temperature conditions.

AAR Subpart C 179.101 Specifications for Pressure Tank Cars also stipulates a safety-relief valve start-to-discharge pressure of 255 psi (with a tolerance of ±7.65 psi) and a minimum vapor-tight pressure of 204 psi. The valve flow rating pressure for the unit is given as 280.5 psi.

However, there is an alternate or optional specification allowed for tank cars carrying certain ladings, including the particular liquefied petroleum gas selected as an example material for this study, delineated in

AAR 179.101-11 which permits increased valve settings "provided the total valve discharge capacity is sufficient to prevent building up pressure in the tank in excess of 90% of the tank test pressure." The corresponding values are; start-to-discharge pressure -- 280.5 psig (with a tolerance of ±8.4 psi), vapor tight pressure (minimum) -- 224 psig and relief valve flow rating pressure -- 306 psig. Current practice has been to employ the optional criteria extensively in new car production, utilizing either the Midland Manufacturing Company A-3480 or the Bastian-Blessing Company A7891 valve as the safety relief device. It is presumed that the alternate standard was adopted to more satisfactorily accommodate the high vapor pressure of anhydrous ammonia and "ethane rich" commercial propane.

It should not be construed that the preceding discussion of current draft and compressive end load criteria represents an evaluation of the existing standards (e.g. "3.7 factor"). The discussion is descriptive only, illustrating that under existing criteria the burst strength specification controls the minimum shell wall thickness for the example 112A340W car.

4.4 DERAILMENT CONDITION DESIGN FACTORS AND FAILURE MODES

In the discussion above we observe that the existing tank car shell is designed essentially to a static-type loading condition based primarily upon a design burst pressure well above the maximum accumulation pressure (flow rating pressure) of the single safety relief valve. As such, it enjoys a substantial margin of safety (against rupture), considering ambient temperature conditions, the allowable tensile strength of the tank material, a structurally sound (undamaged) tank shell, and a correctly functioning safety relief valve of adequate flow capacity preventing pressure accumulations in excess of the flow rating value. Under these assumed conditions, the factor of safety for the shell is as follows:

Maximum Hoop Stress

$$S_t = \frac{P_i r}{t} \qquad P_s = 306 \text{ psig}, \; r = 58 \text{ in.}, \; t = 0.603 \text{ in.}$$

$$= \frac{(306)(58)}{0.603} \tag{16}$$

$$\cong 29{,}000 \text{ psi (tension)}$$

Minimum Tensile Allowable if TC-128-70 Grade B Steel, F = 81,000 psi. Hence,

$$\text{Factor of Safety} = \frac{F}{S_c}, \text{ or } \left(\frac{81{,}000}{29{,}000}\right) = 2.8 \tag{17}$$

In other words there is an indicated ability of the tank shell to withstand approximately three times the expected peak internal pressure controlled by the safety relief valve, ignoring the valve pressure tolerances for simplicity. Since the 81,000 tensile allowable is the lower limit of a specified range of 81,000 to 101,000 psi, it may be seen that the indicated factor of safety may be greater than that calculated above. Additionally, there is a factor of safety of almost 1.7 based upon the yield point of the steel which has a minimum specified value of 50,000 psi. Hence, the tank car emerges as a husky shell fabricated from relatively tough material capable of withstanding several times the maximum expected loads as a static pressure shell. This situation can change drastically when confronted with a train derailment involving cars loaded, for example, with liquefied propane gas.

Reviewing briefly, the present study is principally directed toward preventing the catastrophic failure of pressurized tank cars such as 112A340W series loaded with LPG, subsequent to train derailments resulting from any cause.

The possibility of such occurrences has been dramatically illustrated in recent incidents and requires no further clarification. Of interest here are the several potential modes of tank shell failure, the various factors playing important roles in those failures, and the implications concerning the structural design of the tank car together with its auxiliary equipment.

Diverting for a moment, the initial steps in a design philosophy aimed at preventing catastrophes would be to prevent the tank car from leaving the track. Under present (and probably future) circumstances associated with rail transportation, this is essentially statistically impossible. Given a derailment involving LPG tank cars, the next step in the sequence would be to prevent the penetration of the shell of any of the cars involved. Although some improvement in this area can certainly be effected, it is doubtful that all violations of the tank shell structural integrity could be eliminated as the result of a derailment within the present concept of rail transportation. If we then assume that, as appears likely, at least one car in the derailment will be punctured (and certainly there are many ways in which this can occur, then we face the problem of preventing catastrophic failure from occurring as the result of either a single car exploding, "rocketing" of whole sections of tanks, or "domino" failure of successive cars as the fire spreads, all of which have been demonstrated in recent disasters. The assumption made in this study is that at least one LPG car has been punctured and is burning, with other LPG cars located nearby.

Review of the circumstances surrounding a "typical" derailment suggests that a failure can occur in any one of several modes during or following the actual derailment, including variations in some cases. These postulated failures, which can lead to immediate or subsequent catastrophic failure, may be placed in the following general categories:

(1) Damage to the shell structure resulting in a reduction in strength below the design values.

(2) Reduction in strength of the shell material as a result of elevated temperature conditions.

(3) Overdesign pressurization of the shell as a consequence of inadequate relief valve discharge capacity under fire exposure conditions.

(4) Hydrostatic failure of the shell under the "shell full" condition (no vapor in tank).

These generalized modes of failures are, in turn, affected by a substantial number of design and environmental factors as well as the procedures and techniques employed in the shipment of commodities such as LPG by rail transportation. Included among these are

- Shell temperature
- Heat flux through shell
- Tank internal pressure
- Shell material and mechanical properties over the appropriate temperature range
- Original or as-fabricated condition of the material
- Temperature gradients in the shell
- Local discontinuities in the structure
- Loading density of a two-phase lading such as LPG
- Presence or lack of thermal insulation around the shell
- Safety relief valve discharge capacity in
  - vapor phase
  - liquid phase

- Dynamic loadings at impact during derailment
- Ambient (air and lading) temperature at impact or derailment
- Terrain features at derailment site (obstructions, sharp-edged objects, or construction of any type)
- Survival of the single relief valve and ability to function in that capacity after the accident

There are probably others, but the above are believed to present in large measure the far ranging involvement and broad spectrum of factors which can apply to a complex problem such as attempting to maintain the structural integrity of an LPG laden tank car shell during and following a derailment at any arbitrary location in the track system.

Returning for a brief discussion of some of the failure modes noted above, consider the first category listed, involving damage to the shell structure resulting in a reduction in strength below the original design level. This is perhaps the most simple failure mode which can occur and in the extreme case, could involve the immediate puncturing of the car and disgorging of the LPG contents, thus furnishing the initial heat source which could lead to the now well-documented chain reaction sequence. Puncturing can be accomplished by such items as car couplers, other cars, rails, or any number of sharp-edged or substantial obstructions adjacent to the tank at the derailment scene. In this situation, there is less likelihood of the punctured car failing catastrophically; rather, it could form the source of the large heat flux which causes subsequent massive failures of other LPG cars which survive the initial derailment or which may be left standing undamaged on the track.

Of possibly greater concern in the overall picture is the car which suffers significant structural damage at derailment, but which does not puncture or rupture. This tank shell represents a significant hazard in the presence of large heat fluxes, which can be generated by other cars of the same

or other flammable lading involved in the accident, resulting in increased internal vapor pressure. For example, it is entirely plausible to postulate that the shell rupture strength might be reduced to something less than the 280.5 ±8.4 psi relief valve set pressure such that the tank could rupture, possibly catastrophically, with a normal (but closed) safety valve. Alternatively, the damage sustained by the tank could be less severe, but contribute to the failure when combined with other modes discussed below.

The second category listed involves failure of the tank as a result of a reduced tensile strength of shell material, brought about by the elevated temperature condition present during burning of fuel escaping from one or more cars. Based upon a test sample of TC-128-70 Grade B steel (reference AAR Research Department Report MR-453) considered typical for this alloy, the tensile strength remains fairly constant, decaying only slightly with increased temperature until about 650°F is reached, at which point a rapid, constant reduction in strength is experienced (see Figure 8 in previous section).

For this particular sample, the tensile strength remains adequate to contain the design burst pressure of 850 psi until a temperature of about 770°F is reached. Above this temperature, the strength degrades rapidly until a value corresponding to the maximum allowable accumulation pressure of 306 psi is reached at approximately 1180°F. At this point, the tank would presumably rupture regardless of its condition prior to initiation of heating. If, additionally, the tank shell had suffered structural damage prior to, during or following derailment, its strength could still be further eroded and rupture could occur at greater internal pressure and lower temperature as discussed below.

There likewise remains the additional possibility of the currently employed single safety valve being damaged during impact, thereby preventing the intended pressure relief function from being executed. This, of course, would further aggravate the situation and tend to result in shell rupture at greater pressures, with increasing likelihood of catastrophic results.

The third category of failure mode involves overpressurization of the tank shell resulting from the inability of a functioning safety pressure relief valve to discharge the necessary volume to prevent internal pressure build up.

This type of failure may occur as a result of thermal loads exceeding those anticipated when sizing the valve. Failure could also occur as a result of altered functioning characteristics and/or flow capacity when discharging certain ladings at other than vapor feed conditions (e.g., liquid feed with flashing discharge).

This aspect of the complex thermodynamic problem associated with two-phase ladings such as LPG subjected to large thermal loads, together with the origin and magnitude of the heat flux, etc., is treated in detail in another section of this report. Our interest here is primarily in defining various types of failure modes which must be considered in a systematic evaluation of all possible factors affecting the catastrophic rupture of these tank cars and focusing our attention on the requirements for preventing this occurrence.

Consider the volume-pressure-temperature relationship for a fully loaded 112A340W-series LPG tank car. For discussion purposes, assume the car is loaded at a lading temperature of 60°F. Under current practice, this corresponds to an outage of about 10.3% (summer fill) and an internal tank pressure of approximately 93 psig (108 psia) at the saturated conditions which prevail.

Now, assuming this tank car to be standing in the hot sun on a warm day, the temperature will increase, the pressure will increase, and the outage will decrease. The liquid-vapor relationship of propane is such that, under saturated conditions, at a temperature of approximately 115°F, the outage will decrease to zero percent and the tank will be completely full of liquid propane, i.e. a shell-full condition. The corresponding pressure for

this condition is 213 psig (228 psia). Since the set pressure of the safety relief valve is 280.5 $\pm$ 8.4 psi, it could not be expected to function in any way to prevent this occurrence under the conditions specified. Under normal conditions, a slight increase in temperature will result in a relatively rapid increase in pressure from 213 psig to the relief valve set pressure of 280.5 psig and the excess propane would vent to the atmosphere.

The probability of the temperature increasing to 115°F, yielding the shell-full condition under normal conditions, will vary considerably with a number of factors. The point here is that increasing temperature does result in decreased vapor volume, approaching the shell-full condition at an easily realizable ambient temperature. Under low-intensity thermal exposure conditions, this presents no immediate problem, since a properly functioning relief valve will prevent overpressurization from occurring.

However, consider the situation where, on a hot day, a train hauling LPG tank cars is involved in a derailment resulting in upset tank cars among others. The force of a loaded propane car travelling at 40 to 60 mi/h impacting on another railroad car, ground obstructions such as foundations, buildings, railroad tracks or even a local terrain discontinuity could tend to crush the steel shell locally, resulting in elimination of the vapor space entirely and proceeding immediately to the shell-full condition. The safety relief valve may not be able to dump the volume of propane required in the extremely small time increment involved in the impact to relieve the pressure buildup. Hence, the tank shell could be expected to rupture catastrophically under the hydrostatic pressure generated by the impact. Obviously, the structural damage which might logically occur during such a derailment would further reduce the probability of shell survival.

Of further interest is the fact that a shell-full tank car loaded with LPG could represent a distinct hazard whether directly involved in a derailment or merely standing on the track in a position to absorb a mechanical impact of any nature. Theoretically, any reduction in the internal volume of the tank car will result in an immediate sharp increase in pressure. Under these circumstances, a flying fragment, possibly propelled from another car during a derailment situation, could lead to tank rupture instead of inflicting mechanical damage of, perhaps, minor nature.

4.5 IMPLICATIONS ON FUTURE TANK CAR CONSTRUCTION OR MODIFICATION

It is clear that a new safety relief-system alone will not totally solve car rupture problems. While detailed consideration of other car design factors were beyond the scope of this study, they are mentioned where clear interrelationship to the safety problem exists. The following paragraphs under this subheading, however, refer strictly to application of a safety device.

New car construction should offer the least problems in implementing the finalized requirements to be incorporated into a workable tank car design aimed at meeting the objective of preventing catastrophic tank car rupture such as has been experienced in recent derailments.

For the existing fleet of tank cars a modification program could be initiated to incorporate the necessary changes required to reduce the probability of tank car failure to an acceptable level. Such a program might include a change of the existing single relief safety valve, or an increase in the number or type of safety pressure devices (see Recommendations). Depending upon the finalized requirements, the modification possibly could be confined to the present manway opening at the top center of the car. Conceivably, the necessary changes could be accommodated in a redesigned manway cover fitting into the existing opening.

Alternately, it may require an enlarged manway opening and, in fact, it may prove desirable to locate additional relief devices at other locations. In any event, a preliminary study has confirmed that this general type of modification can, and is being accomplished on tank car shells in routine repair operations. Cutting into an existing shell requires either a preheat or post-weld stress relieving operation, with some sources apparently [illegible]ng the preheat cycle. Either method is amenable to a localized operation which could readily be installed on a production line basis at a suitably equipped tank car modification center. Provision for alteration of existing cars is currently made in T.C. George's Tariff No. 23, Subparagraph 173.31 citing Appendix "R" of the AAR specifications for tank cars.

Section 5
THERMODYNAMIC CONSIDERATIONS IN RELIEF SYSTEM DESIGN

5.1 INTRODUCTION

This section contains a detailed study of the thermodynamic aspects which must be considered in achieving satisfactory relief system design.

Many of the results of this study have general application, but to place their utility in focus, we have treated a particular fire exposure case. The example selected was a nominal 33,000-gallon capacity 112A340W car filled at "summer" levels with propane. Thermodynamic data for pure propane were used for convenience as a close approximation to the values for the wider distillation "cut" of commercial propane.

There are approximately 17,000 cars* of the 112 series in service. LPG ladings, with propane representing the principal constituent, constitute a large shipping volume. This car class and LPG lading have been involved in most of the more spectacular incidents. Therefore, this choice of car and lading gives information of immediate and practical significance.

5.2 COMBUSTION TEMPERATURE FOR PROPANE-AIR MIXTURES

It is necessary to obtain a reasonable estimate of the fire temperature produced in the combustion of propane-air mixtures to assess the temperature potential which is effective in the heating of tank car shells. For propane-air mixtures with insufficient air, as in fire conditions, the products of combustion cannot be established precisely, but one can calculate parametrically the effects of typical product proportions. Consider the reaction

*Estimated from References 20 and 21 and contact with domestic suppliers.

$$(C_3H_8)_g + n\,O_2 + 3.76\,n\,N_2 \rightleftharpoons x\,C + y\,CO + z\,CO_2 + 4(H_2O)_g + 3.76\,n\,N_2 \quad (18)$$

Propane — Oxygen, Nitrogen (Air) — "Free" Carbon — Carbon Monoxide — Carbon Dioxide — Water as Vapor

as the most likely to occur under fire conditions.* Further, a mass balance gives

$$x + y + z = 3 \quad (19)$$

$$n = \frac{y + 2z + 4}{2} \quad (20)$$

Also, the heat release in this reaction, utilizing standard heat of formation data, is

$$Q = z(66{,}767) + y(27{,}202) + 4(57{,}100) \quad \text{calories} \quad (21)$$

The temperature increase of the products at constant pressure may be approximated utilizing the heat release and specific heat data for the constituents, as

$$\Delta T = \frac{Q}{4(M\bar{c}_p)_{H_2O} + x(M\bar{c})_c + y(M_{co}\bar{c}_p)_{co} + z(M\bar{c}_p)_{co_2} + 3.76\,n(M\bar{c}_p)_{N_2}} \quad (22)$$

Where

$(M\bar{c}_p)_{H_2O} \approx 11.5$ $\quad (M\bar{c}_p)_{co_2} \approx 14$ $\quad \frac{Cal}{gm\text{-}mole\ ^\circ C}$

$(M\bar{c})_c \approx 4$ $\quad (M\bar{c}_p)_{N_2} \approx 8.3$

$(M\bar{c}_p)_{co} \approx 8.4$

at temperatures near 1400°C (2500°F). Hence,

*There are other possible constituents including OH, H_2, and CH_4, but these are expected to react quickly on contact with oxygen.

$$\Delta T = \frac{z(66,767) + y(27,202) + 228,000}{46 + 4x + 8.4y + 14z + 31.2n} \quad °C \qquad (23)$$

Using Equations (19), (20), and (23), one can perform a sensitivity analysis by variation of the factors x, y, and z, which represent the amount of carbon in each of the constituents C, CO, and CO_2, respectively. This is shown in Table 4. For this reaction, it is evident that little effect on maximum flame temperature is introduced by incomplete combustion of the carbon. Hence, where combustion is proceeding, temperatures in excess of 3000°F* can be produced. In actual propane-air fire, there will, however, be locations at which the propane vapor contacts essentially no air; and at these locations (generally toward the center of the fire near the fuel supply), temperatures may be considerably below 3000°F. As an example of magnitudes, several simple experiments were conducted using a commercially available Bernzomatic propane cylinder with the air mix nozzle removed. Temperatures within the flame after ignition were measured using an unshielded platinum-platinum 10 percent rhodium thermocouple made from 0.005-inch-diameter wires. The propane flow rate was about 0.1 gm/s, resulting in a fire nominally 0.5 inch in diameter near the propane exhaust port and increasing to about 3 inches in diameter at 10 inches from the port. Figure 9 illustrates maximum temperature values obtained as a function of position above the exhaust. Clearly, the temperatures near the outer layers of the fire are above 2200°F, reaching as much as 2700°F. Further, the central temperature is observed to increase with distance from the exhaust, reaching about 2200°F at 10 inches. It must be noted that the measured temperatures are somewhat low because an unshielded thermocouple was used. In addition, the above calculations of flame temperature are based on adiabatic conditions, whereas fires exhibit some energy loss. Nevertheless, one concludes that flame temperatures in excess of 2200°F are easily produced during combustion of propane without forced mixing of air.

*Lewis and Von Elbe, Combustion Flames and Explosions of Gases, Academic Press, 1951, p. 766, reports a measured flame temperature of 3500°F.

Table 4
SENSITIVITY OF ADIABATIC FLAME TEMPERATURE OF PROPANE TO DEGREE OF COMBUSTION

X = 0				
y	z	n	ΔT, °C	ΔT, °F
0	3	5	1830	3330
1	2	4.5	1750	3180
2	1	4.0	1740	3160
3	0	3.5	1710	3110
X = 1				
y	z	n	ΔT, °C	ΔT, °F
0	2	4	1780	3230
1	1	3.5	1780	3230
2	0	3	1760	3200
X = 2				
y	z	n	ΔT, °C	ΔT, °F
0	1	3	1820	3310
1	2	2.5	1820	3310
X = 3				
y	z	n	ΔT, °C	ΔT, °F
0	0	2	1900	3450

x = MOLES CARBON
y = MOLES CARBON MONOXIDE
z = MOLES CARBON DIOXIDE
n = MOLES ATMOSPHERIC OXYGEN

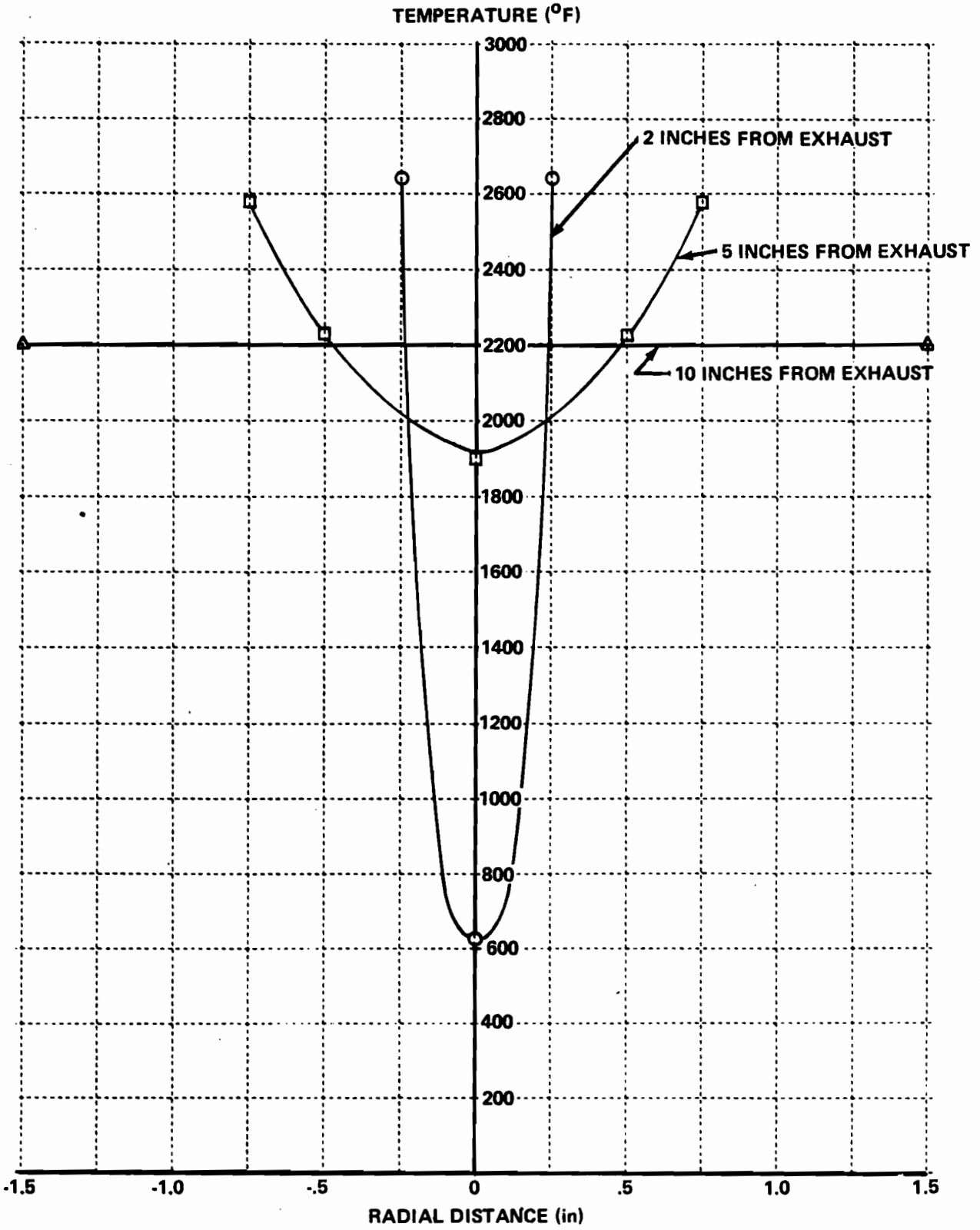

Figure 9 MAXIMUM TEMPERATURES MEASURED IN FLAME FROM SMALL PROPANE CYLINDER WITHOUT PRIMARY AIR MIXING

5.3 FIRE VOLUME AND DURATION

Upon release of saturated liquid propane to ambient pressure, such as when the safety valve opens from excessive pressures or the tank ruptures due to an accident, a significant proportion of liquid can immediately be vaporized without need for external heat supply. Energy for vaporization is extracted from the liquid, which possesses an excess of enthalpy required for saturation at the lower ambient pressure. For any rate of propane release, M_{TOT}, a minimum rate of vapor generation, M_V, given by

$$\dot{M}_V = \dot{M}_{TOT}\left[\frac{h_{f,p_o} - h_{f,amb}}{L_{amb}}\right] \qquad (24)$$

will exist. In this expression

h_{f,p_o} is the enthalpy of the saturated liquid at tank pressure,

$h_{f,amb}$ is the enthalpy of the saturated liquid at ambient pressure,

L_{amb} is the latent heat at ambient pressure.

Table 5 illustrates the rate of vapor generation per unit rate of liquid flow during exhaust to ambient from various initial saturated tank pressures.

Table 5

P_o TANK PRESSURE (psia)	SATURATION TEMPERATURE (^oF)	$\dot{M}_V/\dot{M}_{TOT}$ (lb/s / lb/s)
45.85	10	0.162
65.70	30	0.225
106.90	60	0.32
165.00	90	0.42
243.40	120	0.53
308.40	140	0.61
345.40	150	0.65
473.20	180	0.80
575.00	200	0.94

Obviously, the higher the initial tank temperature at exhaust of liquid, the greater will be the proportion vaporized. At the nominal* zero outage condition ($\approx 115°F$), approximately 50 percent of the liquid released will be in the form of vapor. At a 280.5 psig set pressure of the relief valve, the proportion increases to approximately 60 percent. The effect of this large conversion to vapor is twofold. First, it enhances the possibility that large fires can immediately be produced, and second, it can create substantial propulsive force upon failure of the tank car shell. With respect to the fire volume, the amount of liquid remaining after expulsion from the tank is

$$\dot{M}_L = \dot{M}_{TOT} - \dot{M}_V \tag{25}$$

This liquid must be vaporized by additional energy from the environment. Assuming that the liquid is vaporized solely by radiation from the fire source, the maximum area of the liquid pool for instantaneous vaporization of total pool contents to occur is

$$A_{max} = \frac{\dot{M}_L L_{amb}}{q_{rad}} \tag{26}$$

where
q_{rad} = radiant heat flux

For any area less than this, there will be a liquid storage which will increase the fire duration, but reduce its size. Direct calculation of fire size is difficult to perform, but one can obtain at least an order of magnitude assessment of the fire situation. The heat input due to combustion of a propane-air mixture is of the order of 15,000 Btu/lb propane. Hence, for a liquid release rate of $\dot{M}_{TOT}$, the heat generation is of the order of $15{,}000\,\dot{M}_{TOT}$. Of this heat in a steady fire, some is lost by convection of gases through the fire and some by radiation. As noted earlier, adiabatic temperatures of the order of 3000°F can be produced. Gases flowing through

*Based on authorized filling densities for propane (0.51 specific gravity) at summer conditions.

the fire will absorb a majority of the total heat release. Assuming the gases above the fire to reach temperatures of at least 2000°F, only 1/3 the total input energy is lost by radiation. A cylindrical fire with height equal to its base will lose heat by radiation according to

$$\dot{Q}_{LOSS} = \left[\frac{3\pi D^2}{2}\right] \epsilon \sigma T_g^4 = \frac{1}{3} \dot{M}_{TOT} (15{,}000) \tag{27}$$

D is the fire diameter, ϵ the emissivity of the fire, σ the Stefan-Boltzmann constant, and T_g the fire temperature. At 3000°F,

$$\sigma T_g^4 = 68 \ \frac{BTU}{Ft^2 \ Sec} \tag{28}$$

Thus,

$$D \approx 4 \sqrt{\frac{\dot{M}_{TOT}}{\epsilon}} \quad \text{feet} \tag{29}$$

is the order of diameter of "hot" fire.

The minimum fire duration is governed by the total mass of propane in the tank M_{TOT} and the release rate or

$$\tau_{min} \approx \frac{M_{TOT}}{\dot{M}_{TOT}}$$
$$\epsilon \tau_{min} \approx \frac{M_{TOT}}{\dot{M}_{TOT}/\epsilon} \tag{30}$$

Table 6 illustrates fire diameter and minimum fire duration as a function of $\dot{M}_{TOT}/\epsilon$ for M_{TOT} = 120,000 lb, approximately a typical load in a 33,500-gal 112A340W car with propane lading. Figure 10 further illustrates the tabulated values. The emissivity, ϵ, of the fire is unknown but is limited to a maximum of unity. Stull and Plass [23] report blackbody conditions (ϵ = 1.0) in the flame depth range from 50 to 5 x 10^5 cm, depending upon number and size of carbon particles. Dalzell and Sarofim [24] indicate blackbody conditions to be approached near fire depths of 100 meters. There is little doubt,

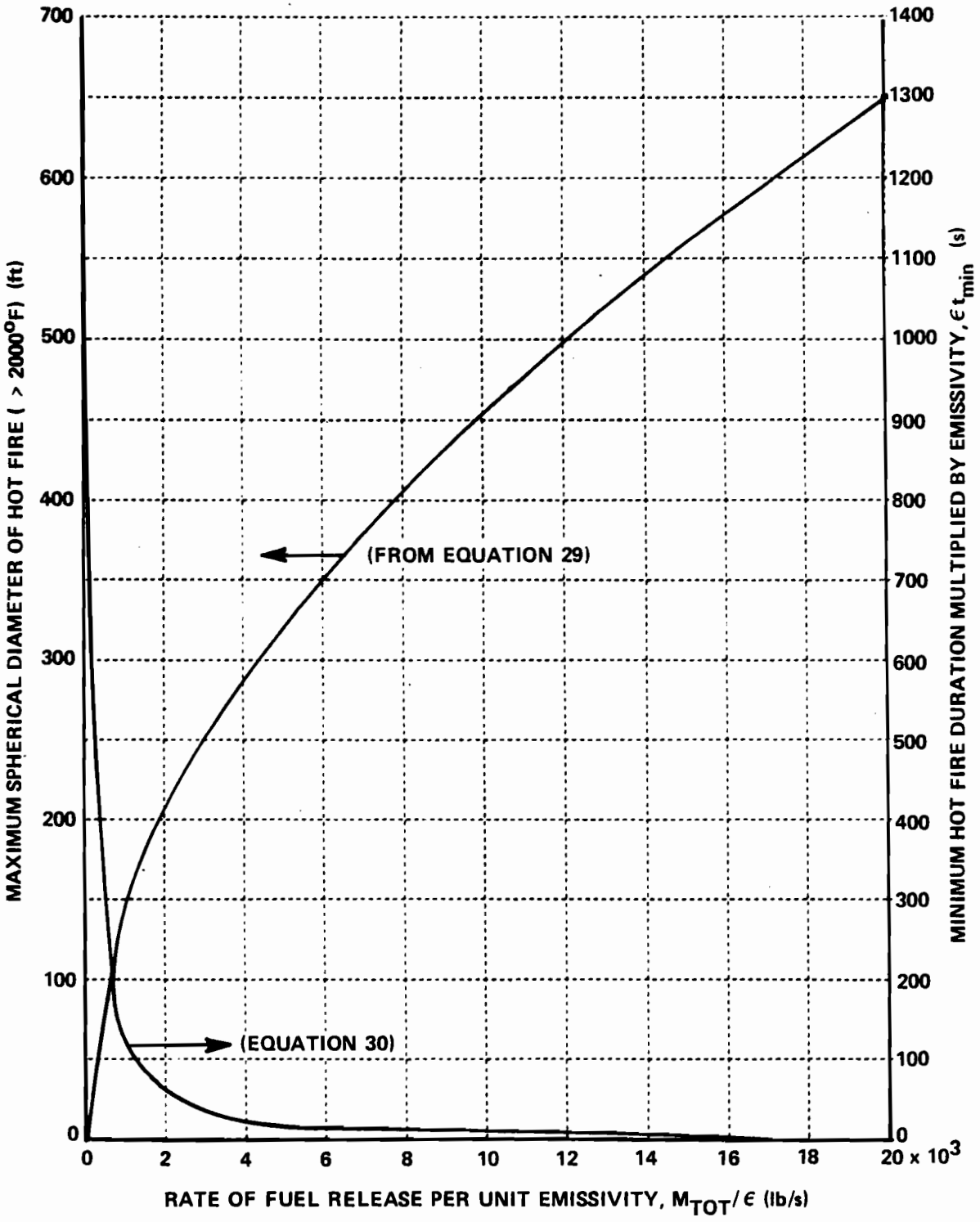

Figure 10 FIRE DIAMETER AND MINIMUM FIRE DURATION, M_{TOT} = 120,000 lb

Table 6

FIRE DIAMETERS AND DURATION

$\dot{M}_{TOT}/\epsilon$ (lb/s/UNIT EMISSIVITY)	D (ft)	D_s EQUIVALENT SPHERICAL DIAMETER (ft)	ϵt min. (s)
20,000	565	650	6
10,000	400	460	12
5,000	280	320	24
2,000	180	210	60
1,000	125	140	120
500	90	100	240
100	40	45	1200

therefore, that blackbody conditions can be approached in very large fires, but, on the other hand, there is little fundamental basis for specifying actual emissivity of real fires. For this reason, calculations here have been normalized with respect to emissivity.

The total heat by radiation per unit area at a distance L from the fire center is approximately

$$Q_{TOT} = \left(\frac{D}{2L}\right)^2 \epsilon \tau_{min} \sigma T_y^4 \quad , \quad L > D/2 \tag{31}$$

and

$$Q_{TOT} = \epsilon \tau_{min} \sigma T_y^4 \quad , \quad L < D/2 \tag{32}$$

From Equations (27), (28), (29), and (30), outside the fire,

$$Q_{TOT} \approx \frac{16 \frac{\dot{M}_{TOT}}{\epsilon}}{4L^2} \left[\frac{M_{TOT}}{\dot{M}_{TOT}/\epsilon}\right] \approx \frac{270}{L^2} M_{TOT} \tag{33}$$

Similarly, inside the fire,

$$Q_{TOT} = \frac{68\, M_{TOT}}{\dot{M}_{TOT}/\epsilon} \tag{34}$$

Hence, the total heat flux by radiation per square foot of surface is relatively insensitive to emissivity at given distances outside the fire center but is directly dependent on emissivity within the fire.

As an example of magnitudes, consider a liquid release of 100 lb/s and $\epsilon = 0.1$. Then, the equivalent spherical diameter of the fire is $D_S = 145$ ft and the fire time is 1200 s. At 100 ft from the fire center, the total heat flux by radiation is

$$Q_{TOT} = \frac{270(120{,}000)}{(100)^2} = 3240 \ \frac{Btu}{Ft^2}$$

and the average flux rate by radiation is

$$q_{ave} = \frac{3240}{1200} = 2.7 \ \frac{Btu}{Ft^2\,sec} = 9700 \ \frac{Btu}{Ft^2\,Hr}$$

Within the fire, the average flux rate by radiation is

$$q_{ave} = 0.1(68) = 6.8 \ \frac{Btu}{Ft^2\,sec} = 24{,}500 \ \frac{Btu}{Ft^2\,Hr}$$

The above are only order of magnitude approximations of the fire situation. Nonetheless, it is apparent that fire volumes much in excess of the tank car volume are possible, and, in fact, this is borne out by photographs of recent propane tank car fires [25]. One must consider, therefore, that the entire tank car shell is exposed to the fire source in calculations of safety requirements.

5.4 HEAT TRANSFER TO TANK CAR SHELLS

The net heat input into the tank car shell engulfed in flame is dependent upon (1) the average flame temperature, (2) the effective emissivity

of the flame, (3) the emissivity of the shell, (4) the effective heat transfer coefficient, and (5) the shell temperature. A relationship that expresses the interdependence of these parameters in the vapor space is

$$q_{net} = h_{ave}(\bar{T}_g - \bar{T}_s) + \sigma\epsilon\epsilon_c(\bar{T}_g + 460)^4 - \sigma\epsilon_o(\bar{T}_s + 460)^4 - \sigma\epsilon_i(\bar{T}_s + 460)^4 - h_v(\bar{T}_s - T_L) \quad (35)$$

h_{ave} = the effective outside heat transfer coefficient (Btu/ft^2-h-°F)

$\bar{T}_g$ = the average flame temperature (°F)

σ = the Stefan - Boltzmann constant (0.173 x 10^{-8} Btu/h-ft^2 -°R^4)

$\bar{T}_s$ = the shell temperature (°F)

ϵ = the effective emissivity of the flame

h_v = the heat transfer coefficient in the vapor space (Btu/ft^2-h-°F)

T_L = the temperature of saturated vapor (°F)

ϵ_o = the emissivity of the outer shell surface

ϵ_i = the emissivity of the inner shell surface.

The magnitudes of the parameters h_{ave}, h_v, $\bar{T}_g$, ϵ, ϵ_o, ϵ_i, and T_L determine the maximum shell temperature produced in a fire. This is obtained by a solution of Equation (35), with q_{net} set equal to zero. In seeking appropriate magnitudes upon which safety requirements can be based, one must account for observed shell temperatures in actual fires. A conservative value for this maximum shell temperature is 1200°F, as structural metals have been found to buckle and exhibit metallic transformations characteristic of this temperature range in fire conditions.* Here, a parametric analysis can be helpful in establishing magnitudes.

*See Reference 19, p. 52906.A41; also References 9, 10, 11, and 26.

The emissivities of the shell, ϵ_o and ϵ_i, are probably not far removed from 0.8 at temperatures near 1200°F, because there will be carbonization due to decomposition of the propane vapors in the tank and soot deposits outside the tank. The liquid temperature, T_L, will be low, say less than 150°F, and the convection coefficient in the vapor, h_v, will be of the order 3 Btu/ft^2-h-°F. Hence, for equilibrium at 1200°F, the average convection coefficient must be

$$h_{ave} = \frac{3(1200-150) + 2\sigma(0.8)(1660)^4 - \sigma(0.8)\epsilon(\bar{T}_g+460)^4}{(\bar{T}_g - 1200)} \tag{36}$$

The range of average flame temperature is from 1500°F to 2500°F. The flame emissivity is unknown. Figure 11 depicts the effect of selection of emissivity ϵ on the required heat transfer coefficient at flame temperatures of 1500, 2000, and 2500°F.

In addition, Figure 11 illustrates the "cold" wall heat fluxes to a surface at 100°F produced by each coefficient-emissivity combination. Figure 11 also indicates that the "cold" wall heat flux must be above at least 25,000 Btu/ft^2-h to produce a shell temperature of 1200°F for the range of possible flame temperatures. In addition, "cold" wall heat fluxes of the order of 35,000 Btu/ft^2-h have been observed where surfaces have been exposed to fire conditions [13]. Figure 11 indicates this flux to be obtainable with the combinations

	$\bar{T}_g$ °F	h_{ave} Btu/ft^2-h-°F	ϵ
(1)	1500	≈ 10	≈ 1.0
(2)	2000	≈ 10	≈ 0.3
(3)	2500	≈ 10	≈ 0.1

Of these combinations, the second appears most reasonable, inasmuch as the first requires an emissivity near unity, which is unlikely, and the third

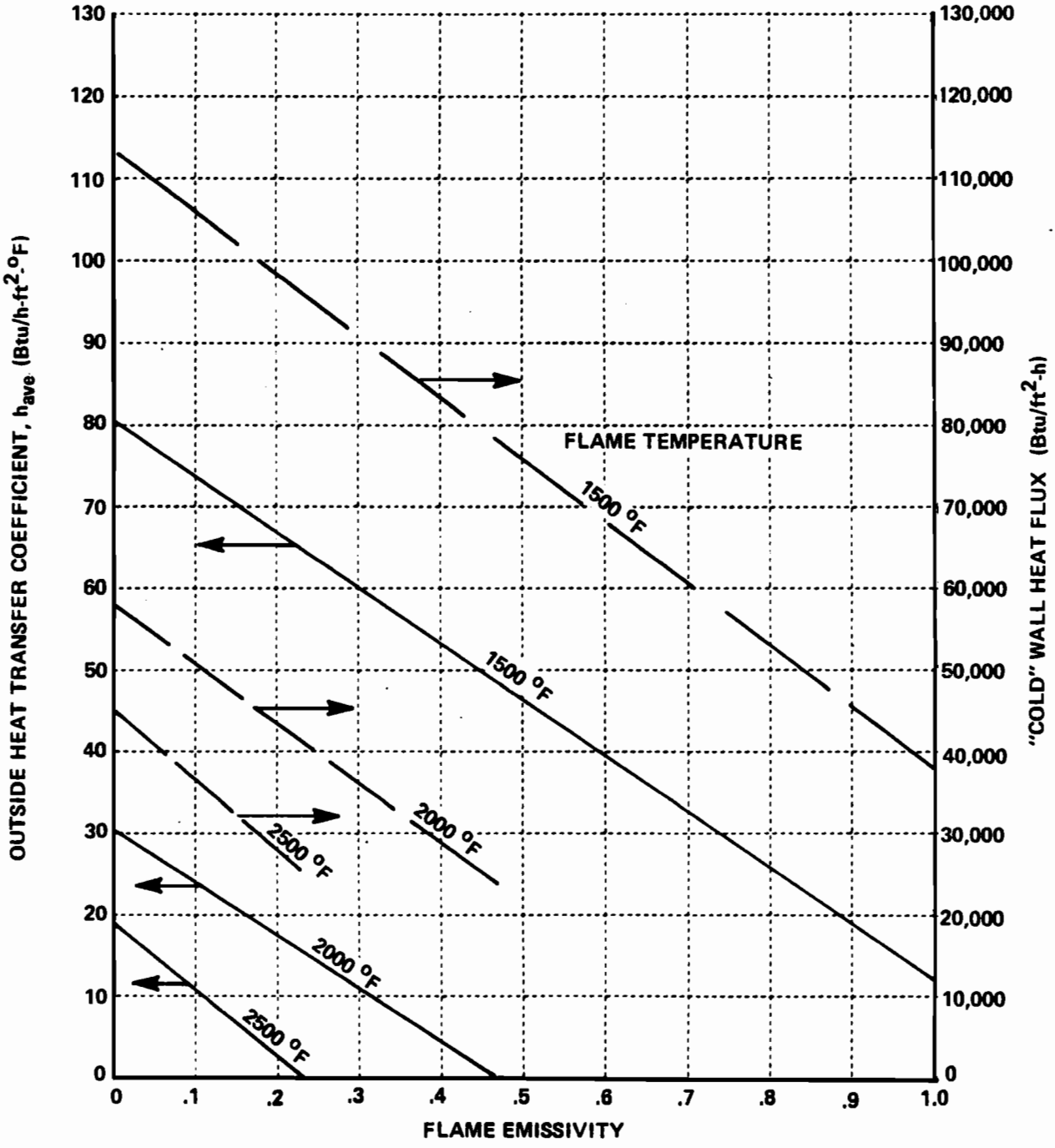

Figure 11 HEAT TRANSFER COEFFICIENT AND HEAT FLUX AS A FUNCTION OF FLAME EMISSIVITY FOR THREE FLAME TEMPERATURES

requires an average flame temperature at the limit of measured values. It must be noted, however, that more severe conditions than given by the second combination are possible, if maximum shell temperatures in excess of 1200°F have actually been produced. Use of the second combination is not particularly conservative to the side of safety and should be considered a near minimum requirement.

5.5 FLOW OF PROPANE THROUGH SAFETY-RELIEF VALVE ON OVERTURNED TANK CAR

For the case of an overturned tank car, the space immediately surrounding the safety-relief valve inlet may contain liquid. If the valve opens because of excessive internal tank pressure, the liquid will be forced through the valve. The flow path through the valve includes one or more orifices. A valve in the open position is shown in Figure 12. The particular valve shown is basically the configuration of an A-3480 model made by Midland Manufacturing Company, Skokie, Illinois. The Midland valve has been flow rated by others and by these data appear to fully meet all the requirements of the existing specifications for flow capacity of propane, given the characteristic shell area of a 33,500-gallon 112A340W car. Three area reductions which, in effect are orifices, are represented by the dashed lines numbered 2, 4, and 6.

As the liquid flows through the valve, the pressure subsides and the liquid begins to vaporize. Each orifice must then be large enough to pass the liquid-vapor mixture. In this study, the flow through a safety-relief valve has been calculated for the sizes of the valve shown in Figure 12. The initial calculations are based on the premise that flow at constant entropy exists within each individual orifice to the orifice exit and that the stagnation enthalpy remains constant throughout the valve. Nonisentropic flow tends to reduce the amount of flow, and some calculations have also been made based on an assumed deviation from isentropic flow.

The results indicate that the maximum flow through the valve shown in Figure 12 is almost independent of the tank pressure for all pressures

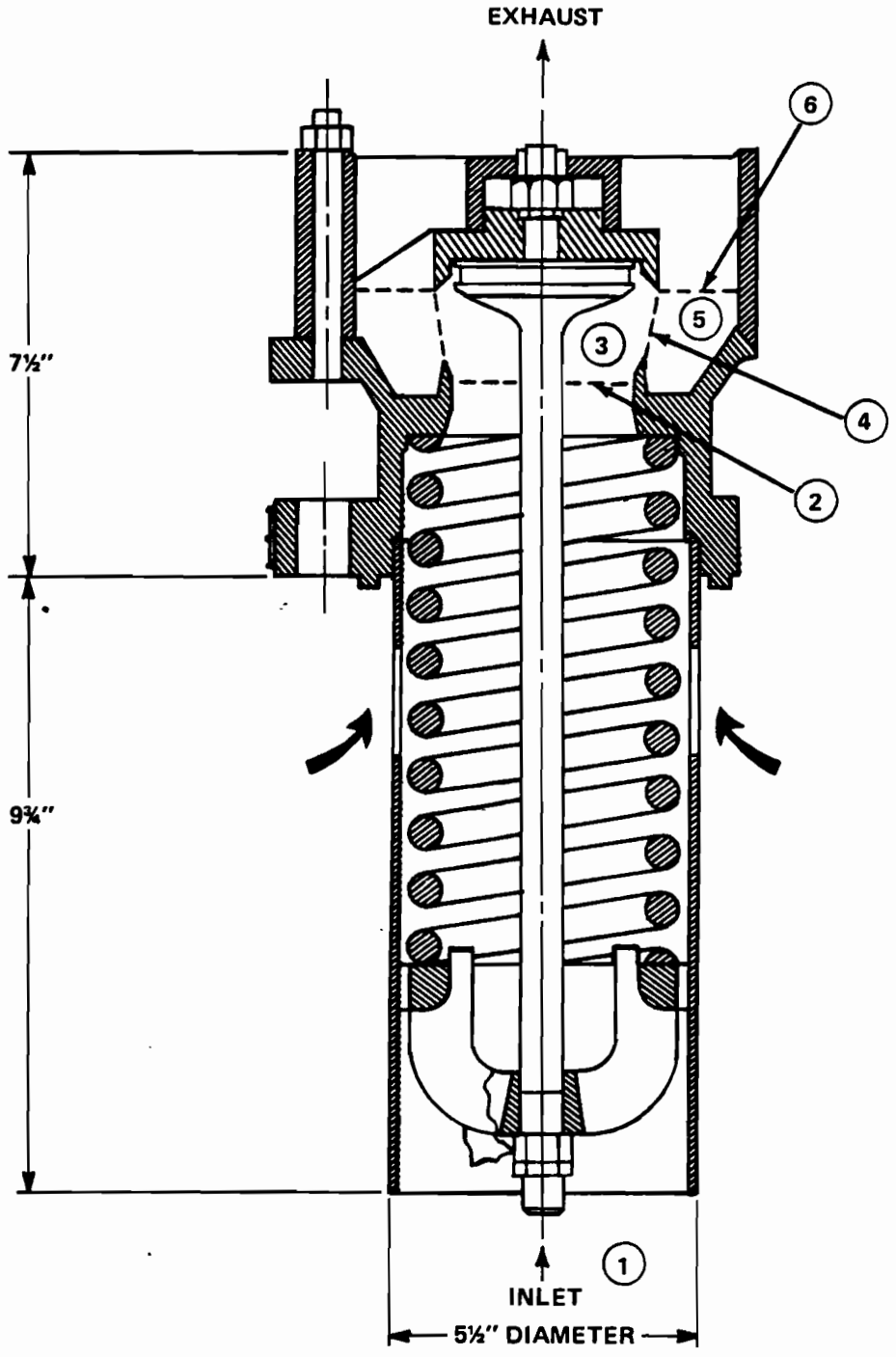

Figure 12 SAFETY RELIEF VALVE IN OPEN POSITION

above the valve full-opening pressure. Furthermore, the flow is limited by the orifice numbered 2 in Figure 12. The calculated maximum liquid flow as a function of tank pressure is shown in Figure 13.

The calculation procedure for obtaining the flow follows. The energy equation per pound of flow may be written as

$$\frac{V_1^2}{2g} + h_1 = \frac{V_2^2}{2g} + h_2 \tag{37}$$

where V is velocity, h enthalpy, and g the acceleration due to gravity. The subscripts 1 and 2 refer to locations given on Figure 12. In the tank (Position 1), the velocity is zero. Therefore, Equation (37) can be rewritten as

$$V_2 = \sqrt{2g(h_1 - h_2)} \tag{38}$$

The mass flow rate $\dot{M}$ is given by

$$\dot{M} = C_L A_2 \frac{V_2}{v_2} \tag{39}$$

where C_L is the flow coefficient for the orifice, A_2 the cross section at 2 and v_2 the specific volume at 2. Combining Equations (38) and (39), the result is

$$\frac{\dot{M}}{C_L A_2} = \frac{\sqrt{2g(h_1 - h_2)}}{v_2} \tag{40}$$

The enthalpy at Position 1, h_1, is the saturated liquid enthalpy which can be found in tables of the thermodynamic properties of saturated propane [27]. Therefore, to determine $\dot{M}/C_L A$, h_2 and v_2 are required.

To calculate h_2 and v_2 for isentropic flow, the following procedure is followed. At a given tank pressure, the saturated liquid entropy s is

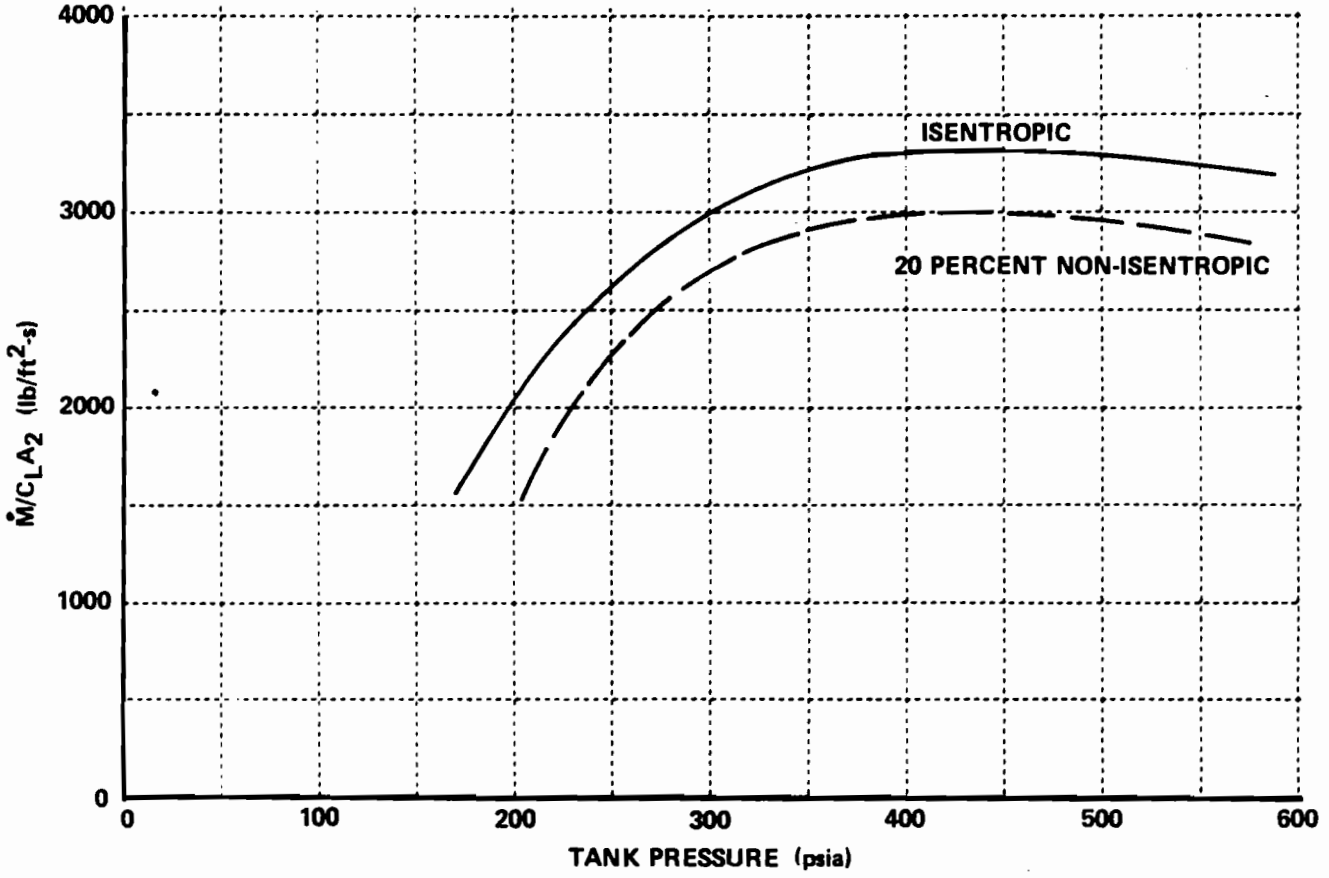

Figure 13 MAXIMUM FLOW OF LIQUID PROPANE THROUGH AN ORIFICE

known. This entropy remains constant throughout the flow through any one orifice. At an assumed pressure at the orifice, the fraction, x, of liquid to total liquid-vapor (i.e., quality), can be determined from

$$x = \frac{s - s_f}{s_g - s_f} \tag{41}$$

where the subscripts f and g designate saturated liquid and vapor at the orifice pressure, respectively. The enthalpy and specific volume can then be determined from the relations

$$h_2 = x(h_g - h_f) + h_f \tag{42}$$

$$v_2 = x(v_g - v_f) + v_f \tag{43}$$

Substituting Equations (42) and (43) in Equation (40), the isentropic flow rate can be obtained. By assuming different values of the orifice pressure, a plot of $\dot{M}/C_L A_2$ vs orifice pressure can be calculated. A series of these curves for different values of tank pressure will give the maximum flow relation shown in Figure 13.

To determine whether or not Orifice 2 (Figure 12) limits the flow instead of Orifice 4 or Orifice 6, the flow rates were calculated in a manner similar to that given above, with orifice entrance properties such that the enthalpy is the same for each orifice, but the entropy increases between orifices. When this calculation was made, it was found that the flow was limited by orifice area No. 2, not by No. 4 or 6.

The 20 percent nonisentropic flow condition shown in Figure 13 was calculated by assuming that the entropy increased 20 percent of the maximum that it could have in any adiabatic flow.

In general, relatively large changes in entropy would be expected in the valve, as well as significant loss of flow energy due to the formation

of liquid droplets and momentum exchange. For this reason, the flow rates indicated above are given in terms of a flow coefficient C_L, which could possibly be considerably below unity.

5.6 RELIEF-VALVE SIZING FOR PROPANE TANK CARS

Previous safety-valve sizing for propane tank cars has been based on the maintenance of some constant internal pressure within the car during application of external heat after the relief valve has opened. Valve size has heretofore been established on the basis of vapor exhaust through the valve. In the case of tank cars involved in a derailment, it is likely that some cars may be overturned and, therefore, need to release liquid to maintain integrity. This is considered in this section.

Assume that the set pressure of the valve has been reached due to external heat, $\dot{Q}$ Btu/s, and the valve is fully opened. At the set pressure, P_o, the saturation temperature of the liquid is T_o, and the latent heat is L_o. The specific volume of the liquid is v_{fo}, and that of the gas is v_{go}. If constant pressure is to be maintained, the rate of generation of vapor is $\dot{Q}/L_o$ lb/s, and the volumetric change in vapor space is $v_{fo}\,\dot{Q}/L_o$. The change in mass of vapor in the tank is

$$\frac{V_{fo}}{V_{go}} \times \frac{\dot{Q}}{L_o} \quad \text{lb/sec} \tag{44}$$

The vapor which must be removed through the valve to maintain constant pressure is

$$\dot{M}_v = \left(\frac{\dot{Q}}{L_o} - \frac{\dot{Q}}{L_o}\frac{V_{fo}}{V_{go}}\right) = \frac{\dot{Q}}{L_o}\left(1 - \frac{V_{fo}}{V_{go}}\right)^{*} \quad \text{Lbs/sec} \tag{45}$$

*For many cases $V_{go} \gg V_{fo}$ $\therefore$ $\dot{M}_v \cong \frac{\dot{Q}}{L_o}$.

If the liquid is to be removed at constant pressure conditions, the valve must pass sufficient liquid to increase the free volume in the tank. Again, the amount of liquid vaporized must be $\dot{Q}/L_o$ lb/s to maintain pressure. Since no internal vapor escapes, this vapor must occupy the change in free space. The free space is increased by $v_{fo}\,\dot{Q}/L_o$ due to internal liquid loss by vaporization, and additional liquid loss through the open valve further increases the free space by $\dot{M}_L\, v_{fo}$. Thus, the rate of change of free space is

$$\frac{dV}{d\tau} = V_{fo}\frac{\dot{Q}}{L_o} + \dot{M}_L V_{fo} \tag{46}$$

This is filled with vapor at specific volume v_{go}. Thus, the change in mass of the vapor in the free space is

$$\frac{1}{V_{go}}\frac{dV}{d\tau} = \frac{V_{fo}}{V_{go}}\frac{\dot{Q}}{L_o} + \dot{M}_L\frac{V_{fo}}{V_{go}} \tag{47}$$

A mass balance on the vapor then gives

$$\frac{\dot{Q}}{L_o} = \frac{V_{fo}}{V_{go}}\frac{\dot{Q}}{L_o} + \dot{M}_L\frac{V_{fo}}{V_{go}} \tag{48}$$

Hence,

$$\dot{M}_L = \frac{\frac{\dot{Q}}{L_o}\left(1 - \frac{V_{fo}}{V_{go}}\right)}{V_{fo}/V_{go}} \tag{49}$$

or

$$\dot{M}_L = \frac{\dot{Q}}{L_o}\left(\frac{V_{go}}{V_{fo}} - 1\right) \text{ lb/s} \qquad \text{LIQUID} \tag{50}$$

Comparing this to the required mass of vapor at the same heat input conditions, one gets

$$\frac{\dot{M}_L}{\dot{M}_V} = \frac{V_{go}}{V_{fo}} \tag{51}$$

below the critical point $v_g >> v_f$. Therefore, considerably more mass of liquid must be exhausted than vapor, if pressure is to be maintained constant. A given valve can possibly exhaust more mass of liquid per unit time than vapor at the same pressure, but the question is: Can it pass enough? It has been shown in an earlier section that the maximum liquid flow through a given valve is

$$\dot{M}_{L_{max}} = 3000\, C_L A \qquad \text{(Propane)} \qquad (52)$$

(295 psia Setpoint)

The maximum vapor loss as given by the AAR Specification* is dependent on pressure and is given approximately by

$$\dot{M}_{V_{max}} = \frac{C_v A V_c}{v_c} \qquad (53)$$

where v_c is the specific volume of the vapor at a pressure

$$P = P_o \left(\frac{k+1}{2}\right)^{\frac{k}{1-k}} \qquad (54)$$

and temperature

$$T = T_o \left(\frac{k+1}{2}\right)^{-1} \qquad (55)$$

and V_c is the acoustic velocity of the vapor at temperature T. The ratio of specific heats, k, for the vapor may be taken as 1.065 over the range $130 < P < 300$. Hence

$$T = T_o \left(\frac{2}{2.065}\right) = 0.97\, T_o \qquad (56)$$

$$P = P_o \left(1.032\right)^{-16.7} = 0.59\, P_o \qquad (57)$$

*The flow should actually be calculated from isentropic flow relations similar to those presented in the previous section for liquid flow. The difference is small, however, for the specific example.

The acoustic velocity is given by

$$V_c = \sqrt{k_g RT} \tag{58}$$

The gas constant, R, for the vapor is about 30 ft-lb/lb-°R and if one considers the valve set pressure to be P_o = 308 psia, T_o = 140°F = 600°R,

$$\dot{M}_{V\,max} = \frac{C_v A}{0.665}\sqrt{(1.065)(32.2)(30)(0.97)(600)} = 1160\, C_v A \tag{59}$$

Comparing this with $\dot{M}_{L_{max}}$, one gets

$$\frac{\dot{M}_{L\,max.}}{\dot{M}_{V\,max.}} = \frac{3000\, C_L}{1160\, C_V} = 2.58\frac{C_L}{C_V} \quad . \tag{60}$$

But at these same conditions, from Equation (51),

$$\frac{\dot{M}_L}{M_V} = \frac{0.32}{0.037} = 8.65 \tag{61}$$

must be capable of being passed. Usually, $C_L < C_V$. Hence, if the relief valve is designed to only pass the correct amount of vapor according to the specifications, it will not pass the required amount of liquid flow. For the above set pressure, the valve would be underdesigned for liquid flow by a factor of at least 8.65/2.58 = 3.35. For this reason, relief-valve sizing considerations must be based upon liquid relief as well as vapor relief to determine the controlling case. Note that the above calculations were based on propane and the results are specific for that case. The analysis was general, however, and can be utilized for other liquefied compressed gases utilizing the appropriate values of the thermodynamic properties.

If the valve is underdesigned for liquid flow as indicated above, there will be a general pressure increase. The pressure rise rate within the tank is functionally dependent upon heat input and liquid discharge rate. At any tank condition, the change in vapor space (when the liquid is being discharged) is given by

$$\frac{dV^{*}_{SPACE}}{dt} = \dot{M}_L v_f + \dot{M}_v u_f \tag{62}$$

rate of change of vapor space volume = volume of liquid discharged + liquid evaporation

This space is continuously filled with vapor at specific volume v_g. Hence,

$$\dot{M}_v v_g = \dot{M}_L v_f + \dot{M}_v v_f = \frac{dV_{space}}{d\tau} \tag{63}$$

$$\dot{M}_v = \dot{M}_L \left(\frac{v_f}{v_g - v_f} \right) \tag{64}$$

This vaporization absorbs heat

$$Q_v = \dot{M}_v L = \dot{M}_L L \left(\frac{v_f}{v_g - v_f} \right) \tag{65}$$

The heat input to the tank is Q_{IN}. The excess of heat input over that taken in vaporization is absorbed by the liquid-vapor mix in the tank, thus increasing the tank pressure. Along the propane-liquid saturation line, near 300 psia, pressure increase per unit internal energy increase is about 5 psi/$\frac{\text{Btu}}{\text{lb}}$. Hence,

$$\frac{dP}{du} = 5 \text{ psi} / \frac{\text{Btu}}{\text{lb}} \tag{66}$$

but,

$$Q_{IN} - Q_v \approx M_{TOT} \frac{du}{dt} \tag{67}$$

Then from Equations (65), (66) and (67),

*Neglecting the small changes in volume due to liquid expansion.

$$Q_{IN} \approx \frac{M_{TOT}}{5} \frac{dP}{dt} + \dot{M}_L \angle \left(\frac{V_f}{V_g - V_f} \right) \tag{68}$$

Equation (68) indicates the required amount of liquid flow to limit the pressure rise rate at any total heat input Q_{IN}. Because the compressed liquid state has not been considered in the development of Equation (68), values of $\dot{M}_L$ must be taken to be greater than needed to prevent failures due to compressed liquid (see Section 5.7). For any liquid flow rate, $\dot{M}_L$, there is a duration for complete loss of liquid. In this duration, one may set a limit on the allowable pressure rise above the safety valve setting. This then determine the allowable heat load, Q_{IN}. For each pressure limit and liquid discharge rate, there is a specific maximum allowable heat load.

The discharge duration is approximately

$$t_o = M_{TOT} / \dot{M}_L \tag{69}$$

If the maximum allowable pressure is P_{MAX} (psi) and the set pressure of the valve is P_o (psi), we get

$$\frac{dP}{dt} \approx \frac{(P_{MAX} - P_o)}{t_o} = \frac{(P_{MAX} - P_o)}{M_{TOT}} \dot{M}_L \tag{70}$$

From Equation (68)

$$Q_{IN})_{ALLOWABLE} \triangleq \left[\frac{(P_{MAX} - P_o)}{5} + \frac{\angle (V_f)}{(V_g - V_f)} \right] \dot{M}_L \tag{71}$$

In terms of heat flux over the entire car, Equation (71) becomes

$$q_{IN})_{ALLOWABLE} \triangleq \left[\frac{(P_{MAX} - P_o)}{5} + \frac{\angle V_f}{(V_g - V_f)} \right] \frac{\dot{M}_L}{A} \tag{72}$$

where

q_{IN} = the heat flux

A = the car area.

Figure 14 illustrates values of q_{IN}/allowable as a function of $\dot{M}_L$ for various maximum allowable pressures P_{MAX}. For any allowable maximum pressure, the allowable heat flux is directly proportional to the liquid discharge rates. As the allowable maximum pressure is increased, the required discharge flow is reduced for any heat flux. If one takes the heat flux as given in the AAR specification $q = 34{,}500\ A^{0.82}$ or about 8000 Btu/ft^2-h on the entire car and an allowable accumulation pressure of 350 psi, the safety valve must be capable of discharging at least 170 lb/s of liquid propane for a 2000 ft^2 car shell area. Valve design based on Equation (72) should result in minimum flow requirements.

5.7 TANK CAR FAILURE DUE TO COMPRESSED LIQUID EXPANSION

A failure mode that must be considered is the possibility that the expansion of the liquid due to heating will be greater than the safety-valve capacity. The specific volume of liquid propane increases more rapidly per unit increase in heat input than for many other liquids. If the liquid volume increases more rapidly than the valve can accommodate, the liquid will be compressed and the pressure in the tank will rapidly increase, and tank failure may result.

The condition of liquid propane filling the tank as a result of thermal expansion and vapor condensation before the car fails due to overheating of the metal over the vapor space is a very real possibility, as can be seen from the following calculation. Consider that the tank car is uniformly heated at a constant rate, q, and that there is no conduction from the metal at the top of the vapor space. The time for the metal to reach a failure temperature, τ_f, is given by

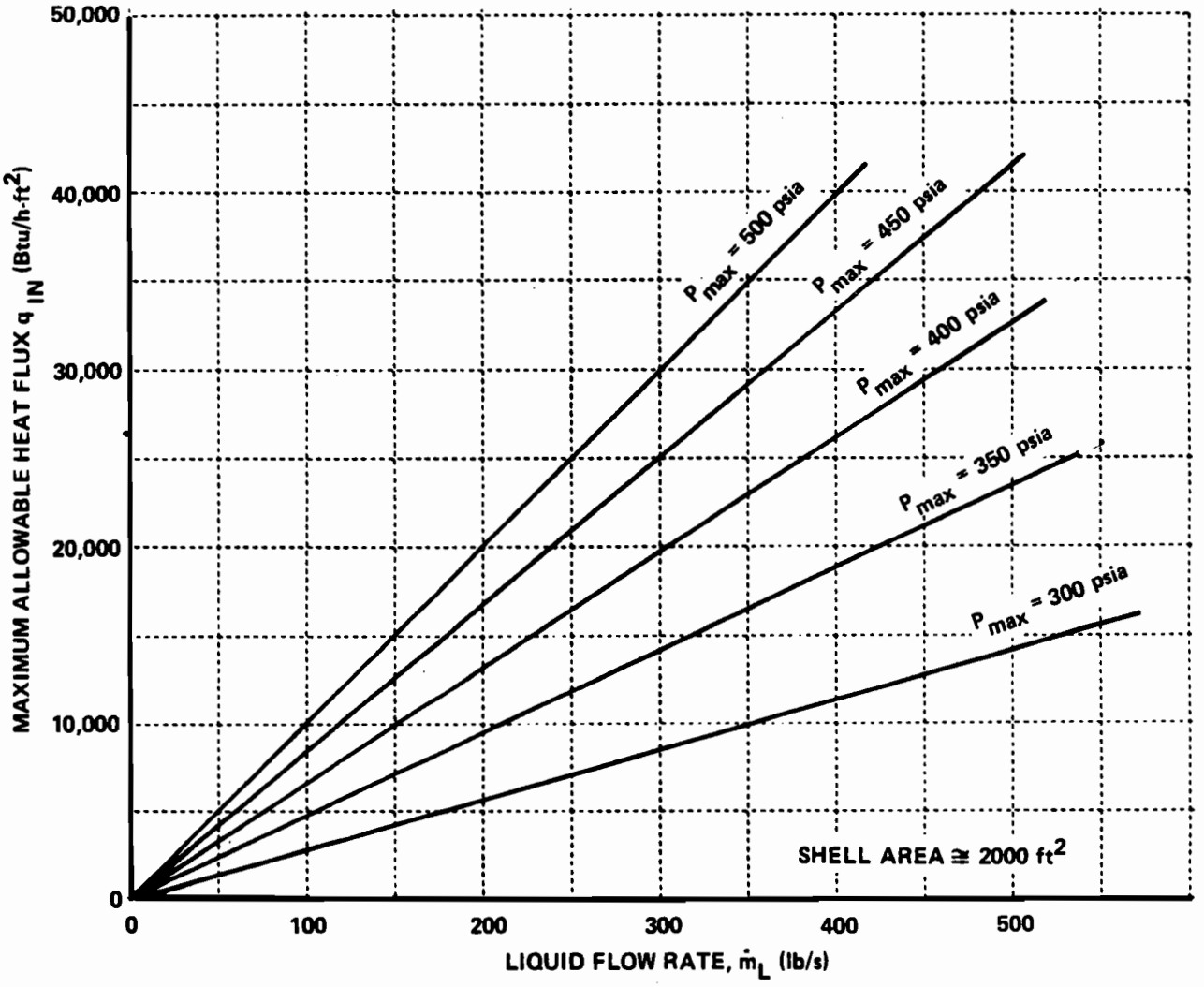

Figure 14 MAXIMUM ALLOWABLE HEAT FLUX AS A FUNCTION OF LIQUID FLOW RATE (PROPANE)

$$t_1 = \frac{c\rho\delta(T_F - T_a)}{q} \tag{73}$$

where c, ρ and δ are the specific heat, density, and thickness of the metal, respectively, T_F the failure temperature of the metal, and T_a the initial tank temperature.

The time it takes for the tank to become shell full with liquid after the start of heating is

$$t_2 = \frac{(u_{fF} - u_{fa}) M_{TOT}}{qA} \tag{74}$$

where u_{fF} is the liquid internal energy at the time the tank is full, u_{fa} the liquid internal energy at the initial tank temperature, M_{TOT} the total mass of liquid in the tank, and A the outside area of the tank.

For the tank to fill with liquid before the vapor space fails, t_1 must be greater than t_2. That is

$$c\rho\delta(T_F - T_a) > \frac{(u_{fF} - u_{fa}) M_{TOT}}{A} \tag{75}$$

Assuming a minimum failure temperature of 1200°F and a tank car starting at 60°F and being full at 115°F (corresponding to the maximum amount of propane that can be carried),* the left side of Equation (75) is about 3550 Btu/ft^2, and the right side is about 2000 Btu/ft^2. Hence, there is little chance that the tank will fail prior to becoming completely full of liquid, if the tank is initially filled to maximum capacity according to specifications. This estimate is conservative, because the heat flux to the metal is actually not constant but rather falls off as the metal temperature increases, whereas the flux to the liquid changes only slightly. Of course, if the tank is loaded less than the maximum, it is possible that the tank would fail before the liquid fills the tank.

*As authorized per DOT regulations for summer conditions.

The possibility of failure due to compressed liquid is calculated in the following manner. The rate of change of internal energy, du_f/dt, in the tank car at the time the safety valve opens is given by

$$\frac{du_f}{dt} = \frac{qA}{M_{TOT}} \tag{76}$$

where q is the input heat flux to the tank car, A the outside area of the tank car, and M_{TOT} the initial mass of liquid in the tank car.

The specific volume of liquid in the tank is given by

$$v_f = \frac{V}{M_{TOT} - \dot{M}\tau} \tag{77}$$

where V is the volume of the tank car, and $\dot{M}$ the mass flow rate out of the safety valve. Differentiation of Equation (77) results in

$$\frac{dv_f}{dt} = \frac{V\dot{M}}{(M_{TOT} - \dot{M}\tau)^2} \tag{78}$$

When Equations (76) and (78) are combined at $t = 0$, the result is

$$\frac{dv_f}{du_f} = \frac{V\dot{M}}{qAM_{TOT}} \tag{79}$$

Using values for the 112A340W tank car in Equation (79), the relation is

$$\frac{dv_f}{du_f} = 6.84 \times 10^{-2} \frac{\dot{M}}{q} \tag{80}$$

for dv_f/du_f in ft^3/Btu, $\dot{M}$ in lb/s, and q in Btu/h-ft^2.

For the tank car to be safe from failure by more rapid expansion of the liquid than the valve can accommodate, dv_f/du_f given by Equation (80) must be greater than dv_f/du_f for the liquid as given by Figure 15 at the opening pressure of the valve. From Figure 15, this value is $dv_f/du_f = 1.57 \times 10^{-4}$ ft^3/Btu. Therefore, the safe condition is represented by

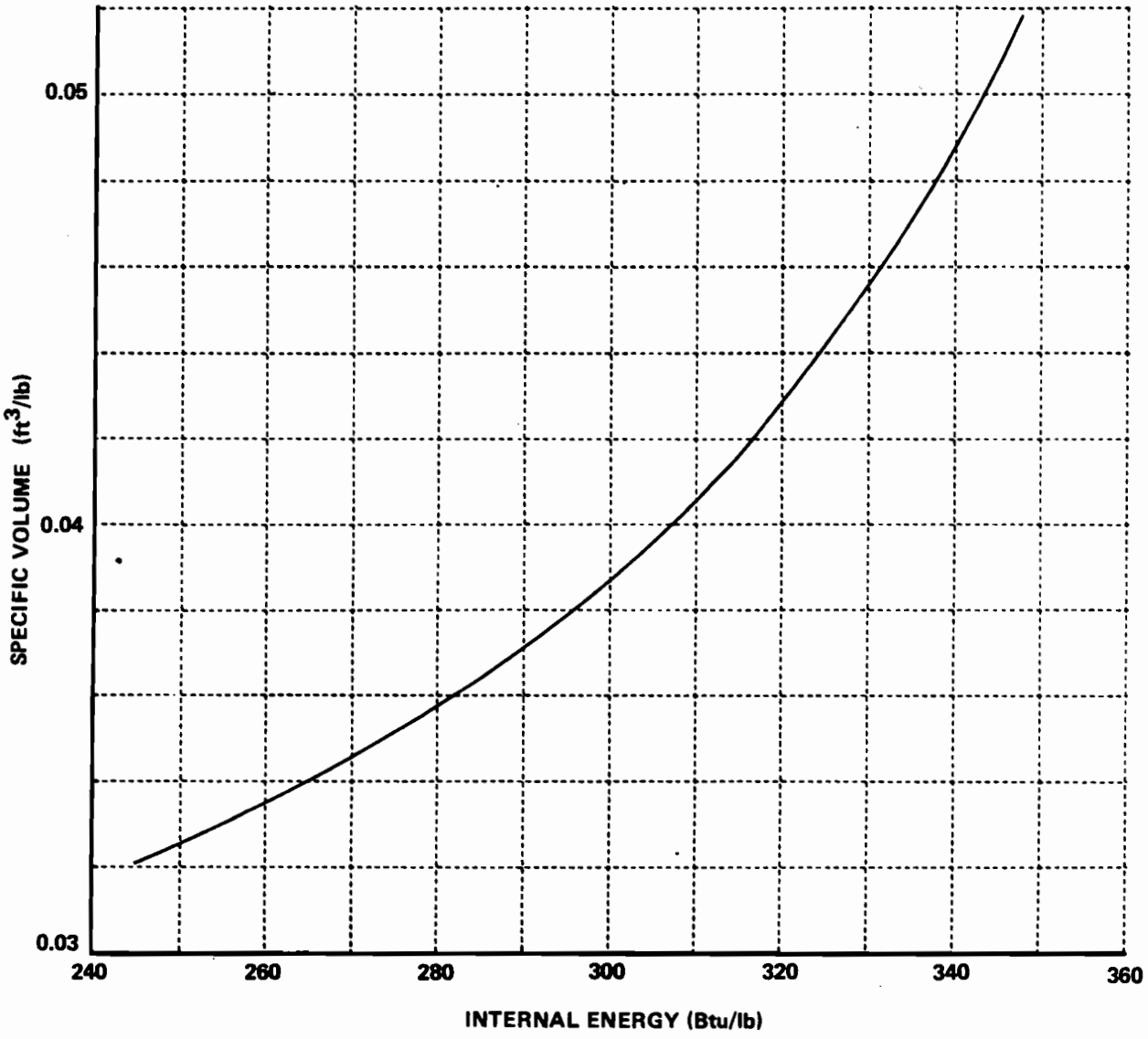

Figure 15 SPECIFIC VOLUME AS A FUNCTION OF INTERNAL ENERGY FOR SATURATED LIQUID PROPANE

$$6.84 \times 10^{-2} \frac{\dot{M}}{q} > 1.57 \times 10^{-4} \tag{81}$$

Solving for $\dot{M}/q$, Equation (81) reduces to

$$\frac{\dot{M}}{q} > 2.3 \times 10^{-3} \tag{82}$$

This relation is shown in Figure 16.

The maximum flow of liquid propane through the safety-relief valve is shown in Figure 13 at the set pressure of 295 psia* to be

$$\frac{\dot{M}}{C_L A_2} = 3000 \quad Lb/Ft^2 s \tag{83}$$

Combining Equations (82) and (83), there is obtained

$$C_L A_2 > 7.7 \times 10^{-7} q \quad Ft^2 \tag{89}$$

This relation is also shown on Figure 16. If the flow coefficient, C_L, for the valve shown in Figure 12 is 0.65 (and there have been experimental data indicating that it may be considerably lower), the value of $C_L A_2$ would be 0.037 ft^2 and the flow rate is 110 lb/s.

Once the tank is full of compressed liquid, the valve must be capable of passing the amount of flow required to remain in the safe region. If the maximum flow is 110 lb/s as given above, the tank would proceed from just becoming full of liquid to saturated liquid at the valve set pressure in about 70 seconds. This is the minimum time that the tank remains within the compressed liquid regime. If there is a flare-up in the fire during this time

*Note from Figure 13 that the result is changed only slightly for any given pressure above this value.

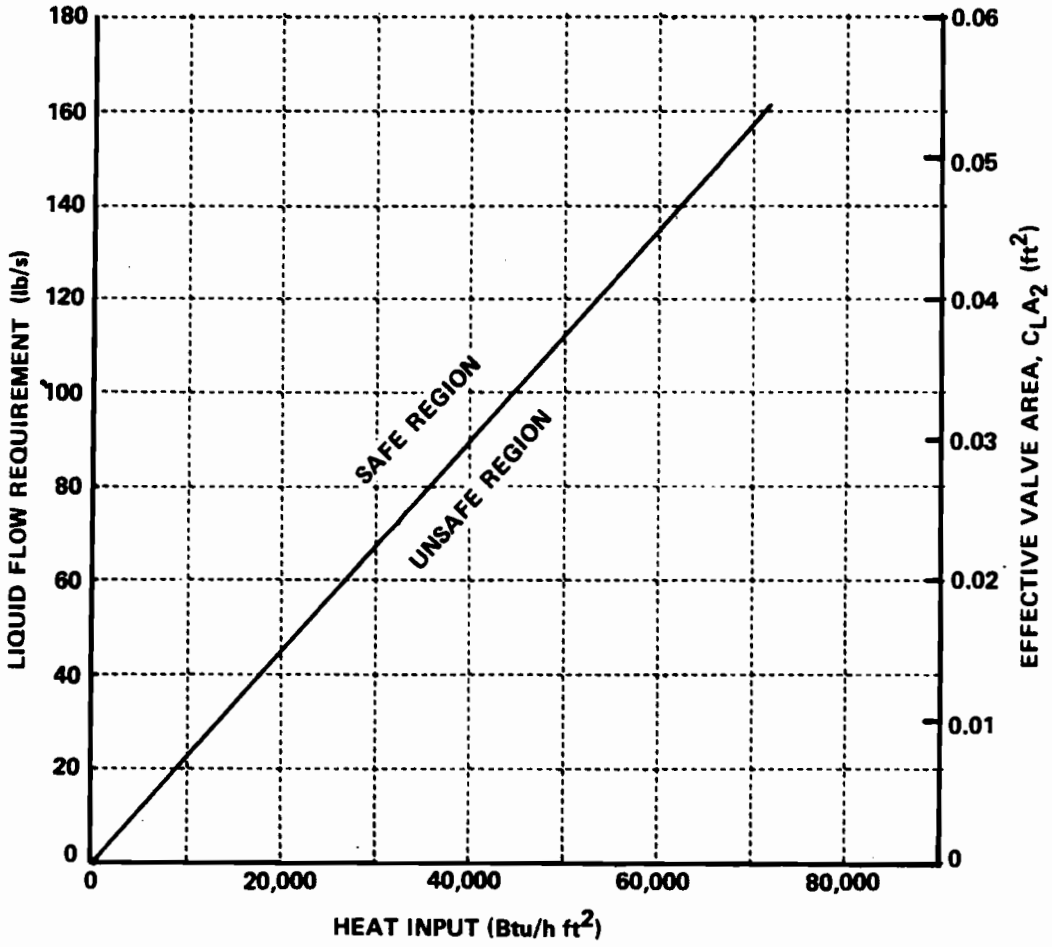

Figure 16 AMOUNT OF LIQUID FLOW REQUIRED TO PREVENT FAILURE DUE TO COMPRESSED LIQUID

such that the heat input is in the unsafe portion of Figure 16, the tank will fail very rapidly. The time to reach this type of failure is limited only by the compressibility of the liquid and the elastic deformation of the tank. The indication from Figure 10 is that for the heat fluxes that may be present in fires involving tank cars (35,000 Btu/h ft^2 or greater, particularly for short times), the present safety-relief valves may be within the unsafe region, depending on the valve flow coefficient.

5.8 RESULTS OF COMPUTER CALCULATIONS

Calculations were performed using the computer routine described in Appendix A. The purpose of the calculations was to determine the time-to-failure for tank cars containing propane subjected to various fire conditions. Fire conditions were specified in the calculations by parametric variation of convection coefficient, H_E, flame temperature, T_E, and flame emissivity, ε_0. In all cases computed, the entire axial length of the car was assumed to be subjected to fire conditions. The time-to-failure is defined as the time after start of heating when the maximum stress in the tank wall at any location becomes greater than the ultimate tensile stress at the prevailing temperatures of that location. Ultimate tensile stress versus temperature values for the tank shell material were taken from Reference 19.

Physically, in the computer, temperatures in the tank shell are calculated as a function of time as well as the temperature and pressure of the liquid and vapor within the tank. As heating proceeds, heat is absorbed chiefly by the liquid in contact with the shell and, because saturation conditions prevail, the pressure within the shell increases. At some time after the start of heating, the internal pressure exceeds the safety-valve setpoint and material (liquid or gas) is discharged. The rate of discharge for vapor or liquid was determined by the use of the relations of section 5.6. In calculations of flow, the valve is assumed to have opened fully at the set pressure. Calculations of internal pressures continue to be made on the basis of the excess or deficiency of input heat over that lost by discharges

of vapor or internal generation of vapor. Tank car shell temperatures continue to be calculated until a temperature is reached at which the corresponding ultimate tensile stress is below the tensile stress produced by internal pressure. At this time, the tank is considered to have failed.

The initial calculations were performed for the purpose of establishing the condition which is more severe, liquid or vapor discharge. It was found that times-to-failure for vapor flow were generally greater than those for liquid flow. This is shown in Figure 17. For this reason, subsequent calculations were made primarily for liquid flow.

Calculations of time-to-failure with liquid feed conditions were made for a number of fire source conditions, namely, convection coefficient, H_F = 5, 10, 20 Btu/h ft^2°F, flame temperature, T_F = 1500, 2000, 2500°F, flame emissivity, ε_0 = 0 to 1.0. In each instance, calculations were performed for the 112A340W tank car containing an amount of propane slightly less than sufficient to fill the car with liquid at the set pressure of the safety valve (295 psia). This filling condition was selected to avoid complications presented by calculations of compressed liquid states. It was felt that this simplification would not change the fundamental failure phenomenon where failure occurs above the safety-valve setpoint.

Time-to-failure as a function of this above fire source conditions and "cold" wall heat flux (Q_{cw}) are shown in Figure 18-20. For these calculations, the discharge coefficient and area of the valve were taken as 0.80 and 0.060 ft^2, respectively. The solutions will be accurate for any other product of discharge coefficient C and discharge area A equalling 0.048 ft^2. It is evident by inspection of the figures that failure time decreases rapidly with increasing flame emissivity at constant convection coefficient and flame temperature. The effect of flame emissivity is most pronounced at the lower flame temperatures. Taking the three most likely source conditions from Section 5.4, namely

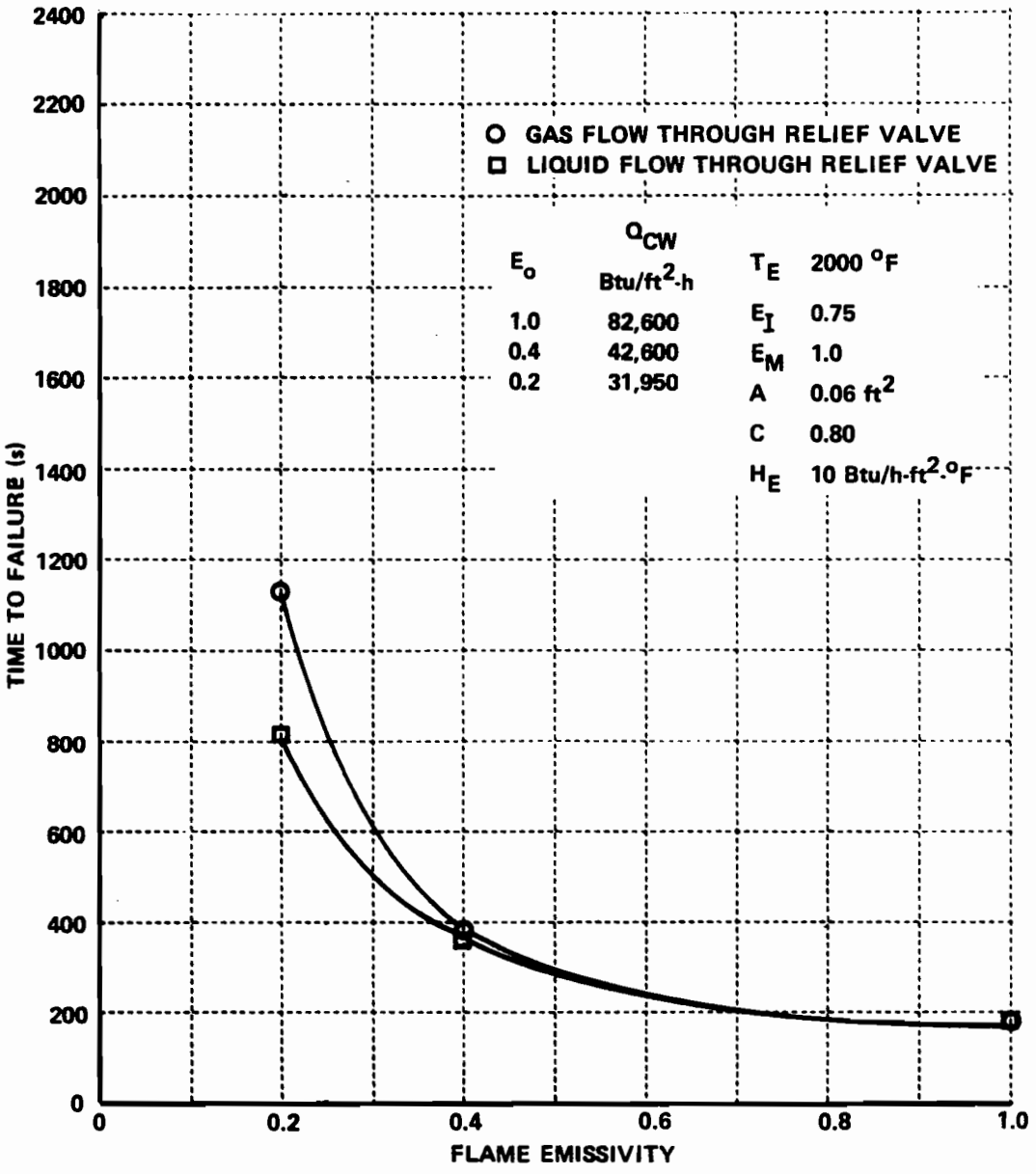

Figure 17 TIME TO FAILURE FOR GAS AND LIQUID FLOW THROUGH RELIEF VALVE, 112A340W CAR, PROPANE

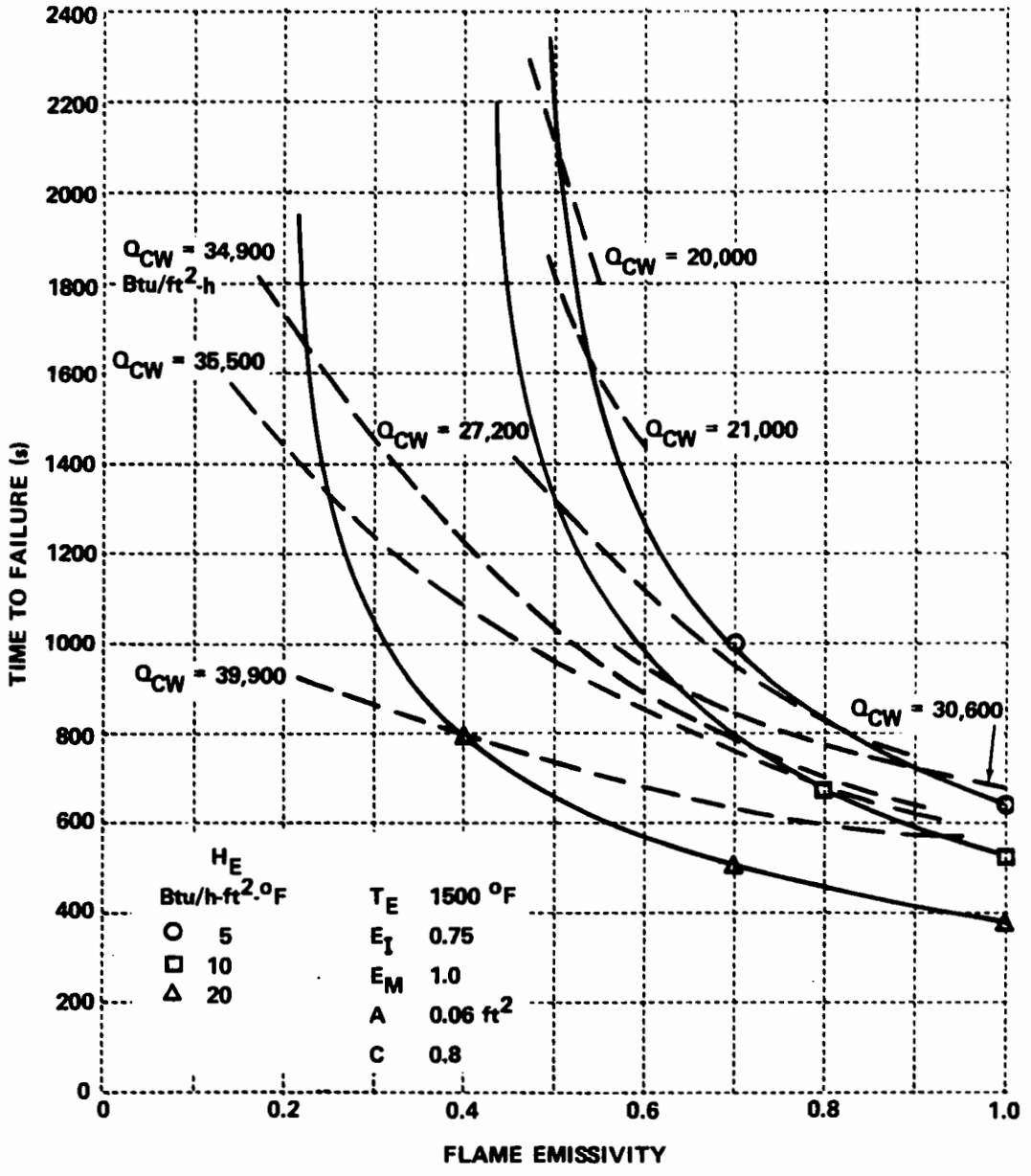

Figure 18 TIME TO FAILURE AS A FUNCTION OF COLD-WALL HEAT FLUX, EXTERNAL HEAT TRANSFER COEFFICIENT, AND FLAME EMISSIVITY (T_E = 1500 ^{o}F), 112A340W CAR, PROPANE

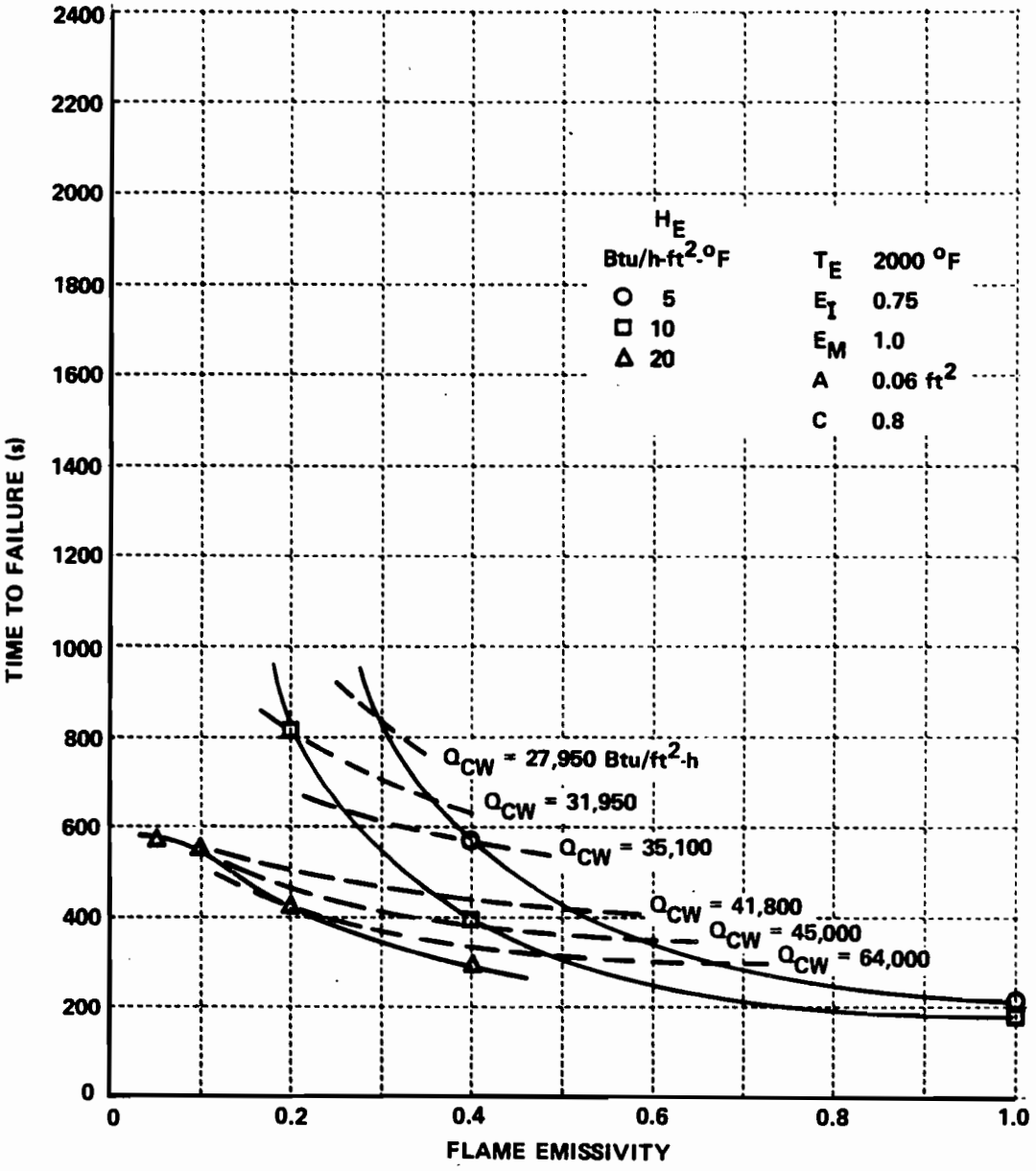

Figure 19 TIME TO FAILURE AS A FUNCTION OF COLD-WALL HEAT FLUX, EXTERNAL HEAT TRANSFER COEFFICIENT, AND FLAME EMISSIVITY (T_E = 2000 ^{o}F), 112A340W CAR, PROPANE

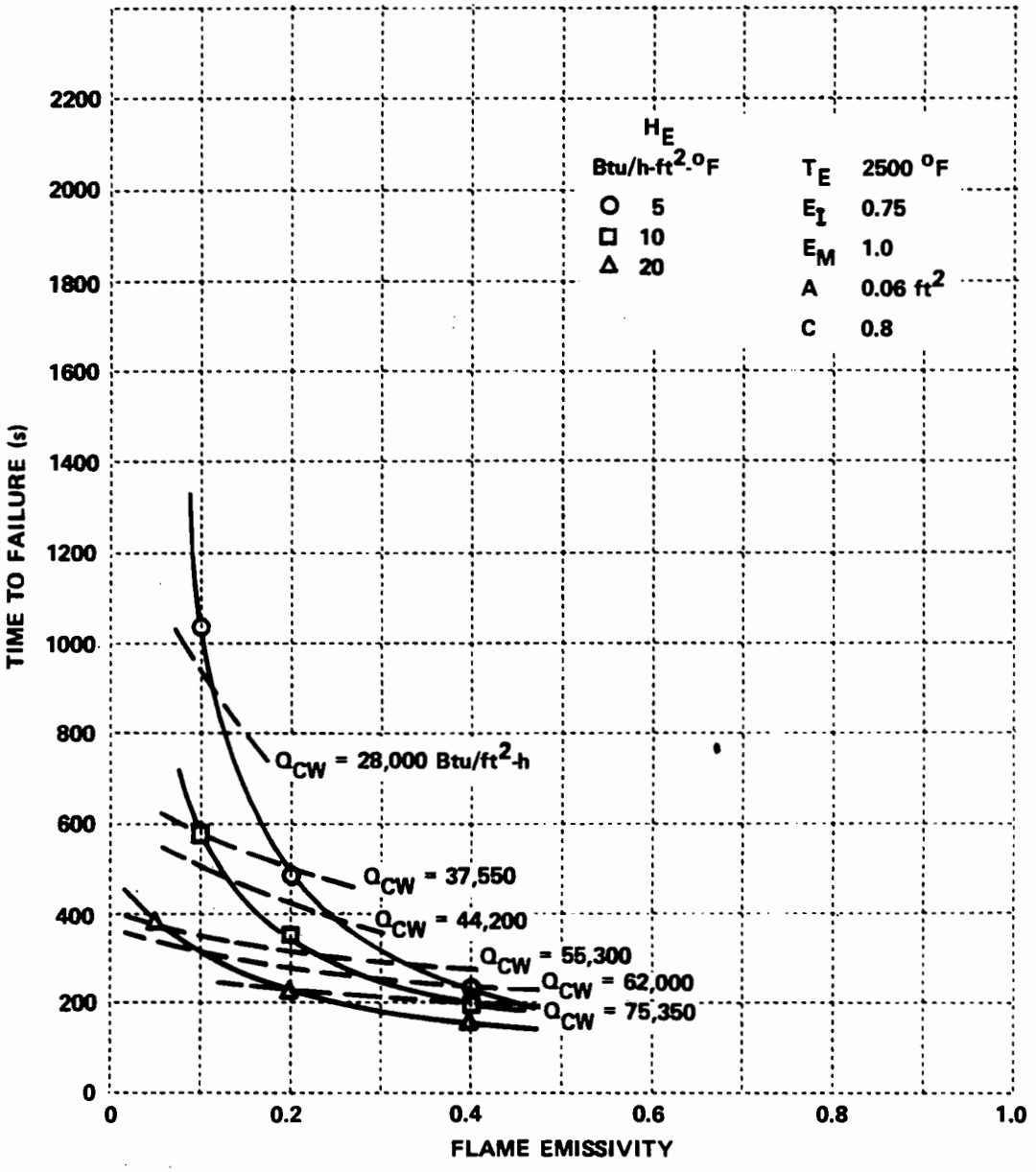

Figure 20 TIME TO FAILURE AS A FUNCTION OF COLD-WALL HEAT FLUX, EXTERNAL HEAT TRANSFER COEFFICIENT, AND FLAME EMISSIVITY (T_E = 2500 ^{o}F), 112A340W CAR, PROPANE

	T_E (°F)	H_E (Btu/h-ft^2 °F)	ε_O
1	1500	10	1.0
2	2000	10	0.3
3	2500	10	0.1

the corresponding failure times are 520, 550 and 575 seconds. Or the failure time is almost independent of the source condition selected. A plot of tank pressure rise rates at the valve setpoint versus cold-wall heat flux is shown in Figure 21 for all liquid cases computed. From this plot, it appears that pressure rise rates above 0.35 psi/s lead to tank failure.

The dashed line in Figure 21 is a plot of Equation (68) of Section 5.6, through which a design equation for the valve had been established. It appears that the equation predicts slightly higher pressure rise rates than indicated by the computer at any input flux, but the error is not great in the heat flux range of interest (0 to 40,000 Btu/ft^2 h). Hence, it is reasonable to utilize Equation (68) for valve sizing.

Several calculations were performed using the computer to establish a valve of sufficient capacity to prevent tank failure at an overall cold-wall heat flux of about 37,000 Btu/ft^2 h. The fire source conditions were flame temperature, T_E = 2000°F, flame emissivity, ε_o = 0.285 and convection coefficient, H_E = 10.0 Btu/h-ft^2°F. The discharge coefficient for the valve was taken as 0.65. Figure 22 illustrates the failure time (based on ultimate tensile stress) as a function of valve areas. As shown, the failure time approaches infinity at a valve area of approximately 0.2 ft^2 for liquid flow and approximately 0.14 for gas flow.

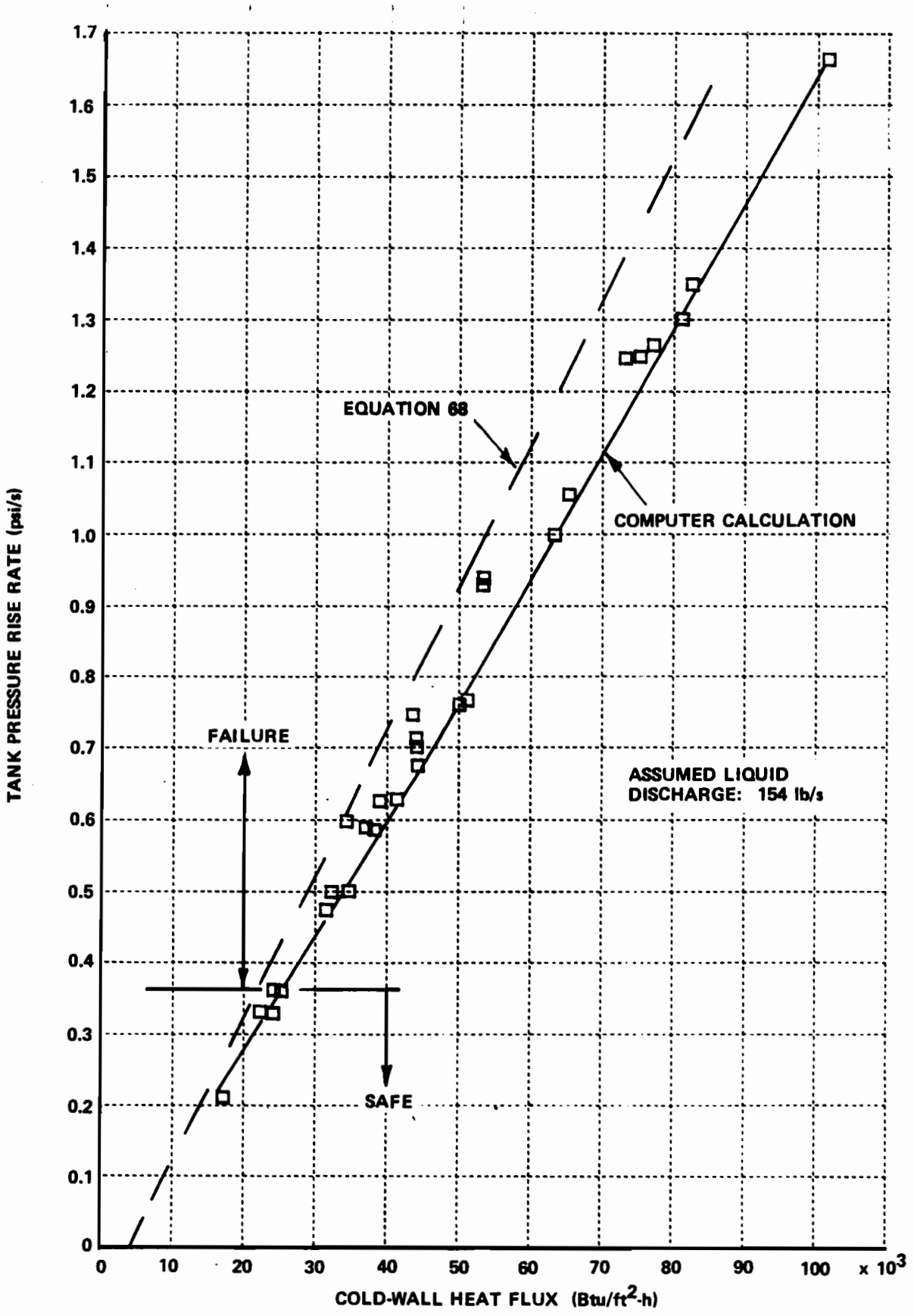

Figure 21 TANK PRESSURE RISE RATE AT THE RELIEF VALVE SETPOINT AS A FUNCTION OF COLD-WALL HEAT FLUX, 112A340W CAR - PROPANE

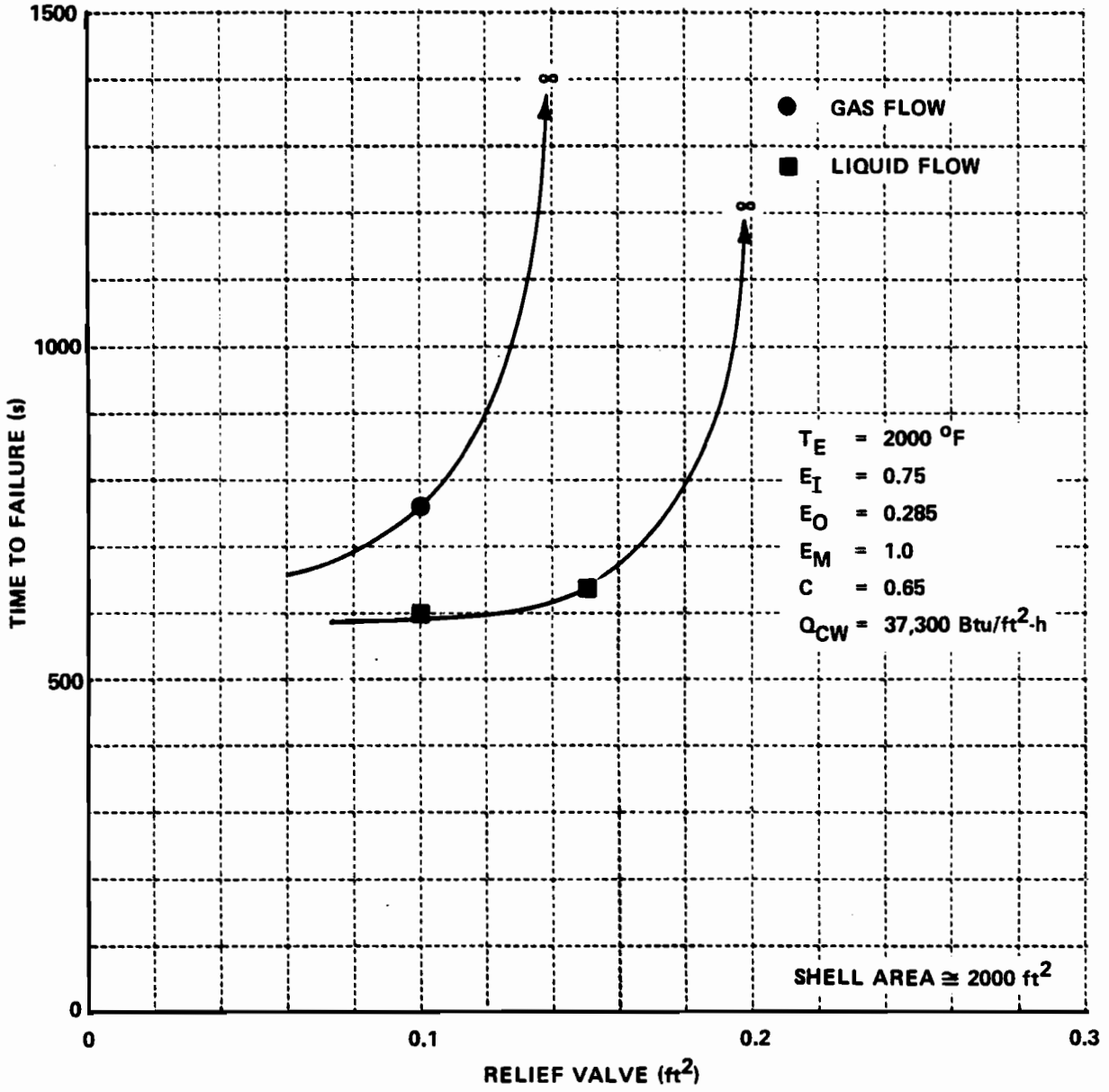

Figure 22 TIME TO FAILURE AS FUNCTION OF RELIEF VALVE AREA, 112A340W CAR - PROPANE

Calculations were performed using a discharge coefficient of 0.65. However, since discharge is dependent upon the product of the discharge coefficient and area, the required valve area for other assumed discharge coefficients is easily determined by ratio. Note that for the given conditions, valves currently in service (approximately 0.06 ft^2 area) would not afford adequate protection even with a discharge coefficient of unity.

Figure 23 shows the conditions within the tank and in the tank shell as a function of time for this "safe" valve area. The tank achieves a maximum pressure of approximately 360 psia with a corresponding topmost shell temperature of 890°F at the time all liquid is discharged. It must be noted, however, that the computations assume the valve to be fully open at the set pressure. Proportionality between amount of opening and tank pressure has not been considered analytically but it appears that any proportionality is undesirable and may lead to higher tank pressure.

Figure 24 shows the temperature distribution in the tank car shell for the same conditions given in Figure 23, The shell temperature over the vapor space rises relatively rapidly but as the liquid fills the tank, the shell temperature drops to essentially the liquid temperature. As time increases, the liquid level then drops and the top of the shell begins to rise in temperature again. It may be noted that the top of the car may reach a temperature which will result in paint discoloration before the car is filled with liquid and then the shell temperature drops to a much lower value. This indicates that observation of discoloration on a failed tank car after the fire has subsided does not necessarily indicate the shell temperature at the time of failure.

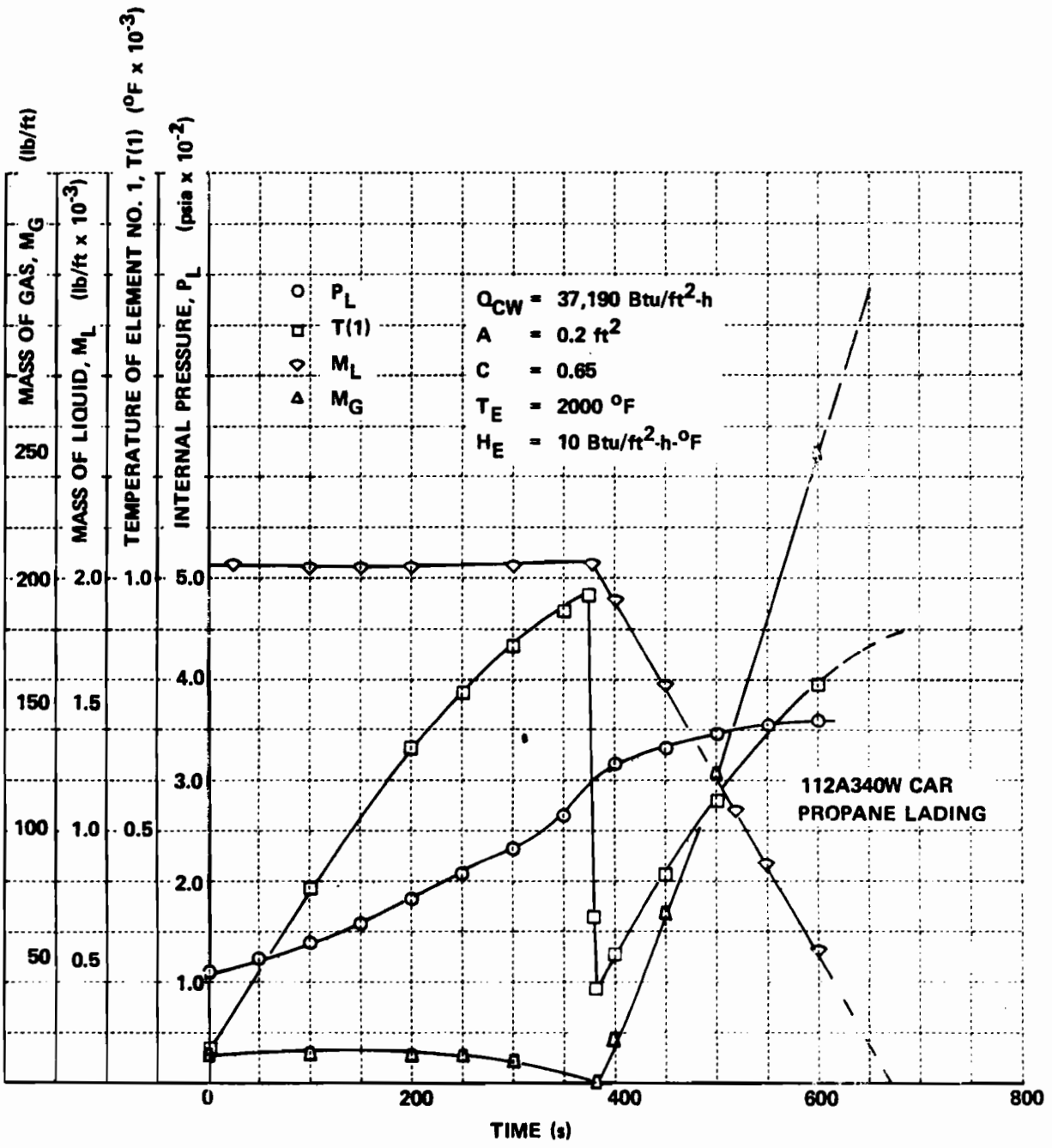

Figure 23 MASS OF GAS AND LIQUID, AND PRESSURE IN TANK AND TEMPERATURE OF TOP ELEMENT OF TANK FOR "SAFE" VALVE AREA

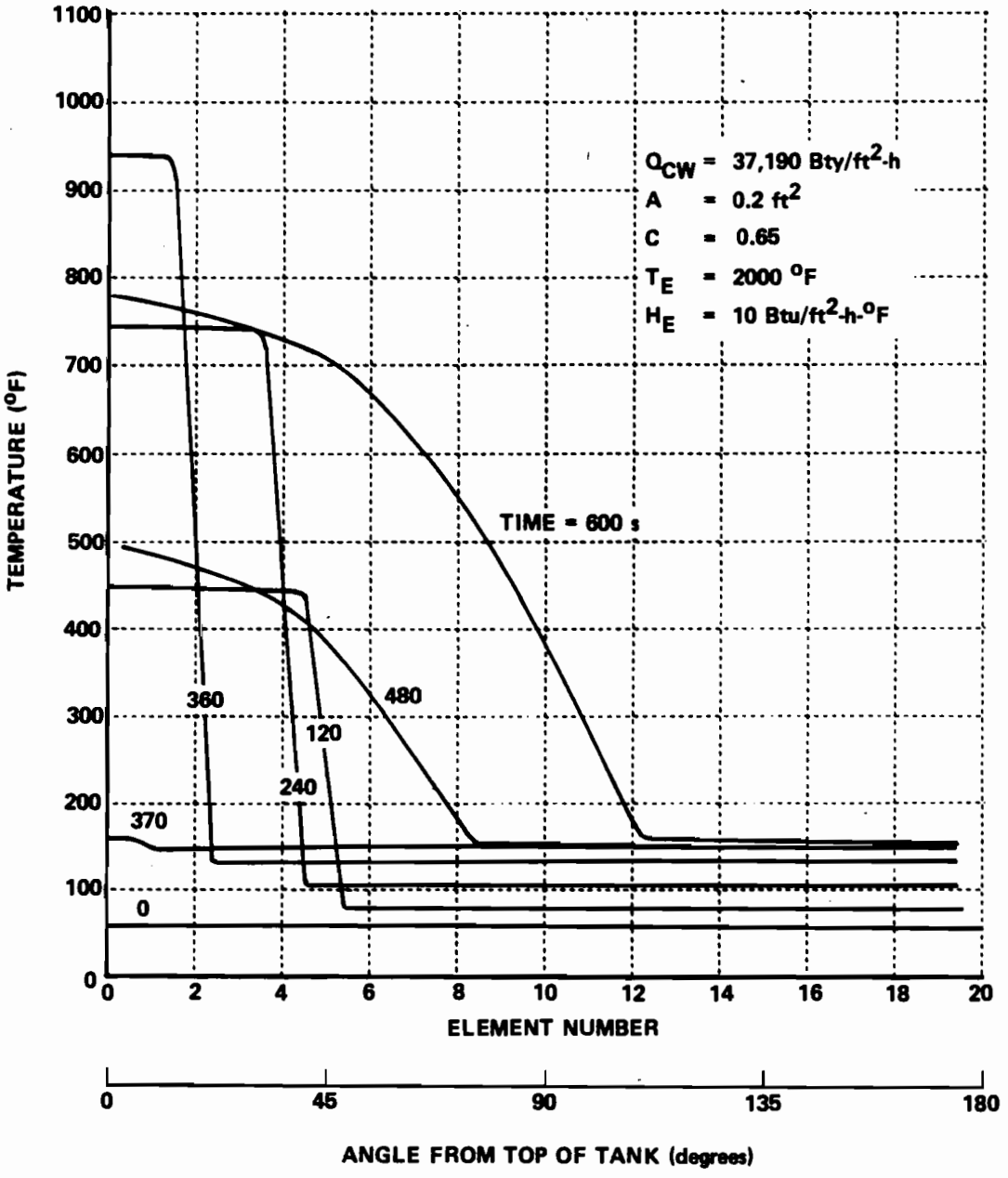

Figure 24 TEMPERATURE DISTRIBUTION IN THE TANK CAR SHELL, 112A340W CAR - PROPANE

Section 6

KEY FINDINGS AND CONCLUSIONS

The key findings and conclusions are:

- Measurements made at CAL indicated that flame temperatures in excess of 2200°F were produced by burning propane without the benefit of induced aeration. Flame temperature for propane combustion in air under ideal conditions has been measured at 3500°F.

- Local unit thermal flux levels of 90,000 Btu/h/ft^2 have been measured at CAL in free-burning hydrocarbon fires.

- Fire volume and burning time calculations, plus data from recent conflagrations, indicate that nearly total car envelopment is possible for significant periods of time.

- The existing A8.01 and A8.02 safety-relief flow capacity formulas given in the AAR Specifications for Tank Cars indicate applicability to compressed gases in general. However, the relationships from which these formulas are derived restricts their applicability to vapor flow of compressed gases for which a saturated liquid phase exists at the flow rating pressure of the safety-relief system.

- The empirical expression 34,500 $A^{0.82}$ Btu/h contained in the AAR Specifications for determining overall thermal input to uninsulated tank shells was intended for use in providing protection against fire exposure of a arbitrarily defined severity, not the worst possible conditions. A questionable correlation technique was utilized for determining the exposure factor for a given vessel area.

- On the basis of available test data in reviewed literature, thermal loads on vessel sizes of interest could be more than double those predicted by the above expression--which is an integral part of the formula for sizing relief valves for tank cars.

- Use of the area exposure factor leading to the $A^{0.82}$ expression, a part of all the AAR Tank Car Specification Appendix "A" relieving capacity formulas, results in a net shift in the undesirable direction of underestimating peak vapor flow.

- Practical consequences of the underestimation of thermal load with regard to relief flow capacity are not necessarily of the same proportion as the degree of underestimation might imply. Physical properties of the lading and car construction factors such as presence or absence of insulation have significant effects. However, underestimation of the thermal load is particularly critical with respect to liquefied compressed gas ladings.

- The existing relief capacity formulas are based exclusively on vapor flow through the valve. Given the possibility that the overturning of a tank car in a derailment might place liquid at the relief inlet, consideration must be given to flashing flow through the valve. (Note: for convenience, we refer to this as the "liquid case", still recognizing that partial or total change of state may occur on passage of the fluid through the relief orifice).

- Considering propane specifically, liquid relief and not vapor relief is the controlling case with regard to relief valve sizing requirements, even though mass flow is greater for the liquid case given equal discharge coefficients. Therefore, even if one assumed the flux rate used in the current flow capacity formulas (which are based upon vapor relief) to be correct, the flow capacity determined by these formulas would be inadequate. Furthermore, the liquid discharge coefficient may be smaller than for vapor discharge.

- Properly functioning safety-relief valves, sized and tested under existing specifications, cannot be expected to pass sufficient propane from a fire enveloped, overturned car to prevent overpressure conditions -- even if they had a discharge coefficient of unity.

- The consequences of inadequate relief capacity -- overpressure -- as a contributor to car failure could be effectively masked by evidence of fire and mechanical damage. Common post-accident testing (e.g., the determination that safety relief valves were operable) will not reveal this condition. Evidence of metal thinning due to overheating of vapor space metal does not sharply delineate the stress level and hence pressure where failure occurred.

- Lowering valve start-to-discharge pressures will not satisfactorily compensate for inadequate relief capacity.

- Flow characteristics of a safety-relief valve operating with a liquid feed, particularly near saturation conditions, may differ markedly from all vapor performance due to the existence of an altered pressure profile through the valve. Existing valve flow capacity tests do not provide for testing at these conditions -- hence, actual relief capacity for a condition likely to occur in an accident is unknown.

- Reported observations at derailment sites of relief flow from cars which subsequently ruptured indicated that actual flow may have been substantially reduced from that anticipated for a fully-opened valve.

- In addition to the current omission of liquid relief capacity requirements, the unrestricted use of the currently permitted extrapolation techniques for vapor flow rating is an area for concern. Turbulence, arising from the flow geometry of a particular valve design, can exert a "choking" action under high pressure, high-mass flow conditions. Onset of "choking" action will not necessarily be predicted by the "four point test" at lower pressure levels. Full capacity flow rating for new valve designs would appear prudent.

- Given the existing safety-relief system, tank car failure due to compressed liquid expansion is possible with propane lading under severe fire exposure conditions.

- The existing tank car specifications fail to relate high-temperature performance characteristics of shell constructional materials, and insulation (if used), to safety-relief requirements in order to establish consistent levels of protection.

- Existing uninsulated pressure cars containing a liquefied compressed gas such as propane may be expected to fail in minutes when involved in a large hydrocarbon fire. Failure, originating in the vapor space, may occur in mechanically undamaged cars, regardless of car orientation.

- Techniques which would reduce rate-of-rise and peak metal temperatures by relatively small margins would greatly improve the chances of preserving shell integrity when coupled with a relief system of adequate flow rating.

- Actual mode of failure of existing uninsulated cars loaded with LPG exposed to fire may be expected to be strongly dependent on car orientation and outage conditions, as well as fire intensity and duration.

- No single idealized specified outage was identified for enhancing the chances of car survival under all derailment and fire conditions.

- "Rocketing" of rupturing LPG cars can be expected for all-gas, all-liquid, or any intermediate condition of fill. The liquid-fill rupture is unlike those produced by hydrostatic failure with a material such as water, where low magnitude compressed liquid expansion and transit time of the stress-relieving wave are the principal considerations in determining thrust. In common with other liquefied compressed gases, LPG will begin to vaporize as the pressure starts to lower with car rupture, with attendant large volume expansion. This results in a much slower pressure drop with time, massive expulsion of mixed vapor and liquid and the development of high thrust. Note that combustion need not be a factor in thrust development.

- Considering the above, and given car construction that does not limit fracture propagation, the relief system must bring internal tank pressures to near outside ambient to prevent rocketing.

- No practical relief system is likely to detect all combinations of damage to a car which may cause the shell to rupture below relief system operating pressure and adjust the relief setpoint(s) accordingly. Therefore, the development of a tank car shell which will limit crack propagation is desirable.

Summary Finding

This study has indicated a need for extensive changes in the existing safety-relief specifications for tank cars. Section 7 will describe a number of general recommendations. The technical text has specifically detailed the mathematical techniques for determining flow capacity requirements for propane. The four-month duration of the program precluded similar analyses with other materials but the general methods to arrive at these requirements are provided. Further recommendations with regard to relief systems as applied to various basic groups of hazardous products are also included in Section 7.

Section 7

RECOMMENDATIONS

7.1 SAFETY-RELIEF SYSTEM FOR LIQUEFIED COMPRESSED GAS SERVICE

Consideration should be given to a parallel, staged safety-relief system. The primary relief system (having the lowest setpoint) could consist of a conventional safety-relief valve. This valve would be sized to relieve minor abnormal conditions such as exposure to high atmospheric ambient temperature or low-level fire exposure and reclose as conditions return to normal. The secondary relief system would involve a high-capacity discharge system with no shutoff provisions. A rupture disc could serve as the secondary device. For reasons we will discuss later, the disc would probably not be of the type currently used in railroad low pressure service. Given a fire of sufficient intensity threatening the integrity of the pressure shell, the secondary system would activate, discharging until ambient pressure is reached.

Location of the primary and secondary relief systems need not be side-by-side. The top centerline location of the car is favored, but it would not be necessary to confine the secondary relief to the manway cover. The placement of the secondary relief device could be similar to that of the current safety-relief on a 111-series nonpressure car. Distance from the vertical mid-line is not particularly critical. However, placement of a device such as a rupture disc close to the end of a car could render it susceptible to premature failure from hydrostatic loading when the liquid lading shifts on hard deceleration. A conceptual arrangement is shown in Figures 25 and 25a. Model specifications with regard to setpoint and flow-capacity requirements appear in Section 7.2.

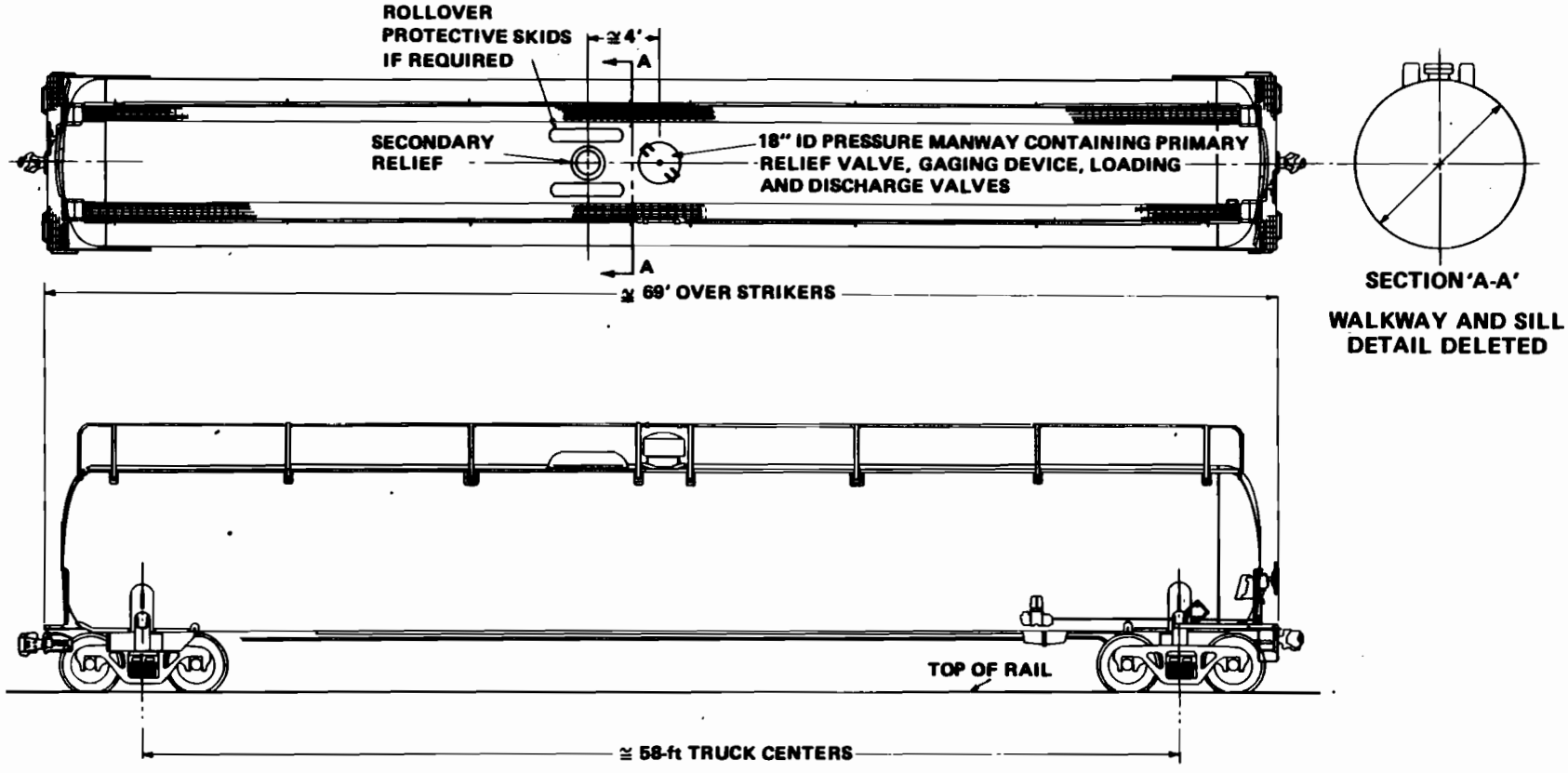

Figure 25 CONCEPTUAL TANK CAR ARRANGEMENT
DOT 112A340-W 33,500 GALLONS

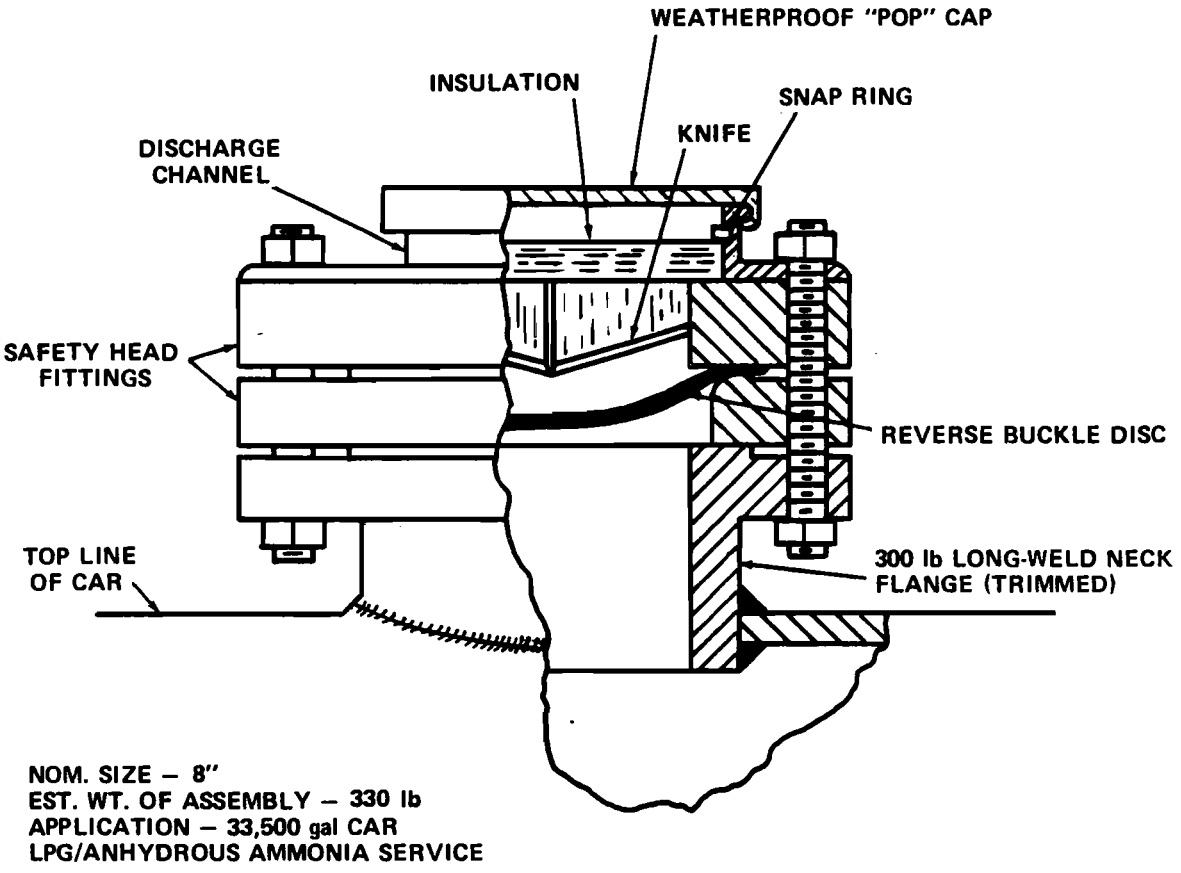

Figure 25a CONCEPTUAL SECONDARY RELIEF DEVICE

7.1.1 Advantages and Limitations

For the moment, we will discuss the proposed safety-relief system of a liquefied gas lading whose primary hazard is flammability. One of the principal arguments against nonclosing relief devices such as a fusible plug or rupture disc is that the contents of the car are entirely discharged, thereby creating a fire much larger than the one which initiated relief action. In the case of the proposed system, there is little reason to believe that a fire of sufficient magnitude to activate the secondary relief system would be under control momentarily. Indeed, the fire is posing a threat to the integrity of the pressure vessel. Therefore, the choice is having a large fire, rather than explosion, rocketing and still having a large fire.

The choice of location is based on the following reasoning. If one distributes devices to assure that at least one device communicates with the vapor space regardless of car orientation then one also insures that one will communicate with the liquid space, and thereby provide a source of fuel to start an undesirable regenerative heating cycle. Internal arrangements to insure communication to a vapor space regardless of orientation have numerous limitations, particularly where two-phase flow is involved. In addition to fire, there are other abnormal conditions (e.g., high pressure due to a faulty purging, or overfilling), which normally occur in the upright car position. In the event of overfilling, for example, at rates exceeding the capacity of the primary relief system, total loss of lading can still be avoided with the top-mounted relief. Initial discharge rate from the secondary relief orifice would be high. However, since the heat necessary for vaporization must come from the car contents, rapid cooling will take place in the absence of an external heat source. As a result, pressure and flow rate will rapidly diminish to controllable levels, with significant lading quantities remaining. It would appear useful to standardize discharge channels to accept plugging and capping equipment to facilitate control similar to that used for pipelines.

We have indicated that a safety-relief valve paralleled with a rupture disc would provide one physical means of achieving system requirements. In the past, principal tank car use of rupture discs has involved the employment of a ruggedized prebulged (tension) type in the DOT 103-series cars. For pressure service, a disc such as the reverse-buckling type may be required. This type permits close relief tolerances and operating margins unobtainable with the standard prebulged discs.

A disc material such as Inconel should be chosen to prevent premature failure due to excessive "derating" at high temperatures. Insulation of the disc material may also be desirable to prevent the disc from reaching failure temperature rapidly due to its low thermal capacity, when the pressure shell metal is not yet threatened by excessive temperature.

It is conceivable that the primary and secondary relief functions could be accommodated in a single device (e.g., a latching relief valve). Such a valve could operate proportionally in its initial stages. If sufficient flow caused a certain percentage of lift to develop, a trip and latch mechanism would drive the valve to full opening and lock it. Regardless of the equipment adopted for secondary relief, it should be flow-rated for both liquid and vapor service.

It should be noted that the choice of flow capacity for both the primary and secondary relief devices is a compromise. There is a conceivable set of circumstances, for example, where a high-intensity torch of sufficient diameter to overcome the effects of wall conduction could impinge on vapor space metal only. Car rupture prior to activation and bleed-down by the secondary relief system could then be possible. Similarly, given a fire of extremely severe intensity, the rate of pressure loss due to discharge from the secondary relief system may not be rapid enough to reach safe levels before failure originating in the vapor space metal occurs. A larger relief orifice, however, could develop reactive loads causing significant skittering of an overturned car while discharging.

It is clear that although the proposed relief system offers a wide range of protection, thermal protection for bare shell metal is very desirable.

7.1.2 Retrofit

The proposed staged relief system is amenable to retrofitting the existing fleet, if this is desired. For example, the existing safety relief valves could perhaps be downrated in capacity by modifying the maximum lift, or they could be replaced. A nipple, reinforcement and safety head could be mounted on the shell along the top certerline of the car for secondary relief. Alteration is currently permitted under existing DOT regulations (T.C. George's Tariff No. 23, Subparagraph 173.31) and AAR Specifications for Tank Cars Appendix "R".

We have principally considered flammable liquefied compressed gases; however, we consider the above recommendations suitable for nonflammable liquefied compressed gas ladings. Considering toxic ladings, it is recommended that an individual study be considered for any material carrying a National Fire Protection Association toxic hazard rating of 4 (NFPA "Guide on Hazardous Materials", 3rd Edition) or a toxic rating of 3, if the vapor density is greater than that of the air.

This study has also indicated potential problems with hazardous ladings carried in nonpressure cars with insufficient relief capacity. The application of a staged relief system with secondary bleed-down for those cars is not necessarily indicated. The retained pressure, given adequate valve capacity, should limit their rocketing potential. Given burst strengths to 500 psig with the 111-series cars, rocketing is a possibility with pressure buildup.

7.2 SUMMARY OF PROPOSED TANK-CAR SAFETY-RELIEF SPECIFICATIONS: LIQUEFIED COMPRESSED LADINGS

Design Basis for Safety-Relief on Uninsulated Car

- Primary System: $q = 4000$ Btu/h/ft^2 — Size orifice on vapor flow basis
- Secondary System: $q = 20{,}000$ Btu/h/ft^2 — Size orifice to largest requirement determined for liquid or vapor basis

Design Basis for Safety-Relief on Insulated Car

Capacity of primary relief may be adjusted from bare car value based on insulation properties within the effective temperature range of the insulation.

No credit for insulation shall be allowed for computing secondary relief requirements.

Flow Capacity Requirements Formulas

For Liquid Flow: $$\dot{W}_L = \frac{qA}{L_o}\left(\frac{V_{go}}{V_{fo}} - 1\right) \text{ lb/h} \qquad (85)$$

For Vapor Flow: $$\dot{W}_V = \frac{qA}{L_o}\left(1 - \frac{V_{fo}}{V_{go}}\right) \text{ lb/h} \qquad (86)$$

, below critical point. $V_{go} >> V_{fo}$ $\therefore$ $\dot{W}_V \cong \frac{qA}{L_o}$ lb/h

where

$\dot{W}$ = weight flow in pounds per hour
q = heat flux in Btu/h/ft^2
A = tank car shell area in ft^2

L_o = latent heat at vaporization at flowing conditions in Btu/lb

V_{go} = specific volume of gas at flowing conditions in ft^3/lb

V_{fo} = specific volume of liquid at flowing conditions in ft^3/lb

All relief devices should be flow rated. Flow rating for all-vapor conditions may be accomplished by existing methods, except it is suggested that the rating of all new designs should not be based on the alternate extrapolation methods.

Flow rating for liquid should be accomplished with the working fluid at temperature and pressure fixed by saturation conditions at the valve setting. If it is found necessary to test at other conditions, the analytical technique used for correction to "flowing" values should be submitted for approval. No generalized technique can be stated, but it is suggested that any analytical treatment include a parametric sensitivity analysis of test variables to assist in defining appropriate safety factors where results cannot be directly obtained by test.

Suggested relief setpoints are plotted in Figure 26. These are based on the following values:

- Hydrostatic Test Pressure: 40% of design burst pressure (existing specification)
- Primary Relief-System Setpoint: 30% of burst (75% of Hydrostatic)
- Secondary Relief System: 33% of burst (82.5% of Hydrostatic) (Nominal Setpoint)
- Maximum System Pressure*: 36% of burst (90% of Hydrostatic)

*Flow Rating Pressure: See equations (68) through (72) for relationship between nominal setpoint and maximum system pressure. Note that simplified capacity formulas (equations 85 and 86) actually provide protection well above 20,000 Btu/h ft^2 when finite total discharge times are considered in connection with maximum permissible pressure rise.

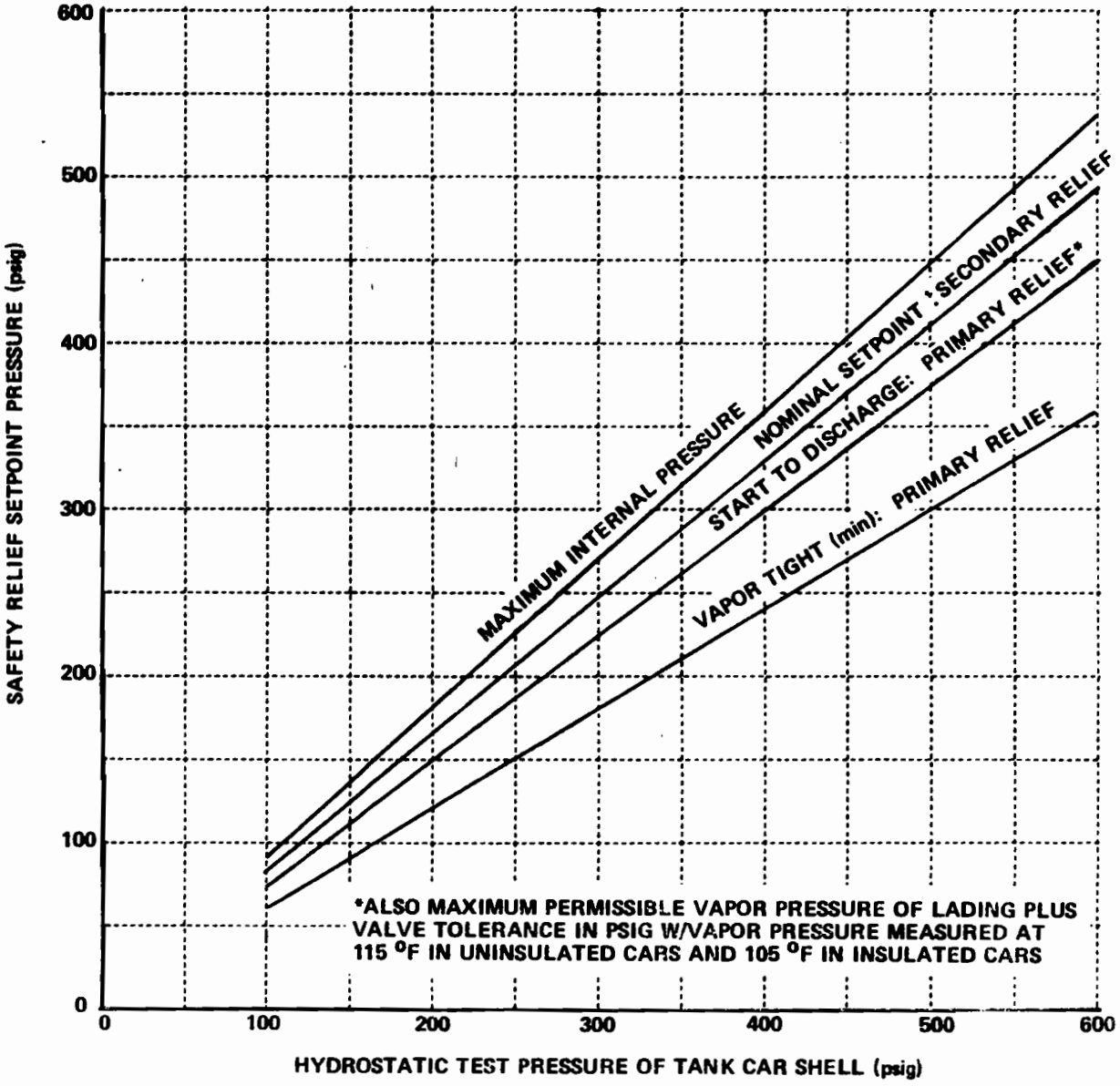

Figure 26 PROPOSED RELIEF SYSTEM SET PRESSURES

Comment:

The weight flow requirements may be stated in terms of air at standard conditions for comparative purposes with existing capacity tables based on air.

$$Q_a \cong \frac{18.34\,W}{C}\sqrt{\frac{ZT}{M}}$$

Q_a = flow of air in standard cubic feet per minute (S.T.P. = 14.7 psig and 60°F)

Z = compressiblity factor

T = Temperature in ° Rankine at flowing conditions

M = molecular weight

C = gas constant = $520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}$ where k = ratio of specific heats.

The choice of 4000 Btu/h ft^2 as the heat flux basis for the primary relief system is a compromise value intended to provide controlled release of contents to alleviate minor abnormal conditions. The primary system would relieve the vapor load generated by a hydrocarbon fire enveloping approximately one-eighth of the wetted area of the tank shell. Primary relief capacity, therefore, would provide approximately one-half the total relieving capacity of existing systems. This relieving capacity would easily be adequate for high temperature conditions resulting from other than direct fire.

Accidental overfilling is a predictable occurrence. In the case of propane the primary relief system for a 33,000-gallon car would provide release rates in excess of 200 gal/min without activating the secondary system. Failure to properly purge a tank of non-condensable gases such as air can result in relief-valve activation during liquid filling operations to relieve pressure developed from compression of trapped gas. The primary system will accommodate non-condensable gas flow due to displacement at the maximum liquid filling rate for which overfill protection is provided.

The base point for establishing a pressure specification for a liquefied gas lading is established by the vapor pressure of that lading. In the case of an "ethane rich" commercial propane, or anhydrous ammonia this pressure will be

approximately 255 psig at 115°F for both materials (pure propane would be 213 psig). It was recommended (Figure 26) that the start-to-discharge setting be the highest anticipated vapor pressure plus the setpoint tolerance of the primary relief device. In other words, if a relief device used for anhydrous ammonia service has a tolerance of ±8 psig, the start-to-discharge setpoint would become 263 psig in an uninsulated car. Again referring to Figure 26, the nominal setpoint for the secondary relief device was established at a practical level taking into account setpoint tolerances between primary and secondary systems, plus operating margin requirements for reverse-buckle rupture discs.

Beyond these points, which are essentially fixed by lading properties and mechanical tolerances, one can adjust permissible stress level specification Given existing cars, there is little room for adjustment between the existing specifications and the dictates of vapor pressure. With new cars, of course, there is wide freedom to set maximum allowable pressure at any particular percentage of either tensile or yield strength and build cars accordingly.

Lowering the pressure setting can be undesirable from a relief standpoint with the existing valves. For the overturned car situation regenerative heating would begin sooner...and rate of pressure rise may actually be increased despite the earlier valve opening. In other words, a reduction in valve setting would not compensate for an inadequate valve capacity. There, of course, may be structural considerations which would dictate a lower setting for a car of particular construction, but it does not alter the valve problem.

The suggested pressure specifications are within the range of accepted practice when compared with other pressure services, though they are not tightly conservative. The widely accepted ASME Unfired Pressure Vessel Code specifies working pressures at 25 percent of tensile strength, with 20 percent pressure accumulation permitted for safety relief under fire exposure conditions. This is equivalent to a maximum system pressure at 30 percent of burst strength, compared to the 36 percent suggested maximum for tank cars. The latter figure would permit retrofit of an adequate relief system to existing equipment as an

interim requirement pending a detailed examination of the structural considerations in tank car design. This study has indicated shortcomings in safety-relief design which need no structural deficiency contribution to account for violent ruptures observed.

7.3 ECONOMIC IMPLICATIONS

Adoption of any of the above recommendations would involve capital expenditures. To determine some order of magnitude approximations of distributed annual costs of a car as a function of its initial cost, plus the cost of a retrofit in terms of remaining car life, Figures 27 and 28 were prepared. These plots are based on 30-year car life and interest at 10 percent.

In January 1969, there were 177,460 tank cars in service [2], of which approximately 10 percent were large-capacity 112-series cars. It is estimated that the 112-series cars generate approximately 10,000 loaded car-miles per car per year. By virtue of their being used for the bulk shipment of low-cost commodities, the cost of a car can assume a significant proportion of the overall transport cost structure and final product price.

It is instructive to consider some facts about a typical 112A340W of a nominal capacity of 33,500 gallons. Light weight of the car will be approximately 90,000 lb, giving a capacity of 173,000 lb with 100-ton trucks and 225,000 pounds with 125-ton trucks.

The car loaded with 0.51 gravity LPG will be volume limited, with a maximum lading weight (summer fill) of approximately 128,000 lb of material worth approximately $0.011/lb "works" price [28]. The same car loaded with anhydrous ammonia will have a lading weight of approximately 159,000 lb of material worth approximately four times the LPG on a per pound basis.* The high-specific-gravity vinyl chloride monomer will be weight limited--225,000 lb (with 125-ton trucks) of a material worth approximately seven times the LPG on a per pound basis.* Of these three common ladings transported in 112A340W cars, it is clear that the cost of the car is most significant with regard to shipping LPG.

*Based on historical price data--Oil, Paint & Drug Reporter.

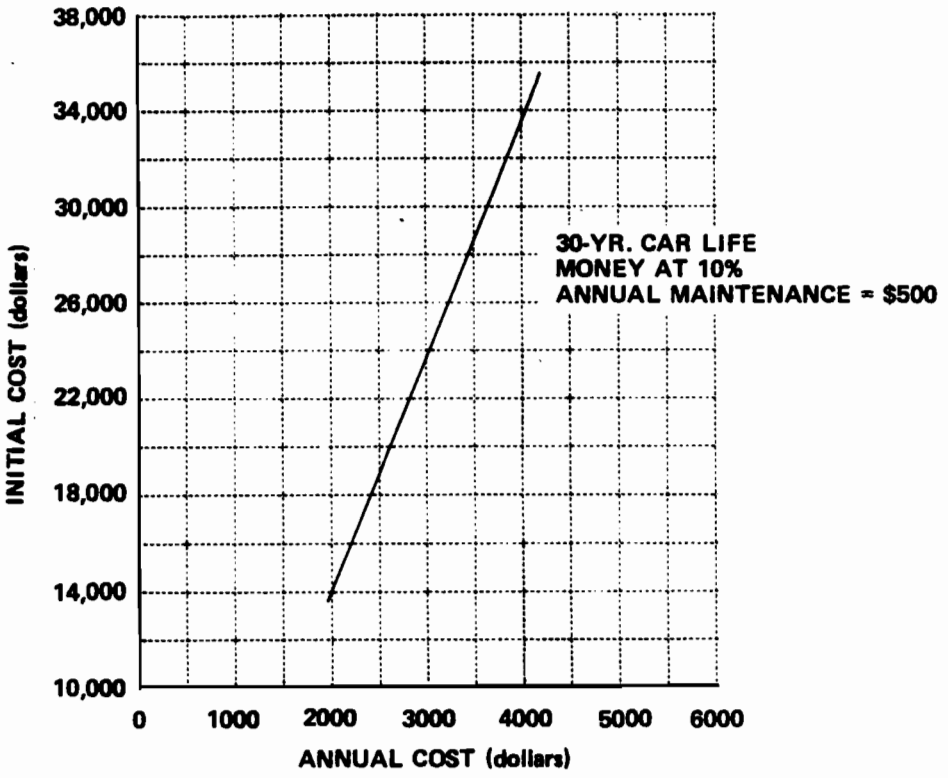

Figure 27 ANNUAL COST OF CAR AS FUNCTION OF INITIAL COST

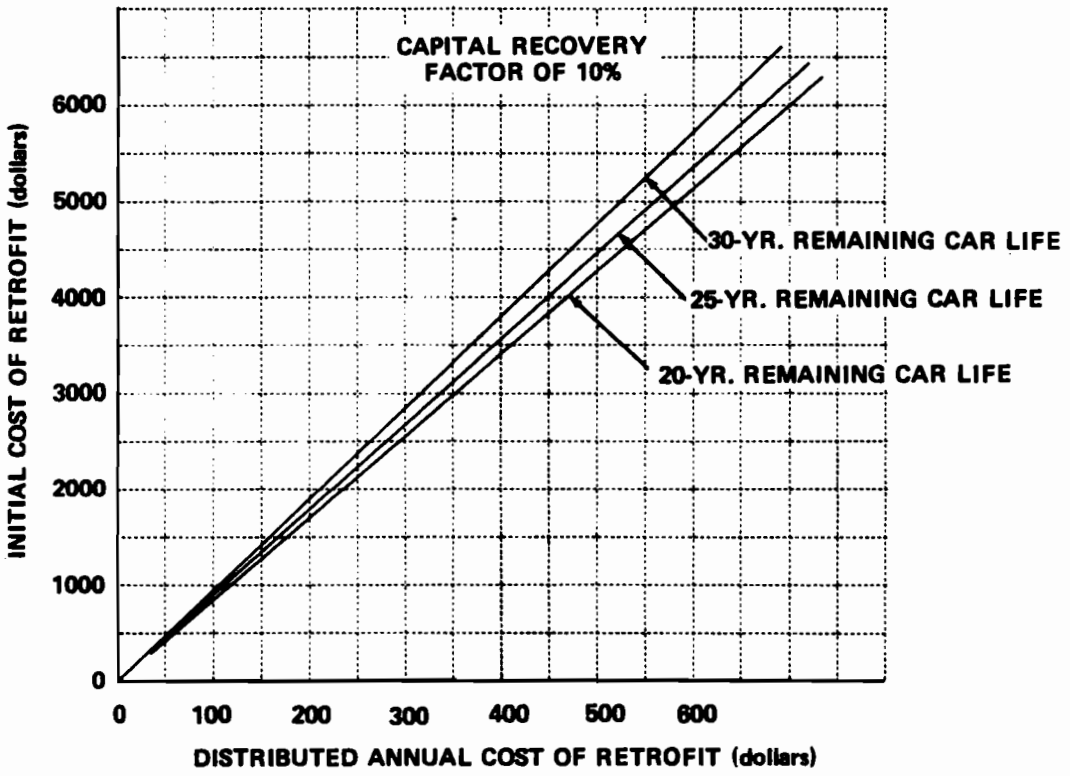

Figure 28 ANNUAL COST OF RETROFIT PROGRAM AS A FUNCTION OF INITIAL EXPENSES AND REMAINING CAR LIFE

Before proceeding, let us examine the effect a change in gross weight in the car as a result of design changes might have on the freight tariff. The principal hauling expense varying as a function of weight would be expected to be fuel cost.

Fuel cost in 1969 amounted to approximately 4% of railroad operating expenses [29]. A 20% increase in weight may be expected to give approximately a 12% increase in train resistance at 40 mi/h on level track. Taking this as a rough estimate for determining operating cost increase, or (0.12 x 0.04 x 100) $\cong$ 1/2% for a 20% increase in gross weight. The insensitivity of cost to gross weight change indicates that sensitivity of unit weight (or volume) billing to changes in payload weight could be significant. Therefore, from a cost standpoint, it would be cheaper to increase car weight for a given payload than to reduce payload to maintain a given car weight --- where gross weight limitations permit. Addition of the safety-relief system recommended in this report would not involve a significant weight change.

If consideration is given to reducing authorized filling densities, for example, to avoid shell-full conditions below safety-relief valve settings, substantial loading reductions would be required. In the case of propane shipped in a 112A340W car with the alternate valve setting, a 5 percent additional weight reduction ($\cong$ 1500 gal) would be required beyond current authorized summer loadings. This is the equivalent of reducing the fleet capacity by the same percentage, in addition to raising per gallon transportation cost.

Still considering a 112A340W car with propane, the lading is worth \$0.048/gal based on Gulf Coast "works" price [28] or \$0.125/gal tank truck quantity retail price in Western N.Y.* At 30,000 gallons per car load, this amounts to total values of \$1440 or \$3750 per carload, respectively. It is estimated that each car, with an initial cost on the order of \$23,000, will make approximately 20 trips annually.

*Quote, Buffalo, N.Y., October 1970.

Pursuing this further, it is useful to obtain some approximate evaluations of sensitivity of original car cost as an increment to the price of propane, while also comparing the effects of other variables. Annual car costs are taken from Figure 27, and various loaded car-miles and miles per trip figures are used. For the trip mileages computed, the ton-mile rail billing is assumed constant as an approximation. The 10,000 loaded car-miles per year, 500 miles per trip base are estimated to most closely reflect the current figures for 112-series cars.

(1) Base: $23,000 car, 8,000 loaded mi/yr, 1000 mi/trip, 30,000 gallons/load at $0.02/ton-mile.

Annual Value of Cargo at Shipping Point	+	Rail Billing	+	Annual Cost of Car	→	Annual Value at Receiving Point
$11,520		$11,280		$2,796		$25,596 or $0.107/gal

Unit Value at Receiver less Value at Shipping Point = Transport Increment

$0.107/gal - $0.048/gal = $0.059/gal Transport Increment

(2) Base: Same as (1) except 10,000 mi/yr
$0.105/gal - $0.048/gal = $0.057/gal

(3) Base: Same as (1) except 12,000 mi/yr
$0.103/gal - $0.048/gal = $0.055/gal

(4) Base: $30,000 car, 10,000 loaded mi/yr, 1000 mi/trip, 30,000 gal at $0.02/ton-mile
$0.107/gal - $0.048/gal = $0.059/gal

(5) Base: $23,000 car, 10,000 loaded mi/yr, 500 mi/trip, 30,000 gal at $0.02/ton-mile
$0.076/gal - $0.048/gal = $0.028/gal

(6) Base: $30,000 car, 10,000 mi/yr, 500 mi/trip, 30,000 gal at $0.02/ton-mile
$0.078/gal - $0.048/gal = $0.030/gal

Comparing examples ⑤ and ⑥, a 7% increase in the cost of transport would accompany a 30% increase in car cost for the conditions given.

Approximate costs* of installation of a staged relief system for both new cars and existing cars are given below. Because utilization factors for both the car and the shop are so highly variable, no estimate is made for possible loss of lease revenue while out of service for shopping. This does not mean these costs are necessarily considered negligible.

Estimated Incremental Cost of Installation of Staged Relief System on New Cars

Material for 8-inch 300-lb carbon-steel flanged safety head, Inconel reverse buckling rupture disc, inlet nozzle and reinforcing	$360.00
Credit for reduced cost of primary safety-relief valve	10.00
Additional Labor	50.00
Total	$400.00

Estimated Cost of Retrofitting Existing 112-Series Cars With Staged Relief System

Material for 8-inch reverse-buckle rupture disc secondary relief	$360.00
Scrapping existing safety relief valve (depreciated valve cost -- typical)	100.00
New Safety-relief valve and adapter fitting for old bolt circle and seal	200.00
Miscellaneous materials expense	40.00
Labor (includes cleaning and retest expense)	700.00
Total	$1400.00

*Cost data partially derived from Chemical Engineering, 2 November 1970, and Black, Sivalls and Bryson Safety System Catalog.

If 17,000 cars are in existence, the cash outlay for retrofitting would be on the order of 22 million dollars. True cost of retrofitting would be larger, since the 22 million dollar figure does not include the loss due to premature scrapping of the existing valves shown in the above itemization, the loss of lease revenue, or capital recovery factors.

On the other hand, dollar loss attributable to catastrophic rupture and rocketing of tank cars is significant. Therefore, even though the proposed relief system would not prevent derailment and localized fire, savings due to reduced total loss could be significant. It would be speculative, however, to attempt to fix a dollar value for a projected loss reduction.

7.4 RECOMMENDATIONS FOR FURTHER RESEARCH

A priority test program should be instituted to investigate the performance of existing safety-relief valves with liquid feed, particularly at saturation conditions. Thermodynamic considerations indicate the possibility that a valve with demonstrated "pop" action and rated flow capacity under test with a gas may respond entirely differently with a liquid feed. The possibility exists that with liquid feed, the valve may act as a proportioning device, essentially releasing material consistent with expansion of the lading with temperature rise ---rather than maintaining constant internal pressure.

A scenario of what could take place in an accident involving an LPG lading and the above postulated relief valve response follows:

One car has been punctured as a result of the derailment and escaping fuel has ignited. After the initial large burst of flame has subsided, the remaining fire is partially enveloping an adjacent overturned car. The magnitude of the fire is low enough that the thermal flux is well below the nominal rated capacity of the safety-relief valve. After an extended period--possibly several hours--thermal expansion of the liquid in the heated car results in a shell-full condition (typically at approximately 115°F). Under compressed liquid conditions, pressure rise to the valve setpoint is rapid -- and the valve opens. The discharged fluid ignites, increasing the fire intensity; the beginning of a regenerative heating cycle which will end in catastrophic

rupture. Although the fire intensity has increased, it will not be to the extent expected with the 100+ lb per second flow of a fully open valve. Actual flow could be on the order of 10 to 20% of that anticipated. The valve would remain continuously open or exhibit rapid cycling (a common instability produced in valves passing flashing liquids), but it would not remain closed for extended periods. Because the valve is acting as a simple liquid relief, the car remains in shell-full condition, and thus for a period of time the shell is protected from overheating even with the increased fire intensity. Internal pressure in the car would increase with the rise in temperature of the contents, closely following the saturation curve of a pressure-temperature diagram. At some point in time--possibly several hours after initial safety relief valve functioning--the car would violently rupture at high pressure, with sufficient energy to hurl fragments weighing tons for extended distances. The instantaneous release of fuel would be sufficient to produce a fireball hundreds of feet in diameter.

The reasons for placing a number one priority for the liquid flow tests may be summarized as follows:

- Of all postulated failure modes, the restricted liquid flow failure could occur with the lowest intensity of fire required to produce failure.
- The highest potential energy at burst is possible with this failure mode.
- The reported action [30] of Car No. 28 at the Crescent City, Ill., disaster gives rise to* the suspicion that this failure mode may have been involved.

*This car was heated 30 to 50 min before the safety valve opened. This time indicates a heat input of approximately 4000 Btu/h ft^2, which is less than the valve design capacity. The valve apparently remained open for more than 2 1/2 hours, and then the car ruptured. The average flow through the valve in this time would have been less than 14 lb/sec for flow to be maintained for 2 1/2 hrs whereas the calculated liquid flow for a fully opened valve is 110 to 170 lb/sec depending on the discharge coefficient (Equation 83).

- While the restricted flow mode is thermodynamically possible--actual test of specific valves is required to prove or disprove its existence with a particular design and fluid.
- Existing specifications and flow tests do not encompass this potential problem area.
- Determination whether the problem exists only in theory, or is in fact a real problem with existing valves could affect a decision on the urgency of retrofitting existing cars to updated design standards.

In addition to the liquid relief tests, any existing valve designs which have not been vapor flow rated at maximum service pressures should be so rated at actual peak pressures.

Program to Verify Efficacy of Proposed Relief System

A scaled test program is recommended to verify the efficacy of the proposed relief system. As a minimum, propane would be utilized as a test fluid. The program could easily be expanded to cover additional materials of interest. Tank models and appropriately scaled relief systems would be subjected to varying thermal loadings, with the relief system in various orientations, to investigate performance. A relief system sized to the currently existing specifications would be used as a control. As with the case of testing existing relief valves, care must be taken to insure that the proposed safety-relief devices are tested under conditions which realistically indicate their effectiveness as safety devices on tank cars. Test conditions that must duplicate or properly simulate, those of a relief device on a tank car in a fire include: pressure and temperature of the lading, liquid and gas phases at the relief entrance, rate of increase of pressure, exit flow conditions, and, most importantly, the internal flow geometry of the test devices.

Even though tests would be scaled, simulation requirements will probably dictate sizes of equipment necessitating an outdoor test range. Instrumentation should be complete enough to determine a time, temperature, pressure, and liquid-vapor interface level history in response to the thermal loading, as

well as to characterize the nature of the thermal load itself. The use of free-burning hydrocarbon fuel fires as a thermal source would aid in achieving radiation and convective components of heat transfer in proportion to that which is likely to occur in an actual derailment and fire. Some tests simulating the "upset case" condition should be allowed to proceed regeneratively as they would under field conditions, with no attempt to hold a constant thermal loading by reducing external fuel feed.

As an output the program would be expected to provide:

- Characterization of the thermal load resulting from a free-burning hydrocarbon fire (temperatures, radiative and convective components of heat transfer, etc.)
- Response of the lading to thermal load (temperature, pressure, change in liquid-vapor interface, etc.) for comparison to results predicted from theoretical considerations.
- Temperature-time history at multiple locations on the shell (Note: If the scaled tank is matched for burst characteristics, wall thickness differences will have to be properly evaluated in interpreting results for full-size tanks.)
- Response of existing and proposed relief systems to thermal load. Information derived would go beyond "saves/does not save" the car in that indications of operating margins would be secured, and alterations proposed, if required, prior to full-scale trials.

A computer program was developed in conjunction with this study to analyze car and lading conditions with various thermal loads as a function of time. It is suggested that this program be refined and expanded to handle additional conditions, e.g. compressed liquid, to make it a more useful tool for future safety studies. The proposed test program to investigate the efficacy of relief systems would provide valuable data for refinement of the computer simulation. It is anticipated that at some future time, full-scale fire tests of tank cars with flammable ladings must be conducted. A great deal of "spade work" can be done with computer simulation along with scaled tests to check critical variables to assure that a maximum amount of information can be gleaned from desirable but expensive full-scale tests.

There is one "full-scale" test program we would recommend for the near-term. The controversy of the "cold-wall" heat flux rates developed in liquid filled tanks as a function of tank size has been the source of disagreement for over 30 years. The assumed flux levels for tank cars, challenged in this report, were apparently derived via curve fitting and extrapolation from small scale tests. It is suggested that a tank car shell of approximately 34,000 gallon capacity, openly vented to the atmosphere and filled with nonhazardous medium such as water, be utilized as the test vessel in fire exposure heat flux determinations. Free-burning hydrocarbon fueled fires would constitute the heat source. Tests with the tank "on-the-ground" as well as elevated to normal position when mounted on trucks would be useful.

The existing data base is meager with respect to tests with vessels of the size, range, geometry and orientation of tank cars. Therefore such a test program could be expected to yield useful data directly applicable to the tank car case and unencumbered by tenuous extrapolation procedures.

It would be too much to expect, perhaps, that all controversy on the matter would be ended with such tests, since validity of test conditions can always be argued. Nevertheless, the data foundation would definitely be stronger than now exists for the case at hand.

Appendix A

DEFINITION OF TERMS USED IN COMPUTER PROGRAM

A	Relief valve area--ft^2
AEL	Cross-sectional area of each element of the tank car shell--ft^2
C	Specific heat of tank car shell material--Btu/lb-°F
CG	Relief valve flow coefficient
DANG	Included angle of each tank car shell element--radians
DELTA	Time increment for calculations--seconds
DHG	Change in gas enthalpy--Btu/lb
DMG	Time rate of change of mass of gas generated by vaporization--lb/hr
DML	Time rate of change of mass of liquid due to vaporization--lb/hr
EI	Emissivity of inside surface of tank car shell
EM	Emissivity of outside surface of tank car shell
EO	Emissivity of fire
FLIQ	Program flag. FLIQ = 0 for gas flow through relief valve; FLIG = any integer for liquid flow through relief valve

G	Gravitational constant (32.2 ft/sec^2)
HE	Heat transfer coefficient for external tank car environment--Btu/ft^2-hr-°F
HF(M)	Specific enthalpy of liquid at time = time (M)--Btu/lb
HG(M)	Specific enthalpy of gas at time = time (M)--Btu/lb
HGT	Gas heat transfer coefficient for internal tank car environment--Btu/ft^2-hr-°F
HLT	Liquid heat transfer coefficient for internal tank car environment--Btu/ft^2-hr-°F
K	Thermal conductivity of tank car shell material--Btu/ft-hr-°F
KP	Ratio of specific heats
M	Time index
MG(M)	Mass of gas in tank car per unit length--lb/ft
ML(M)	Mass of liquid in tank car per unit length--lb/ft
MR(M)	Mass flow of material through relief valve per unit length--lb/ft-hr
MTOT(M)	Total mass in tank car per unit length--lb/ft
N	Element position index
NEL	Number of tank car shell elements chosen for one-half of the tank car shell circumference
NG	Number of last element to use gas heat transfer coefficient

PL(M)	Pressure in tank car at time = time (M)--psia
PR	Pressure at which relief valve opens--psia
PRL	Low pressure limit for relief valve operation--psia
PS	Sonic pressure for gas flow through relief valve--psia
QG(N)	Gas heat transfer rate per unit area for one element of tank car shell--Btu/ft^2-hr
QGSUM(M)	Total heat input to the internal gas environment from the tank car wall--Btu
QINTO(M)	Heat transfer rate per unit area applied to the outside wall of the tank car--Btu/ft^2-hr
QL(N)	Liquid heat transfer rate per unit area for one element of the tank car shell--Btu/ft^2-hr
QLSUM(M)	Total heat input from the tank car wall to the liquid--Btu
RHO	Density of tank car shell material--lb/ft^3
RP	Gas constant--ft-lb/lb °R
RTANK	Inside radius of tank car shell--ft
SIG	Stefan-Boltzman gas radiation constant (0.173×10^{-8} Btu/hr-ft^2-°R^4)
SIG C	Circumferential stress in tank car shell--lb/in^2
SIG T	Transverse stress in tank car shell--lb/in^2
T(N)	Average temperature of tank car shell element--°F

TAU	Shear stress at 45° plane in tank car shell element--lb/in^2
TE	Fire temperature--°F
TG	Temperature of gas in tank car--°F
THET(M)	Angle from θ = 0 to liquid-gas interface at tank car shell--degrees
THETA(N)	Position of the centroid of each element of the tank car shell--radians
THICK	Tank car shell thickness--inches
TI(N)	Temperature of inside surface of tank car shell element--°F
TL(M)	Temperature of liquid in tank car--°F
TO(N)	Temperature of outside surface of tank car shell element--°F
TS	Sonic temperature for gas flow through relief valve--°R
U_C	Critical velocity--ft/sec
VF(M)	Specific volume of liquid in tank car--ft^3/lb
VG(M)	Specific volume of gas in tank car--ft^3/lb
VOL	Total water volume of tank car per unit length--ft^3/ft
VOLG(M)	Volume of gas in tank car per unit length--ft^3/ft
VOLL(M)	Volume of liquid in tank car per unit length--ft^3/ft

Appendix B

DESCRIPTION OF COMPUTATIONAL PROCEDURES FOR COMPUTER PROGRAM PROJECT REDHOTT--BURST

by W.J. Baran

The following computer program was written to describe the thermodynamic state of the contents of a container of given wall thickness and the container wall temperatures when the container is subjected to an external environment which can provide a net heat input.

A basic observation that is integral to the method by which the majority of thermodynamic properties of both the liquid and the gas in the container are obtained is that the contents of the container are, at all times, in equilibrium. It is, therefore, necessary that these equilibrium conditions be described by the liquid-vapor saturation curves.

The saturation value of HF, PL, TL, VG, VE and L are supplied to the computer in the form of a table in which HF is the independent variable. Values are obtained from the table on the basis of an interpolation between points using a three-point Lagrange curve fit.

The mass and volume of the liquid and gas within the tank car are computed using the equations below:

1. $$ML(M) = \frac{VOL - MTOT(M) \times VG(M)}{VF(M) - VG(M)}$$

2. $$MG(M) = MTOT(M) - ML(M)$$

3. $$VOLL(M) = ML(M) \times VF(M)$$

4. $$VOLG(M) = MG(M) \times VG(M)$$

MTOT(M) is reduced by the amount of mass removed via the relief valve when it is in operation.

From the values above the position of the liquid-vapor interface on the tank car wall is computed, and this value is compared to THETA(N) for each wall element. On the basis of the comparison an element number is chosen, and its value is assigned to NG, the index separator for liquid and gas calculations. For all values of $N < NG$ a gas heat transfer coefficient is used to compute the heat input to the interior of the container. For all values of N greater than NG, a liquid heat transfer coefficient is used.

The heating rate per unit area input to the external surface of the container is computed using the following equation.

$$QINTO(M) = HE(TE-T(N,M)+SIG\cdot EO\cdot EM(TE+460)^4 - SIG\ EM(T(N,M)+460)^4 \quad [5]$$

The heating rates per unit area and the total heat input to the interior of the container are computed as shown below.

$$QG(N,M) = HGT(T(N,M)-TL)+SIG\cdot EI((T(N,M)+460)^4 - (TL(M)+460)^4) \quad [6]$$

$$QGSUM(M) = 2.0\cdot RTANK\cdot DANG \times \frac{DELTA}{3600} \sum_{N=1}^{NG} QG(N,M) \quad [7]$$

$$QL(N,M) = HLT\ (TN,M) - TL(M)); \quad HLT \quad 4000\ Btu/ft^2\text{-}hr\text{-}^\circ F \quad [8]$$

$$QLSUM(M) = 2.0\cdot RTANK\cdot DANG\cdot \frac{DELTA}{3600} \cdot \sum_{N=NG+1}^{NEL} QL(N,M) \quad [9]$$

Once the values above are obtained for all elements from N=1 to N=NEL, the average element temperatures are computed as follows.

$$DAO = (RTANK+THICK)\cdot DANG\cdot \left(\frac{DELTA}{3600}\right) \qquad CON = K\cdot THICK\cdot \left(\frac{DELTA}{3600\ AEL}\right)$$

$$D = RTANK\cdot DANG\cdot \left(\frac{DELTA}{3600}\right) \qquad CRV = C\cdot RHO\cdot THICK\cdot AEL$$

For elements N=1 to N=NG

$$T(N,M) = (QINTO(M)\cdot DAO+CON(T(N-1,M)-T(N,M)) + CON(T(NH,M)-T(N,M)) - QG(N,M)\cdot D+CRV\ T(N,M))/CRV \quad [10]$$

For elements N=NG+1 to N=NEL

$$T(N,M) = (QINTO(M)\cdot DAO + CON\ (T(N-1,M)-T(N,M))+CON(T(N+1,M)-T(N,M)) - QL(N,M)\cdot D+CRV\cdot T(N,M))/CRV \quad [11]$$

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 valve. The second case distinguishes between liquid or gas flow out the relief valve.

CASE 1*:

MR = 0

$$DMG(M) = \frac{(MG(M)-MG(M-1))\cdot 3600}{DELTA} \quad [12]$$

$$HF = \frac{(QLSUM(M)+QGSUM(M)-DMG(M)\cdot L(M)\cdot\frac{DELTA}{3600})}{ML(M)+(DML(M)\ \frac{DELTA}{3600}} \quad [13]$$

$$HF(M+1) = HR(M)+\ HF \quad [14]$$

*The following calculation procedures were utilized for determination of conditions within the tank to approximate more precise, but much more difficult iterative procedures, whereby the new specific volume of the mix, V_m, is established after each time interval as well as the new mix internal energy, U_m, or enthalpy, H_m, depending upon constant volume or flowing conditions. After having established both V_m and H_m, one can determine the state of the mix and hence, the mix pressure. Unfortunately, this technique requires an iterative solution for pressure, P_m, which unjustifiably increases total machine time substantially.

CASE 2:

For gas flow through the relief valve

$$MR(M) = \frac{CG \cdot A \cdot UC(M)}{VC(M)} \qquad [15]$$

where $$UC(M) = \left(KP \cdot G \cdot RP \cdot (TL(M)+460)^{\frac{2}{KP+1}}\right)^{\frac{1}{2}} \qquad [16]$$

and $$VC(M) = \frac{RP \cdot (TL(M)+460)^{\frac{2}{KP+1}}}{PL(M)\left(\frac{2}{KP+1}\right)^{\frac{2}{KP-1}}} \qquad [17]$$

$$MTOT(M) = MTOT(M-1) - MR(M) \cdot \frac{DELTA}{3600} \qquad [18]$$

$$DHG = HG(M) - HG(M-1) \qquad [19]$$

$$DHF = \frac{QLSUM(M) + QGSUM(M) - (DMG+MR) \cdot L \cdot \frac{DELTA}{3600} - MG \cdot DHG}{ML(M) + (DML-MR) \cdot \frac{DELTA}{3600}} \qquad [20]$$

$$HF(M+1) = HF(M) + DHF \qquad [21]$$

For liquid flow through the relief valve

$$MR(M) = 192000 \cdot CG \cdot A \qquad [22]$$

$$MTOT(M) = MTOT(M-1) - MR(M) \cdot \left(\frac{DELTA}{3600}\right) \qquad [23]$$

$$DHF = \frac{QLSUM(M)+QGSUM(M)-DMG \cdot L \cdot (\frac{DELTA}{3600})-MG \cdot DHG}{ML(M)+\frac{(DML-MR)\ DELTA}{3600}} \quad [24]$$

$$HF(M+1) = HF(M)+DHF \quad [25]$$

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). If the two tests outlined below are negative, the calculations are restarted at equation [1] for the next time step.

The computer program will stop automatically if either of the two conditions below are met.

(A) ML(M) = 0 for any time step

(B) If the pressure inside the container exceeds the value obtained from the supplied burst pressure table for any element temperature.

In addition to the above values the following are also calculated.

$$TI(N,M) = T(N,M)-\frac{QINTO(M) \cdot THICK}{6K} \quad [26]$$

$$TO(N,M) = 3.0 \cdot T(N,M)-2.0 \cdot TI(N,M) \quad [27]$$

$$SIGC(M) = \frac{(PL(M)-14.7)\ RTANK}{THICK} \quad [28]$$

$$SIGT(M) = 0.5\ SIGC(M) \quad [29]$$

$$TAU(M) = 0.25\ SIGC(M) \quad [30]$$

A complete list of input and output printout values and the computer program printout in Fortran IV computer language are included in the following pages.

COMPUTER PROGRAM INPUT ITEMS:

1. A
2. C
3. CG
4. DELTA
5. EI
6. EM
7. EO
8. FLIQ
9. G
10. HE
11. HGT
12. HF(1)
13. K
14. KP
15. MTOT
16. NEL
17. PR
18. PRL
19. RHO
20. RP
21. RTANK
22. SIG
23. TE
24. THICK

COMPUTER PROGRAM OUTPUT ITEMS:

1. DMG
2. HF
3. MG
4. ML
5. MR
6. NG
7. PL
8. PS
9. QINTO
10. QGSUM
11. QLSUM
12. SIGC
13. SIGT
14. T(N)
15. TAU
16. TG
17. THET
18. TI
19. -TIME
20. TL
21. TO
22. TS
23. VOLL
24. VOLG

Appendix C

COMPUTER PRINTOUT

```
// JOB
// DUP
// FOR REDHT
*ONE WORD INTEGERS
*PUNCH
*LIST ALL
*IOCS(CARD),1443 PRINTER,TYPEWRITER,DISK,KEYBOARD)
*NON PROCESS PROGRAM
      REAL  KK,KP,MTOT,L,MG,ML,LT(25),MGG,MR
      DIMENSION  T(1000),TT(1000), HFT(25),TLT(25),PLT(25),VFT(25),
     *VGT(25),        TTT(30),PBT(30) ,H(1000) ,X(4,5),S(5)
      DATA HFT/         210.7,  216.6,  222.3,  227.9,  233.8,  239.6,
     * 245.7,  251.9,  258.2,  264.6,  271.1,  278.0,  285.2,  292.7,
     * 300.2,  308.4,  317.5,  327.5,  339.2,  353.5/
      DATA  TLT/        10.,    20.,    30.,    40.,    50.,    60.,
     * 70.,    80.,    90.,    100.,   110.,    120.,   130.,    140.,
     * 150.,   160.,   170.,   180.,   190.,   200./
      DATA  PLT/        45.85,  55.00,  65.70,  77.80,  91.50,  106.9,
     * 124.3,  143.6,  165.0,  188.7,  214.8,  243.4,  274.5,  308.4,
     * 345.4,  385.0,  426.0,  473.2,  523.4,  575.0/
      DATA  VFT/        .02930, .02970, .03011, .03055, .03101, .03150,
     * .03209, .03269, .03329, .03390, .03452, .03532, .03612, .03702,
     * .03817,  .03962, .04132, .04367,0.04712, .0521/
      DATA  VGT/        2.30,   1.93,   1.60,   1.33,   1.14,   .984,
     * .854,   .745,   .643,    .558,  .487,   .426,   .370,   .320,
     * .278,   .240,   .208,   .180,   .149,   .113/
      DATA  LT/         0.,     0.,     0.,     0.,     0.,     152.6,
     * 148.7,  144.5,  140.1,  135.6,  130.8,  125.8,  120.2,  114.3,
     * 108.2,  99.4,   91.1,   80.1,   65.4,   44.8/
      DATA  TTT/        0.,     50.,    100.,   150.,   200.,   250.,
     * 300.,   350.,   400.,   450.,   500.,   550.,   600.,   650.,
     * 700.,   750.,   800.,   850.,   900.,   950.,   1000.,  1050.,
     * 1100.,  1150.,  1450.     /
      DATA  PBT/        0.,     1055.,  1050.,  1048.,  1044.,  1042.,
     * 1036.,  1032.,  1024.,  1017.,  1006.,  982.,   972.,   950.,
     * 916.,   870.,   804.,   750.,   678.,   605.,   540.,   467.,
     * 400.,   360.,    50./
      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
    1 READ  (5,100)  C,EI,EO,G,KK,KP,RHO,RP,SIG,FLIQ
    2 READ (5,101)  A,CG,DELTA,HGT,HF1,MTOT,NEL,PR,RTANK,THICK
      READ  (5,100)  TF,HE,PRL,EM
      SIG=SIG/1000000.
      WRITE  (6,105)
      WRITE  (6,102) C,EI,EO,G,KK,KP,RHO,RP,SIG ,FLIQ
      WRITE  (6,106)
      WRITE  (6,103) A,CG,DELTA,HGT,HF1,MTOT,NEL,PR,RTANK,THICK,TE,HE,
     *PRL,EM
      WRITE  (6,104)
```

```
      TIME=DELTA
  100 FORMAT  (10F8.0)
  101 FORMAT  (6F8.0,I8,3F8.0)
  102 FORMAT  (8F8.3,F16.12,F4.0)
  103 FORMAT  (2F8.4,F8.1,3F8.2,I8,2F8.3,F8.4,F8.0,F8.2,F8.3,F8.4)
  104 FORMAT  (1H )
  105 FORMAT ('        C       EI       EO        G        K       KP       RHO
     *      RP                SIG FLQ')
    5 FORMAT ('        A       CG    DELTA      HGT      HF1     MTOT      NEL
     *      PR    RTANK    THICK       TE       HE      PRL       EM')
  110 FORMAT (1H1)
  111 FORMAT ('    NEL   TIME     HF     MG       ML     PL      QINTO      QGSU
     *M      QLSUM      T  THET    NG     TL   VOLL   VOLG     TI     T0')
  112 FORMAT (I5,F7.2,2F7.2,F8.2,F6.1, F10.0,F10.5,F10.2,F6.1,F6.2,I4,
     *F7.2,2F6.2,2F6.0)
  113 FORMAT  ('    BURST TABLE LIMITS ELEMENT ',I6)
  114 FORMAT  (' T ',F7.2,'(',I4,')  PL',F6.2,'  PB',F6.0,' TIME',F7.0)
  120 FORMAT(1H ,  ' TIME       MR     PS       SIGC       SIGT        TAU
     *      TG     PL       ML       MG    T(1) THETA   QINTO    QGSUM   DMG    L
     **')
  121 FORMAT  (F6.0,F8.2,F6.2,4F10.0,F6.1,2F8.2,F7.1,F6.2,F7.0,F6.2,
     *F7.0,F5.0)
  122 FORMAT  ('  LIQUID ZERO, ML= ',F8.2)
  123 FORMAT  (10F10.2)
  124 FORMAT  ('        DAO        CON     T(N-1)          QG          QL
     *  D     T(N+1)          TT          TL')
    3 THICK=THICK/12.
      DANG=3.14/NEL
      AEL  =(RTANK+.5*THICK)*DANG
      DA   =DELTA/3600.*AEL
      CRV  =C*RHO*THICK*AEL
      VOL  =3.14*RTANK*RTANK
      CON  =KK*THICK*DELTA/(AEL*3600.)
      QLSUM=0.
      QGSUM=0.
      FLAG=0.
      FLG=0.
      DO  6  J=1,20
      IF  (HFT(J)-HF1)  6,6,7
    6 CONTINUE
    7 H1=HFT(J-2)
      H2=HFT(J-1)
      H3=HFT(J)
      H4=HFT(J+1)
      J=J-2
      DO 77 I=1,4
      X(I,1)=TLT(J)
      X(I,2)=PLT(J)
```

```
      X(I,3)=VFT(J)
      X(I,4)=VGT(J)
      X(I,5)=LT(J)
   77 J=J+1
      DO 78  J=1,5
      XV1=XLAGR(H1,H2,H3,HF1,X(1,J),X(2,J),X(3,J))
      XV2=XLAGR(H2,H3,H4,HF1,X(2,J),X(3,J),X(4,J))
   78 S(J)=(XV1+XV2)*.5
      TL=S(1)
      PL=S(2)
      VF=S(3)
      VG=S(4)
      L =S(5)
      HG=HF1+L
      HF=HF1
      TG=TL
      I=NEL+1
      NPRNT=40
      DO  8  J=1,I
      T(J)=TL
    8 TT(J)=TL
    9 IF  (MTOT-ML)  408,408,409
  408 ML=MTOT
      GO TO 402
  409 ML=(VOL-MTOT*VG)/(VF-VG)
  402 IF  (ML)  12,12,11
   11 MG=(VOL-MTOT*VF)/(VG-VF)
      IF  (TIME-DELTA)  800,800,801
  800 MGG=MG
      HGG=HG
  801 VOLL=ML*VF
      VOLG=MG*VG
      GO TO 13
    2 WRITE  (6,122)  ML
      PAUSE
      GO TO 1
   13 V=6.28*VOLG/(VOLG+VOLL)
      Y=V/3.
  214 W=Y-SIN(Y)
      IF  (IFIX((V-W)*1000.))  312,313,312
  312 CALL DATSW  (11,K11)
      GO TO (310,311),K11
  310 WRITE  (6,102)  V,Y,W
  311 Y=Y+(V-W)*.2
      GO TO 214
  313 THET=.5*Y
      THE=DEG(Y)*.5
      DO  27  N=I,NEL
```

```
      IF  (N-1)  16,16,17
   16 XX=TT(1)
      GO TO 18
   17 XX=TT(N-1)
   18 NG=IFIX(THET/DANG)
      DAO=DANG*(RTANK+THICK)*DELTA/3600.
      QINTO=HE*(TE-T(N))+SIG*(FO*EM*(TE+460.)**4.-EM*(T(N)+460.)**4.)
      IF  (N-NG)  20,20,21
    0 D=RTANK*DANG*DELTA/3600.
      QG=HGT*(T(N)-TL)+SIG*EI*((T(N)+460.)**4.-(TL+460.)**4.)
      QGSUM=QGSUM+2.*D*QG
      T(N)=(DAO*QINTO+CON*(XX-T(N))+CON*(T(N+1)-T(N))-QG*D+CPV*T(N))/CPV
      QL=0.
      GO TO 22
   21 IF  (FLAG)  220,220,224
  224 QL=QINTO
      GO TO 321
  220 QL=4000.*(T(N)-TL)
  321 QLSUM=QLSUM+2.*D*QL
      T(N)=(DAO*QINTO+CON*(XX-T(N))+CON*(T(N+1)-T(N))-QL*D+CPV*T(N))/CPV
      QG=0.
   22 TI=T(N)-QINTO*THICK/(6.*KK)
      IF  (TI-TL)  222,223,223
  222 TI=TL
  223 GO TO (228,229),K11
  228 WRITE  (6,104)
      WRITE  (6,124)
      WRITE  (6,123) DAO,CON,XX,QG,QL,D,T(N+1) ,TT(N)  ,TL
      WRITE  (6,111)
  229 TO=3.*T(N)-2.*TI
      NDL=IFIX(DELTA)
      NTIM=IFIX(TIME)
      IF (NTIM/(NDL*60)*NDL*60-NTIM) 241,240,241
    0 JJJ=2
      GO TO 242
  241 CALL DATSW  (8,JJJ)
  242 GO TO (27,26),JJJ
   26 IF  (NPRNT-40)  24,23,23
   23 WRITE  (6,110)
      WRITE  (6,111)
      WRITE  (6,104)
      NPRNT=0.
   24 WRITE  (6,112) N  ,TIME,HF,MG,ML,PL,QINTO,QGSUM,QLSUM,T(N),TIME,NG
     *,TL,VOLL,VOLG,TI,TO
      NPRNT=NPRNT+1
   27 CONTINUE
      DO 227  J=1,NEL
  227 H(J)=T(J)
```

```
      T(NEL+1)=T(NEL)
      IF  (TIME-DELTA)  30,30,31
   30 DML=0.
      DMG=0.
      GO TO 32
   31 DMG=(MG-MGG)*3600./DELTA
      DHG=HGG-HG
      HGG=HG
      MGG=MG
      DML=-DMG
   32 MR=0.
      IF (FLG)  1141,1141,1139
 1139 IF (PL-PRL)  1140,1150,1150
 1140 FLG=0.
 1141 IF  (PL-PR)  136,50,50
   50 FLG=1.
 1150 IF  (FLIQ)  1151,1151,1152
 1152 MR=192000.*CG*A
      MTOT=MTOT-MR*DELTA/3600.
      DHF=(QLSUM+QGSUM-DMG*L*DELTA/3600.-MG*DHG)/(ML+(DML-MR)*DELTA/
     *3600.)
      GO TO 1153
 1151 PS=PL*(2./(KP+1.))**(KP/(KP-1.))
      TS=(TL+460.)*(2./(KP+1.))
      VC=RP*TS/(PS*144.)
      UC=SQRT(KP*G*RP*TS)
      MR=(CG*A*UC)/VC *60.
      MTOT=MTOT-MR*DELTA/3600.
  136 DHF=(QLSUM+QGSUM-(DMG+MR)*L*DELTA/3600. -MG*DHG)/(ML+(DML-MR)*
     *DELTA/3600.)
 1153 HF=HF+DHF
      GO TO (700,701),K11
  700 WRITE (6,123) QLSUM,QGSUM,DMG,MR,L,ML,DML,MR,HF
    1 DO 137 J=1,20
      IF  (HFT(J)-HF)  137,137,37
  137 CONTINUE
   37 H1=HFT(J-2)
      H2=HFT(J-1)
      H3=HFT(J)
      H4=HFT(J+1)
      J=J-2
      DO 177 I=1,4
      X(I,1)=TLT(J)
      X(I,2)=PLT(J)
      X(I,3)=VFT(J)
      X(I,4)=VGT(J)
      X(I,5)=LT(J)
  177 J=J+1
```

```
      DO 178 J=1,5
      XV1=XLAGR(H1,H2,H3,HF ,X(1,J),X(2,J),X(3,J))
      XV2=XLAGR(H2,H3,H4,HF ,X(2,J),X(3,J),X(4,J))
  178 S(J)=  (XV1+XV2)*.5
      TG=TL
      TL=S(1)
      PL=S(2)
      VF=S(3)
      VG=S(4)
      L =S(5)
      HG=HF+L
      DO 41  K=1,NEL
      DO  38 J=1,25
      IF  (TTT(J)-T(K))  38,38,39
   38 CONTINUE
      WRITE  (6,113)  K
   39 Z= (T(K)-TTT(J-1))/(TTT(J)-TTT(J-1))
      PB=Z*(PBT(J)-PBT(J-1))+PBT(J-1)
      IF  (PL-(PB+14.7))  41,40,40
   40 WRITE (6,114)  T(K),K,PL,PB,TIME
      WRITE  (6,110)
      PAUSE
      CALL DATSW  (9,KFX)
      GO TO (41,1),KEX
   41 CONTINUE
      SIGC=(PL-14.7)*RTANK/ THICK
      SIGT=.5*SIGC
      TAU=.25*SIGC
      CALL DATSW  (12,K12)
      GO TO (560,60),K12
  560 GO TO (561,60),JJJ
   60 WRITE  (6,120)
      WRITE  (6,121)  TIME,MR,PS,SIGC,SIGT,TAU,TG,PL,ML,MG,T(1),THE,
     *QINTO,QGSUM,DMG,L
   61 GO TO (61,62),JJJ
   62 WRITE (6,110)
   61 TIME=TIME+DELTA
      QLSUM=0.
      QGSUM=0.
      DO  28  J=1,NEL
   28 TT(J)=T(J)
      CALL DATSW  (10,KST)
      GO TO (1,9),KST
C                  DATSW  8  ELIMINATE ELEMENT PRINT
C                  DATSW  9  CONTINUE AFTER BURST
C                  DATSW 10  START NEW CASE
C                  DATSW 11  PRINT SUB-TOTALS
C                  DATSW 12  PRINT EVERY 60TH POINT
```

Appendix D

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