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Thermal Model Upgrade for the Analysis of Defective Thermal Protection Systems

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Transportation Development Centre
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Transport Canada

by

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Kingston, Ontario

January 2005

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Since some of the accepted measures in the industry are imperial, metric measures are not always used in this report.

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16. Abstract

This report summarizes the development of an upgraded tank-car thermal model specifically developed to model fire effects on dangerous goods tank cars that have defects in their thermal protection systems.

The upgraded model, called Insulation Defect Analyzer IDA 2.1, has been partially validated with recent fire test data by Birk et al. (TP 14366E). In these tests, 500-gal. ASME code propane tanks were tested with simulated thermal protection defects. The IDA code was able to predict the PRV pop time, tank pressure, tank fill, tank wall and jacket temperatures and time to failure in reasonable agreement with fire test results.

The model was then used to study thermal protection defects on 112J type tank-cars. It predicted that even small defects can lead to tank rupture if the defect is located in the tank vapour space and if the fire is severe and engulfs the tank. The model also showed that the condition of the remaining thermal protection system plays an important role in the response of the tank. If the overall thermal protection system is in good condition, this slows the rate at which the liquid level drops in the tank, thus delaying failure. If the overall condition of the tank is not very good, then the liquid level drops more rapidly, exposing thermal protection defects earlier and leading to earlier failure.

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16. Résumé

Le rapport résume les travaux de développement d'un modèle thermique de wagon-citerne amélioré, expressément conçu pour l'étude des effets du feu sur des wagons-citernes à marchandises dangereuses présentant des défauts dans leurs systèmes de protection thermique.

Le modèle amélioré, appelé IDA 2.1, (IDA pour *Insulation Defect Analyzer*), a été partiellement validé par les résultats d'essais au feu menés récemment par Birk et coll. (TP 14366E). Ces essais portaient sur des citernes à propane de 500 gallons conformes au code de l'ASME, dont on avait endommagé la protection thermique. La pression d'ouverture de la soupape de sûreté, la pression interne de la citerne, le niveau de remplissage de la citerne, la température de la paroi de la citerne et de la jaquette en acier, et le temps avant défaillance prévus par le code IDA concordaient raisonnablement avec les résultats des essais au feu.

Le modèle a ensuite été utilisé pour étudier les défauts de la protection thermique de wagons-citernes de type 112J. Les résultats ont révélé que même des défauts de petites dimensions peuvent entraîner la rupture de la citerne lorsque le défaut est situé au sommet de la zone de phase gazeuse et lorsque le feu est intense et qu'il enveloppe la citerne. Le modèle a également révélé que l'état du reste de la protection thermique influe considérablement sur la réaction de la citerne. Ainsi, lorsque le système de protection est globalement en bon état, le niveau de liquide baisse moins vite, ce qui retarde d'autant la rupture. Inversement, si la citerne dans son ensemble est dans un mauvais état, le niveau de liquide baisse plus rapidement, les défauts de protection thermique sont exposés plus tôt au feu, ce qui devance le moment de la rupture.

17. Mots clés Marchandises dangereuses, wagon-citerne, modèle thermique, effets du feu, défauts de protection thermique, modèle d'analyse de défauts d'isolation, IDA	18. Diffusion Le Centre de développement des transports dispose d'un nombre limité d'exemplaires.

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Executive Summary

This report summarizes the development of an upgraded tank-car thermal model specifically developed to model fire effects on dangerous goods tank cars that have defects in their thermal protection systems. This model is needed because it is known that some dangerous goods tank-cars have defects in their thermal protection systems.

Thermal protection systems are designed to protect tank-cars from accidental fire impingement. These systems are designed to stop thermal ruptures when the tank-cars are exposed to an engulfing fire for 100 minutes or a torching fire for 30 minutes. The systems of interest here consist of thermal insulation covered with a steel jacket. Defects may form if the blanket slips, or tears and drops down due to vehicle motion, or is crushed under the jacket. These defects can then lead to an open space between the jacket and the tank-car wall, and heat can be transferred across this air gap by thermal radiation and convection. The steel jacket does provide some protection by itself as it acts as a thermal radiation shield. This reduces the heat flux from the fire by about a factor of two, compared to the non-thermally protected tank-car. In a tank-car with intact thermal protection, the heat flux is reduced by about a factor of 10 relative to the non-thermally protected tank-car. In the worst case, the jacket is pressed hard against the tank wall and provides no protection at all.

Limited field surveys of tank-cars have shown that some older tank-cars may have significant defects in their thermal protection systems. The question now is what level of defect is acceptable from a safety standpoint? Work has been under way to answer this question. The work has included computer modelling together with actual fire testing of tank-car thermal protection systems and reduced scale propane tanks with thermal protection defects.

The computer code described in this report is called Insulation Defect Analyzer (IDA) 2.1. The code is based on the thermal model by Birk [1] and uses some methods very similar to those of AFFTAC by Johnson [2]. Some methods used are very different than AFFTAC, including full 3D tank shape, two-node lading thermal model, cycling pressure relief valve (PRV), high-temperature stress-rupture failure prediction, and others.

The upgraded code has been partially validated with recent fire test data by Birk et al. [3]. In these tests, 500 gal. ASME code propane tanks were tested with simulated thermal protection defects. These tests showed that tanks with even 8% defect (i.e., 8% of tank surface area) and defect lengths along the tank axis of about 70% of the tank diameter could lead to rupture within the scaled time allotted. The IDA code was able to predict the PRV pop time, tank pressure, tank fill, tank wall and jacket temperatures, and time to failure in reasonable agreement with fire test results. The IDA model was also able to predict the failure time and overall behaviour in reasonable agreement with the full-scale tank-car test RAX 201 of Townsend et al. [4]. In general, the IDA code predictions for tank failure are conservative by 3-5 minutes (i.e., IDA predicts failure early). This may be

due to the fact that the model assumes the fire is 100% on at time = 0 and in the fire tests it takes several minutes to get the fire up to full intensity.

The model was used to study thermal protection defects on 112J type tank-cars. The model predicted that even small defects can lead to tank rupture if the defect is located at the top of the tank vapour space and if the fire is severe and it fully engulfs the tank. The model also showed that the condition of the remaining thermal protection system plays an important role in the response of the tank. If the overall thermal protection system is in good condition, this slows the rate at which the liquid level drops in the tank, thus delaying failure. If the overall condition of the tank is not very good, then the liquid level drops more rapidly, exposing thermal protection defects in the vapour space earlier and leading to earlier failure. The problem is that we do not know the overall condition of the thermal protection system on a tank with local defects in the thermal protection system.

The following conclusions have been made:

i) The IDA 2.1 code has been reasonably validated against the summer 2004 fire testing of a 500 gal. propane tank (both baseline and with thermal protection defects).
ii) The IDA 2.1 code is in reasonable agreement with the RAX 201 fire test results of a full-scale unprotected rail tank-car.
iii) There are some differences between the IDA 2.1 model and test results. IDA 2.1 tends to predict a more rapid increase in wall temperatures, which leads to failure prediction a few minutes earlier than observed in tests. This can partly be explained by how the fire is modelled. Real fires take some time to build up whereas in IDA the fire is on 100% at time = 0.
iv) The model appears to be reasonable and conservative in the prediction of tank failure.

The IDA 2.1 program has not been fully validated and therefore it should be used with caution.

The following conclusions have been made based on the modelling reported herein. It has been assumed that the critical thermal protection defect size is 1.2 m measured along the tank car (112J) axis by 0.4 m wide as determined from the fire testing conducted by Birk et al. [3].

i) A critical thermal protection defect can lead to tank rupture if it is located in the tank-car vapour space during a fire engulfment accident.
ii) The failure of a tank-car with thermal protection defects depends not only on the size and location of defects, but also on the quality of the remainder of the thermal protection system that is not defective (including all direct condition links in the tank structure). The better thermally protected the tank is, the more capable it is of surviving with local thermal protection defects. This is because the overall thermal protection system determines how fast the liquid level will drop when the tank is exposed to fire.

iii) The total allowable defect area is very strongly affected by the area average thermal conduction properties (i.e., k/w where k = thermal conductivity and w = insulation thickness) of the tank thermal protection insulation during fire conditions. It is estimated that this value of thermal conductivity is in the range of 0.15 to 0.3 W/mK for high-temperature ceramic blanket insulation under fire exposure conditions.

iv) A tank with 13 mm ceramic blanket thermal protection with an area average thermal conductivity of 0.15 W/mK (at fire conditions) can probably allow up to 8 to 9% of its surface to be defective of thermal protection. This assumes that there is at least one critical defect in the vapour space. This also assumes that the PRV has a flow capacity greater than about 5000 scfm at 120% of the PRV set pressure (280.5 psig assumed here).

v) A tank with 13 mm ceramic blanket thermal protection with area average thermal conductivity of 0.20 W/mK (at fire conditions) can probably allow up to 4% of its surface to be defective of thermal protection. This assumes that there is at least one critical defect in the vapour space. This also assumes that the PRV has a flow capacity greater than about 4000 scfm at 120% of the PRV set pressure (280.5 psig assumed here).

vi) A tank with 13 mm ceramic blanket thermal protection with area average thermal conductivity of 0.30 W/mK (at fire conditions) cannot allow any critical defects (i.e., longer than 1.2 m along tank axis by 0.4 m wide). This effective thermal conductivity is the maximum allowable for a 13 mm blanket that meets the original plate test standard for thermal protection systems. If a tank has this average thermal conductivity, then a 3500 scfm PRV is probably too small for that tank.

vii) If there are no defects larger than 1.2 m x 0.4 m, then more defect area may be acceptable, but this should be determined on a case-by-case basis by running the IDA 2.1 code for the specific tank. For this case, insulation samples should be taken so actual *k* values can be measured. At least 10 samples should be taken so that a truly representative average *k* can be determined.

viii) 112J type tank cars equipped with 3500 scfm PRV should not be allowed to have any defects unless the overall thermal protection properties can be defined.

The reader is reminded that this study did not consider the following:

- end failures
- defective PRVs
- defects in primary shell
- corrosion
- impact damage
- torching fires
- rolled tanks
- hard contact between the jacket and tank shell.

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4. Townsend, W., Anderson, C.E., Zook, J., Cowgill, G., 1974, *Comparison of Thermally Coated and Uninsulated Rail Tank-Cars Filled with LPG Subjected to a Fire Environment,* Rep. FRA-OR&D 75-32, Federal Railroad Administration, U.S. Department of Transportation.

Sommaire

Le rapport résume les travaux de développement d'un modèle thermique de wagon-citerne amélioré, expressément conçu pour l'étude des effets du feu sur des wagons-citernes à marchandises dangereuses présentant des défauts dans leurs systèmes de protection thermique. Ce modèle est nécessaire car on sait que certains wagons-citernes à marchandises dangereuses comportent des défauts d'isolation.

Les systèmes de protection thermique sont conçus pour protéger les wagons-citernes contre l'effet de flammes en cas d'accident. Ces systèmes doivent en principe protéger la citerne de la rupture pendant 100 minutes lorsqu'elle est soumise à un feu en nappe, ou 30 minutes, lorsqu'elle est soumise à une flamme de chalumeau. Le système qui nous intéresse ici est formé d'un matelas isolant recouvert d'une jaquette en acier. Des défauts peuvent apparaître dans l'isolant, s'il glisse ou se déchire et s'abaisse sous l'effet des mouvements du véhicule, ou s'il est écrasé par la jaquette. Il peut alors se créer un vide entre la jaquette et la paroi de la citerne, et les phénomènes de rayonnement et de convection peuvent entraîner un transfert de chaleur dans cette lame d'air. La jaquette en acier comme telle assure une certaine protection, agissant comme un bouclier thermique. De fait, même en l'absence d'isolant, elle réduit de moitié, environ, le flux thermique provenant du feu, comparativement à un wagon-citerne dénué de toute protection thermique. Par ailleurs, un wagon-citerne dont la protection thermique est intacte est soumis à un flux thermique environ 10 fois plus faible que s'il n'était doté d'aucune protection thermique. Dans le pire des cas, la jaquette est complètement pressée contre la paroi et elle n'assure aucune protection.

Des essais limités sur le terrain ont indiqué que certains wagons-citernes anciens peuvent effectivement comporter des défauts d'isolation importants. La question qui se pose maintenant est de savoir jusqu'à quel point un défaut peut être acceptable du point de vue de la sécurité. Des travaux ont été entrepris pour répondre à cette question. Ceux-ci ont pris la forme d'une modélisation informatique et d'essais réels de comportement au feu de systèmes de protection thermique de wagons-citernes et de citernes de propane à échelle réduite dont la protection thermique avait été endommagée.

Le code informatique, décrit dans le rapport, est appelé IDA 2.1 (IDA pour *Insulation Defect Analyzer*. Ce code, qui est inspiré du modèle thermique de Birk [1], utilise des méthodes dont certaines sont très semblables à celles du modèle AFFTAC de Johnson [2]. Mais d'autres sont très différentes. Par exemple : forme de citerne tridimensionnelle, modèle thermique de remplissage à deux nœuds, soupape de sûreté à cycles d'ouverture et de fermeture variables, prévision de la rupture par fluage à haute température.

Le code amélioré a été partiellement validé par les résultats d'essais au feu menés récemment par Birk et coll. [3]. Ces essais portaient sur des citernes à propane de 500 gallons conformes au code de l'ASME, dont on avait endommagé la protection thermique. Ils ont révélé que des défauts couvrant aussi peu que 8 p. 100 de la surface de la citerne et des défauts d'une longueur dans l'axe représentant environ 70 p . 100 du

diamètre de la citerne pouvaient mener à la rupture en-deçà du délai alloué, toutes proportions gardées. La pression d'ouverture de la soupape de sûreté, la pression interne de la citerne, le niveau de remplissage de la citerne, la température de la paroi de la citerne et de la jaquette en acier, et le temps avant défaillance prévus par le code IDA concordaient raisonnablement avec les résultats des essais au feu. Les prévisions du modèle IDA se sont également révélées raisonnablement conformes aux résultats de l'essai RAX 201 en vraie grandeur de Townsend et coll. [4] pour ce qui est du temps avant défaillance et du comportement général de la citerne. En général, le code IDA prévoit des temps avant défaillance de 3 à 5 minutes trop courts. Cela peut être dû au fait que le modèle suppose un feu à 100 p. 100 dès le moment = 0, tandis que lors des essais au feu, le feu prend plusieurs minutes pour atteindre sa pleine intensité.

Le modèle a ensuite été utilisé pour étudier les défauts de la protection thermique de wagons-citernes de type 112J. Les résultats ont révélé que même des défauts de petites dimensions peuvent entraîner la rupture de la citerne lorsque le défaut est situé au sommet de la zone de phase gazeuse et lorsque le feu est intense et qu'il enveloppe la citerne. Le modèle a également révélé que l'état du reste de la protection thermique influe considérablement sur la réaction de la citerne. Ainsi, lorsque le système de protection est globalement en bon état, le niveau de liquide baisse moins vite, ce qui retarde d'autant la rupture. Inversement, si la citerne dans son ensemble est dans un mauvais état, le niveau de liquide baisse plus rapidement, les défauts de protection thermique sont exposés plus tôt au feu, ce qui devance le moment de la rupture. Le problème est que la présence de défauts localisés dans la protection thermique d'une citerne ne dit rien sur l'état général de cette protection.

Voici les conclusions tirées des travaux :

i) Les essais au feu d'une citerne de propane de 500 gallons (dans deux configurations : citerne de référence et citerne avec protection thermique défectueuse) menés au cours de l'été 2004 ont permis de valider de façon satisfaisante le code IDA 2.1.
ii) Les résultats obtenus avec le code IDA 2.1 concordent raisonnablement avec les résultats de l'essai au feu RAX 201 d'un wagon-citerne en vraie grandeur sans protection thermique.
iii) Certains écarts ont été constatés entre la modélisation IDA 2.1 et les résultats d'essais. Ainsi, le code IDA 2.1 a tendance à surestimer la vitesse de montée en température des parois, et, par conséquent, à devancer de quelques minutes le moment de la rupture, par rapport aux résultats des essais. Cet écart peut s'expliquer en partie par la façon dont le feu est modélisé. En effet, un feu réel n'atteint pas instantanément sa pleine intensité, tandis que dans le modèle IDA, le feu est à 100 p. 100 dès le moment 0.
iv) Le modèle semble prévoir de façon raisonnable et prudente la rupture de la citerne.

Le programme IDA 2.1 n'a pas été complètement validé et il est donc sujet à caution.

Les conclusions ci-après ont été tirées des travaux de modélisation exposés dans le rapport. Les résultats des essais au feu menés par Birk et coll. [3] ont permis de poser comme hypothèse qu'un défaut d'isolation est critique lorsqu'il mesure 1,2 m dans l'axe du wagon-citerne (112J) sur 0,4 m de largeur.

i) Un défaut d'isolation critique, s'il est situé dans la zone de phase gazeuse du wagon-citerne, peut entraîner la rupture de la citerne lorsque celle-ci est soumise à des flammes enveloppantes lors d'un accident.

ii) La rupture d'un wagon-citerne qui comporte des défauts d'isolation ne dépend pas seulement de l'étendue et de l'emplacement des défauts, mais aussi de la qualité du reste de la protection (y compris tous les contacts directs entre la jaquette et le récipient intérieur de la citerne). Meilleure est la protection thermique de la citerne, plus elle est capable de survivre, malgré des défauts localisés. C'est que l'ensemble de la protection thermique détermine la vitesse à laquelle le niveau de liquide baisse dans la citerne lorsqu'elle est exposée au feu.

iii) La superficie totale admissible des défauts d'isolation est très fortement tributaire des propriétés de conduction thermique moyenne de la surface (c.-à-d., k/w, où k = conductivité thermique et w = épaisseur de l'isolant) de l'isolant thermique de la citerne lorsqu'elle est exposée à des flammes. On estime cette valeur de conductivité thermique entre 0,15 et 0,3 W/mK dans le cas d'un matelas isolant en fibre céramique haute température exposé au feu.

iv) Dans une citerne protégée par un matelas isolant en fibre céramique de 13 mm d'épaisseur dont la surface présente une conductivité thermique moyenne de 0,15 W/mK (dans des conditions d'incendie), on peut probablement admettre des défauts de protection thermique couvrant de 8 p. 100 à 9 p. 100 de la surface de celle-ci. Cela en supposant qu'il y a au moins un défaut critique dans la zone de la phase gazeuse. On suppose également que la soupape de sûreté a une capacité d'écoulement supérieure à environ 5 000 pi^3/min (standard), à 120 p. 100 de sa pression d'ouverture (présumée à 280,5 lb/po^2).

v) Dans une citerne protégée par un matelas isolant en fibre céramique de 13 mm d'épaisseur dont la surface présente une conductivité thermique moyenne de 0,20 W/mK (dans des conditions d'incendie), on peut probablement admettre des défauts couvrant 4 p. 100 de la surface de celle-ci. Cela en supposant qu'il y a au moins un défaut critique dans la zone de la phase gazeuse. On suppose également que la soupape de sûreté a une capacité d'écoulement supérieure à environ 4 000 pi^3/min (standard), à 120 p. 100 de sa pression d'ouverture (présumée à 280,5 lb/po^2).

vi) Dans une citerne protégée par un matelas isolant en fibre céramique de 13 mm d'épaisseur dont la surface présente une conductivité thermique moyenne de 0,30 W/mK (dans des conditions d'incendie), on ne peut admettre aucun défaut critique (c.-à-d. mesurant plus de 1,2 m dans l'axe de la citerne sur 0,4 m de largeur). Cette conductivité thermique efficace est la valeur maximale admissible pour un matelas isolant de 13 mm d'épaisseur qui respecte les paramètres des essais standard sur plaque des systèmes de protection thermique. Si une citerne présente cette conductivité thermique moyenne, une soupape de sûreté dont la capacité d'écoulement est de 3 500 pi^3/min (standard) est probablement inadéquate (la capacité d'écoulement est trop faible).

vii) Si la citerne ne comporte aucun défaut de plus de 1,2 m x 0,4 m, on peut admettre des défauts couvrant une plus grande surface, mais la surface maximale doit être déterminée au cas par cas, en appliquant le code IDA 2.1 à la citerne en question. Il faut alors prélever des éprouvettes d'isolant afin de mesurer les valeurs *k* réelles. Au moins 10 éprouvettes doivent être prélevées de façon que la valeur *k* soit véritablement représentative.

viii) Aucun défaut ne doit être admis dans les wagons-citernes de type 112J équipés de soupapes de sûreté de 3 500 pi^3/min (standard), à moins que l'on puisse caractériser avec précision la protection thermique globale.

Le lecteur doit se rappeler que les éléments suivants n'ont pas été pris en compte dans l'étude :

- défaillance des extrémités
- soupapes de sûreté défectueuses
- défauts dans le récipient intérieur
- corrosion
- dommages dus à un impact
- flammes de chalumeau
- citerne renversée
- contact direct entre la jaquette et le récipient intérieur de la citerne

Références

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3. Birk, A.M., Poirier, D., Davison, C., Wakelam, C., 2005, *Tank-Car Thermal Protection Defect Assessment: Fire Tests of 500-Gallon Tanks with Thermal Protection Defects,* TP 14366E, Centre de développement des transports, Transports Canada.

4. Townsend, W., Anderson, C.E., Zook, J., Cowgill, G., 1974, *Comparison of Thermally Coated and Uninsulated Rail Tank-Cars Filled with LPG Subjected to a Fire Environment,* Rep. FRA-OR&D 75-32, Federal Railroad Administration, U.S. Department of Transportation.

Table of Contents

APPENDICES

List of Figures

List of Tables

1. Introduction

This work is a follow on from previous projects where methods were developed to identify thermal protection defects in tank-cars [1]. The defects were analyzed and guidelines were presented on how to assess them [2]. It was then decided by committee that a more systematic and repeatable assessment system was needed. This project was to upgrade the tank thermal model by Birk [3] so that it could be used to analyze a tank with insulation defects for the purpose of defect assessment.

This report is an overview of this computer model. It should be noted that this is a model in development.

1.1 Background

This report describes a computer model of a rail tank-car subjected to accidental fire impingement. The code, called Insulation Defect Analyzer (IDA) 2.1, was developed specifically to model thermal protection system defects. IDA 2.1 can account for the following as a function of time from when a fire is started:

- i) heat addition from a torch or engulfing fire
- ii) effects of thermal protection with or without local degradation
- iii) tank wall temperature in the vapour space and liquid space
- iv) temperature and pressure rise in the propane lading
- v) pressure relief valve (PRV) mass flow
- vi) tank wall material strength
- vii) tank failure

For this project it was necessary to refine the modeling methods to include local insulation defects. Certain enhancements have been included to ensure that predictions are credible and, where there is uncertainty, conservative.

The model used in this work is a hybrid model based on the TANKCAR series of codes by Birk [3]. Many of the methods used are similar to those of the AFFTAC model of Johnson [4].

1.2 Objective

The objective of the work was to upgrade the basic thermal model so that it can account for the effects of defects in tank-car thermal protection systems. Specifically this included the following sub objectives:

- i) Make the code quasi 3D so that it can account for a wide range of possible defect locations and geometries.

ii) Include graphical output features to confirm defect geometries.
iii) Refine lading thermal model to account for liquid temperature stratification and its effect on tank pressure.
iv) Refine lading thermal model to include effects of defect position on tank pressurization (i.e., more rapid pressurization if heating is near liquid level).
v) Refine PRV model to include real-world PRV operating characteristics (i.e., cycling between pop and close pressure).
vi) Refine vapour space and PRV model to account for liquid entrainment into PRV at very high fill levels.
vii) Refine tank failure model to include high-temperature stress rupture.
viii) Ensure that the code runs in a reasonable length of time (< 2 hours for the simulation of a 100 minute fire scenario).

1.3 Scope

The scope of this model development is limited as follows:

i) The code will not include a graphical user interface.
ii) The code will have limited validation due to the lack of data.

This code is used for research purposes and as such is continuously undergoing further development as new data becomes available. As such, it has not been developed for general release.

2. Program Overview

The Insulation Defect Analyzer (IDA) 2.1 code was developed to model rail rank-cars in fires. The code was specifically developed to model thermal protection defects of any shape and location. The objective was to be able to predict time to failure or time to empty for the tank-car and thermal protection system. To do this properly requires that the following be predicted with some accuracy:

i) tank pressure
ii) tank fill
iii) peak vapour space wall temperatures
iv) tank material properties
v) tank stress

All of these must be calculated as a function of the time that the tank is exposed to fire. To do this the program must be able to include the following processes:

i) heat transfer from a fire by thermal radiation and convection
ii) wall heat conduction (including jacket, thermal insulation and primary shell)
iii) vapour space convection and thermal radiation
iv) liquid space convection and boiling (including 2-phase swell during boiling)

v) thermodynamic process in liquid and vapour space, including temperature stratification in the liquid (needed to predict early pressurization)
vi) thermodynamic and transport properties of lading (propane)
vii) PRV activation and flow rates (for vapour and 2-phase)
viii) degradation of wall material properties (stress rupture)

IDA 2.1 was developed to run on a desktop PC in a reasonable length of time. The code solves the time transient response of a tank and its propane lading when exposed to an accidental fire. A P4 3.0 GHz processor can run a typical 100 minute real-time simulation in about 1-2 hours (depending on time step and tank grid resolution). To achieve this fast simulation requires numerous simplifications. As a result, the IDA code is not a full field analysis but is rather a partially empirical zone type analysis. Commercial CFD (computational fluid dynamics) or FE (finite element) codes could be assembled to carry out many of the computations, but the run times would be two or three orders of magnitude longer with little improvement in accuracy.

The basic model dimensions are as follows:

i) 3D tank shape
ii) 2D wall heat conduction
iii) 1D vapour convection
iv) 0D liquid convection
v) 3 zone vapour space radiation model
vi) 2 zone lading thermodynamic model
vii) On-off PRV model
viii) 0D stress model

Appendix A contains additional information about the program and the methods used.

3. Tank 3D Geometry Model

The tank-car geometry has been modeled as a cylinder with 2:1 elliptical or hemi heads.

The tank is divided into steps in the axial direction to create rings. These rings are divided into arcs by angular steps. In this way the tank surface is divided into zones with a rectangular or quadrilateral shape (except at the very ends of the tank, where they are triangular in shape). The grid at one end of the tank is shown in Figure 1.

This 3D shape allows us to input insulation defects of almost any shape and location. The resolution accuracy of the defect is defined by the size of the wall zones. At present the resolution is set to 200 axial steps and 72 angular steps. The upper limit on these is 399 axial steps and 399 angles.

Each of the surface zones created in this way is treated separately for the following:

- i) heat input from fire
- ii) temperature
- iii) material property
- iv) thickness
- v) ultimate tensile strength
- vi) with or without insulation defect

The total heat input to the tank is the sum of the heat inputs to all of the separate zones. An energy balance is taken for each wall zone to determine the wall temperature. Further details on this can be found in Appendix A.

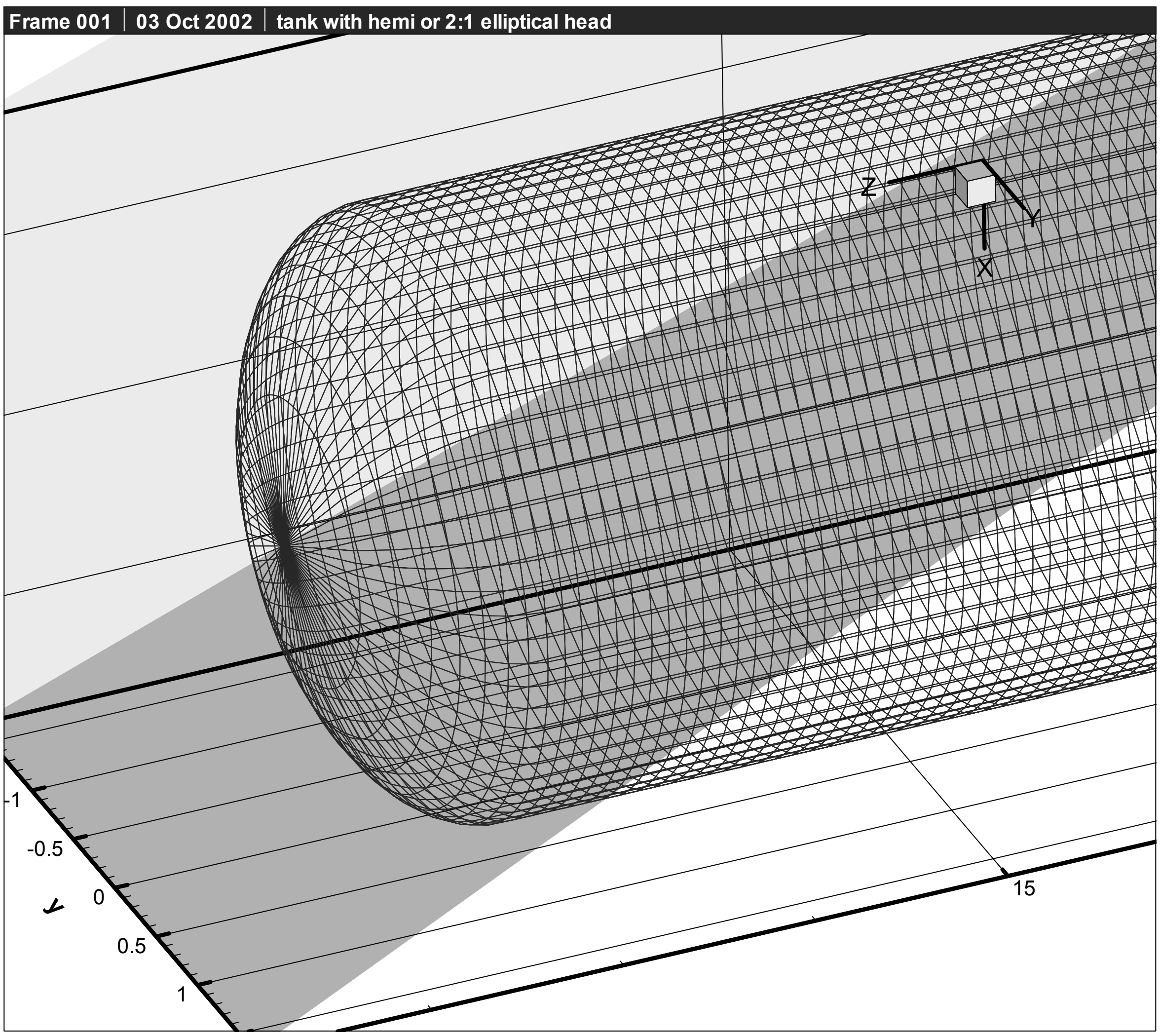

Figure 1 Sample 3D grid of tank

4. Thermal Protection Defects

The tank is assumed to be thermally protected using a blanket of thermal insulation covered with a steel jacket. The case modeled here is the 13 mm blanket of high temperature ceramic fibre insulation. This insulation is then covered by 3 mm thick steel jacket.

If there is an insulation defect, it is assumed the blanket is not present and that heat can radiate directly from the steel jacket to the tank wall. The current model assumes an air gap in defect locations. Convection in the air gap is neglected. Direct, hard contact between the jacket and the tank wall is not modeled specifically in IDA 2.1 but it is basically an unprotected wall.

Defects are located on the tank surface by defining the defect in 3D space as follows:

i) defect x range from x_1 to x_2
ii) defect y range from y_1 to y_2
iii) defect z range from z_1 to z_2

where x, y and z are as shown in Figure 2. Figure 2 shows a sample tank with three defects located (two defects are overlapping). The tank can have multiple defects defined.

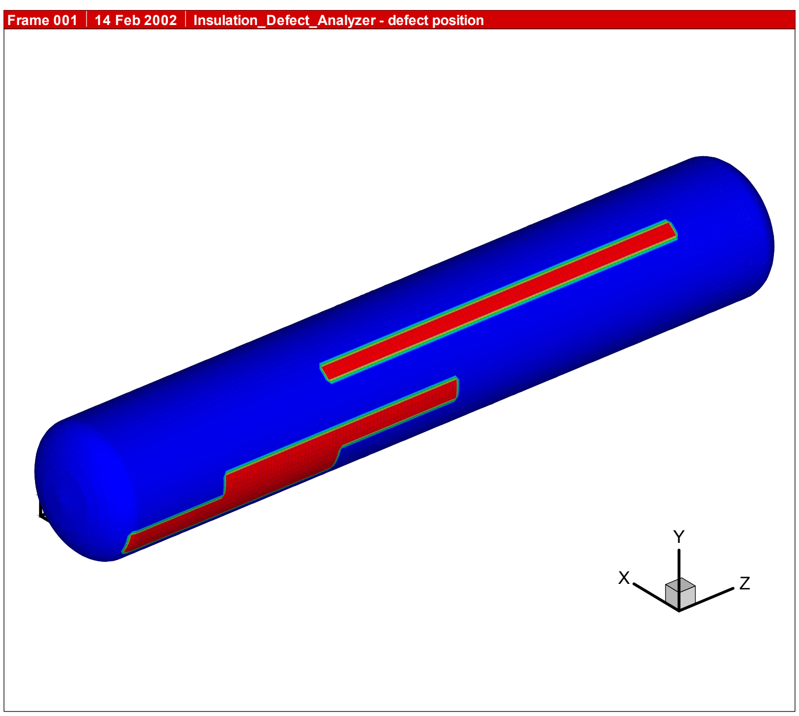

Figure 2 Tank with defects located

5. Failure Analysis

5.1 Failure Models

A key feature of the IDA 2 code is the failure model. There are two failure models in IDA 2.1 as follows:

i) When a factor of safety (FOS) based on hoop stress and material ultimate tensile stress at the peak wall temperature drops below some set value, failure is assumed to take place.
ii) Failure time is calculated based on high temperature stress rupture data.

5.1.1 Failure by FOS

The IDA calculates a factor of safety (FOS) for the tank. This FOS is defined as:

$$FOS = \frac{P_b}{P}$$

where
P_b = calculated burst pressure = $2\sigma_{ult}$ t/D
P = tank pressure
D = tank diameter
t = wall thickness
σ_{ult} = steel ultimate strength at peak wall temperature

In a very simple failure analysis we would define failure when FOS = 1.0. We may choose to set a failure criteria at an FOS greater than 1.0 if we want to account for uncertainties in the analysis. Birk and Cunningham [2] suggested a minimum allowable FOS of 1.4 – 1.6 to include uncertainties in materials, PRV operation and fire conditions.

The determination of a more accurate burst pressure would involve an elastic-plastic-creep analysis that includes wall temperature distribution, details of the defect size and location, and detailed high-temperature material properties of the steel. However, this requires a detailed 3D stress analysis that accounts for plastic deformation and creep. This is beyond the scope of the present program. Such an analysis is best calculated by a commercial finite element program. In a recent collaboration between A.M. Birk and Battelle Memorial Institute, a full 3D elastic-plastic-creep analysis by finite elements produced similar failure times to the above-mentioned FOS analysis. This was true for severe fire heating cases.

The FOS failure analysis in IDA is based on the following simple assumptions:

i) The tank cylinder is assumed to fail in an area where the wall is heated due to an insulation deficiency.

ii) The stress in the defect area is assumed to be the nominal hoop stress for the tank-car cylinder.

iii) The tank wall material strength in the defect area will be decreased as a result of the calculated peak wall temperature.

Figure 3 shows the ultimate strength of TC 128 steel as a function of temperature. The data shown is for both the minimum allowable ultimate strength (550 MPa at ambient temperature) and a more typical ultimate strength (620 MPa at ambient). This data is used in IDA to predict failure using the FOS method.

This failure criteria does not account for any other causes that can contribute to failure, such as corrosion, dents and other impact damage, fatigue cracks, stress raisers, etc. For this reason a factor of safety greater than 1.0 should be used.

The stress in the head due to internal pressure is a function of the size and shape of the head, and the geometry of the head welds. In a classical analysis, the head stress is maximum in the knuckle region where the head curvature is greatest. Predicting failure in the head is a difficult process. IDA does not attempt to predict head failure.

This FOS model of failure is believed to be conservative if the heating is very severe and widespread (unprotected tank-car in a fully engulfing fire). However, if the fire heating is localized and less severe (like in a tank with local thermal protection defects), this increases the failure time and the FOS approach may not be conservative. In this case high-temperature stress rupture analysis is needed.

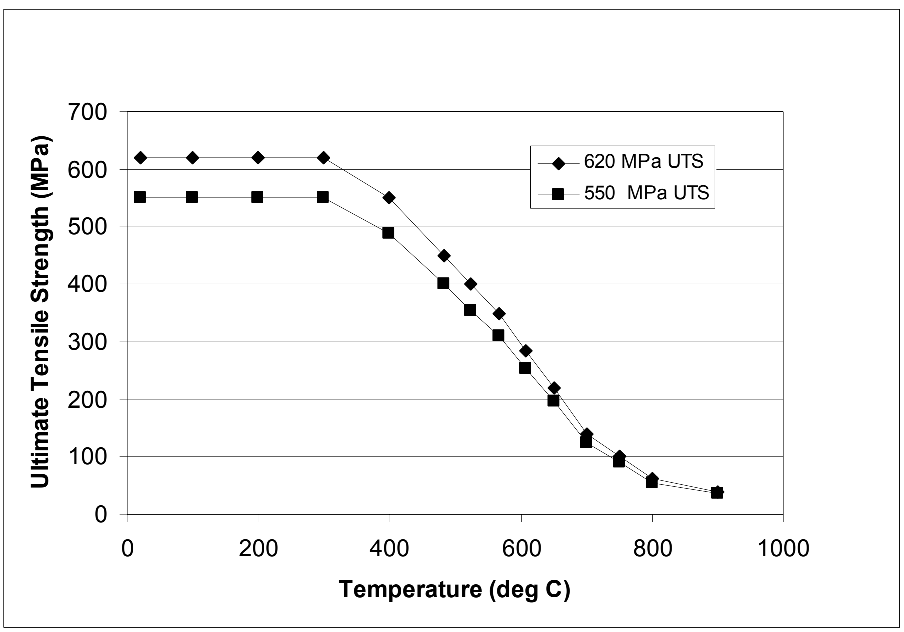

Figure 3 Estimated ultimate tensile stress for TC 128B as a function of temperature

5.1.2 Failure Analysis by High-Temperature Stress Rupture (SR)

The SR failure is based on a simple stress analysis combined with high-temperature stress rupture data for the tank-car steel. The basic method of calculating stress rupture damage is described in [5].

This method appears to be valid for long cylindrical tanks where failure is expected in the cylinder. This method does not consider:

i) tank end effects
ii) tank fixtures, penetrations, etc
iii) material or weld defects
iv) corrosion, etc.

We expect the stress rupture method and the FOS method to agree on failure time for intense fully engulfing fires as in the case of RAX 201 [6]. This will be shown to be true in section 6.

It is recognized that the stress field will vary over the tank surface and through the tank wall thickness. For example, it is known that the tank stress is higher at the inner surface due to temperature gradients through the wall thickness (see, for example, Birk [3]). It is also known that the tank wall will experience bending stresses due to the non-uniform temperature patterns (hot on top, cold on bottom). These effects have been disregarded in this analysis.

In this failure analysis the principal damage stress is assumed to be the hoop or Von Mises stress (see, for example, [7]) in the cylinder section. For widespread heating of the cylinder, the hoop stress appears to give the best failure time estimates. For more localized heating, the Von Mises stress (0.877 x hoop) should give better failure time estimate. Predicting exact failure times is difficult. In this analysis the hoop stress tends to give a pessimistic failure time and the Von Mises tends to give a more reasonable failure time (but usually conservative by a few minutes).

The prediction of failure is accomplished using high-temperature stress rupture data for TC 128B pressure vessel steel (from Birk and Yoon [8]). This steel has a minimum ultimate tensile stress (UTS) of 550 MPa (80,000 psi). The actual UTS is more like 620 MPa.

Figure 4 shows recently obtained high-temperature stress rupture data for TC 128B steel [8]. This applies for TC 128B with an ultimate tensile strength of 620 MPa.

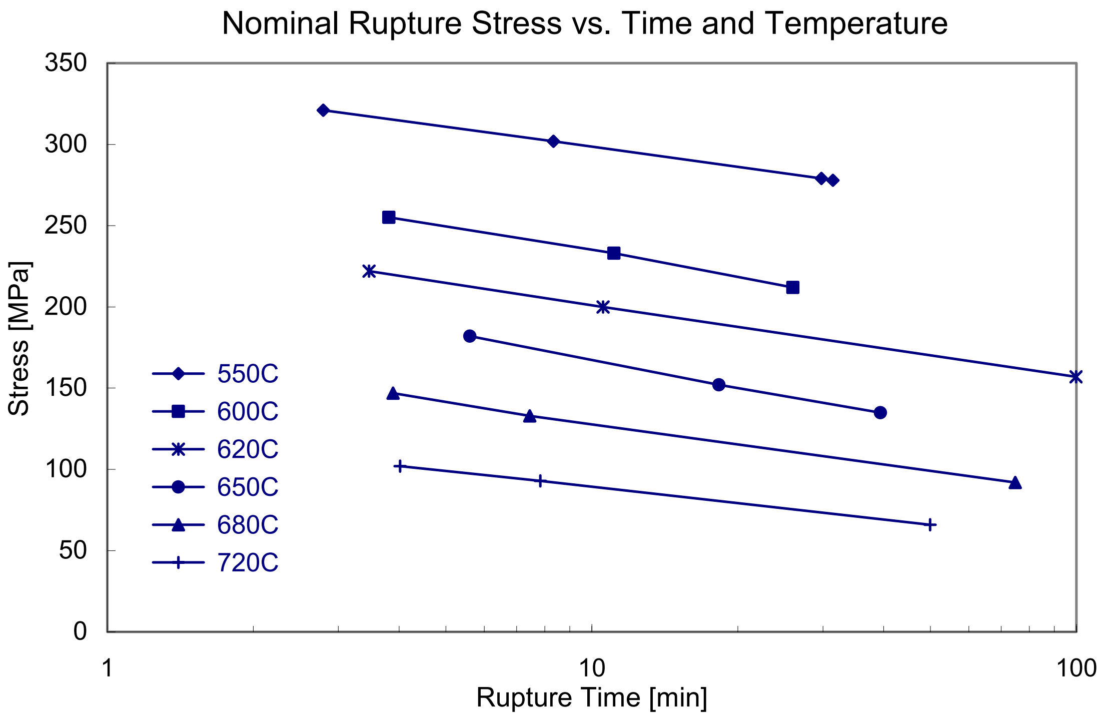

Figure 4 High-temperature stress rupture data for TC 128 steel [8]

This data was obtained from test samples under constant sample tensile load conditions. The samples were heated to a uniform temperature and then loaded until failure. The time to failure and percent elongation was recorded. Each line shown applies for a single sample temperature.

Figure 4 can be used as follows:

i) Determine the sample temperature and pick the appropriate line in the graph.
ii) Calculate the nominal tensile stress in the sample (σ = Force/area).
iii) Locate the stress on the y-axis – move horizontally to the right until you intersect the temperature line – then drop vertically down to the x-axis and read off the stress rupture time.

The failure analysis in IDA 2.1 uses the following procedure (see, for example, [5]):

i) The nominal hoop stress and Von Mises stress are calculated in the cylinder.
ii) The peak wall temperature is identified in the cylinder.
iii) Based on the nominal Von Mises stress and peak wall temperature, the time to failure is determined from the stored stress rupture data (Figure 4).

iv) Stress rupture damage is calculated using the following formula:

$$SR = \sum_{time} \frac{\Delta t}{t_f(\sigma_{vm}, T)}$$

where

SR = stress rupture damage
σ_{vm} = average wall nominal Von Mises stress
T = peak wall temperature
Δt = simulation time step
t_f = time to fail at stress and T

The SR value accumulates and increases over the time that the tank is heated and stressed. When SR = 1, failure is indicated.

The Von Mises stress in this case is simply 0.87 x the cylinder hoop stress. Plastic deformation of the wall is not accounted for in the stress calculation. This deformation is already accounted for in the stress rupture data obtained at constant load. The tank wall is under near constant load once the PRV set pressure has been achieved.

We realize that the tank wall will deform and bulge out before it fails and this will change the stress field. This stress is different than a simple stress rupture tensile test sample. As the tank bulges, its radius of curvature decreases and this reduces the stress. As the wall thins, the stress increases. The net effect is that the tank wall should fail more slowly than the simple tensile sample for the same nominal stress. In other words, we expect this analysis to be conservative.

Figure 5 shows how the predicted failure time varies with wall temperature for a given stress condition.

5.2 Effect of Defect Size, Shape and Location

Defect location, size and shape are already accounted for in the IDA model in terms of fire heating and wall temperatures. However, defect size, shape and location are not modeled in terms of the local stress field. The failure is based on nominal stress only. For very localized heating, a more detailed stress analysis may be required.

As defects get very small, the failure time will tend to increase. The current failure model should be conservative for all defect sizes. Data shown in section 6 suggests that the stress rupture approach used in IDA 2.1 gives good estimates for heated lengths (along the tank axis) of about 90 wall thicknesses or longer. For a 112J type tank this is a heated length of about 1.4 m.

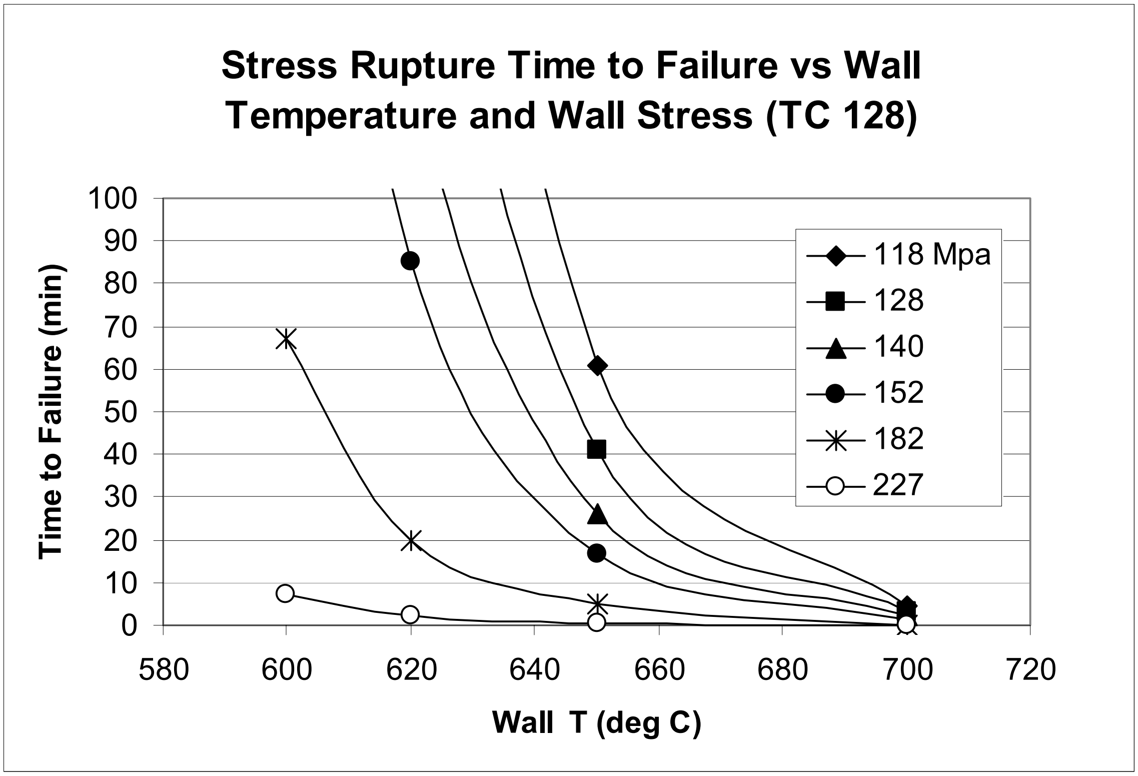

Figure 5 Estimated failure time (TC 128B with UTS = 620 MPa) as a function of wall temperature and wall stress. Failure based on Von Mises Stress (data from Birk and Yoon [8]).

6. Validation and Sample Results

Validation of IDA 2.1 was done by comparing its predictions with results from actual fire tests. This section is intended to prepare the reader for the following sections.

To some, validation means the model results match the experimental results within some acceptable tolerance. In simple problems where all the boundary conditions are well defined this is usually easy to do. However, with tanks in fires this is not so easy to do.

In this problem the fire dictates everything. Most fire test data is from open pool fires and this means the following:

i) the fire takes a finite time to get going (2-3 minutes)
ii) the fire intensity varies over time due to fuel delivery details
iii) wind can dramatically change the fire effects on the tank
iv) test stand geometry can cause swirling, etc. of local air currents

These may all be present in real-world fire scenarios, or they may not.

With open pool fires you can have one test where the tank survives without rupture and in the next test the identical type of tank fails violently with a BLEVE (boiling liquid expanding vapour explosion) outcome. The difference was probably different fire conditions. To be conservative, the model should be validated to the BLEVE outcome. For this reason it is very difficult to use one or two open fire tests to study some design detail such as thermal protection defects.

In this study we model the fire as follows:

i) At time = 0, the fire is on 100%.
ii) The fire is 100% engulfing.
iii) The fire is specified by an effective blackbody temperature.
iv) The fire conditions are constant over time.
v) There is no wind effect.

The code will actually allow the fire temperature to vary with time. However, it becomes difficult to define how the fire varies based on limited test data; for that reason, all simulations presented here use a constant fire temperature. For tank-car simulations, a fire temperature of 816^{o}C is used. It should be noted that this assumption is not conservative as large hydrocarbon pool fires can have effective blackbody temperatures as high as 900^{o}C [9].

The wall temperature in the vapour space, the tank pressure, and the wall thickness and material properties determine the time to failure. Wall temperatures are driven by local fire conditions. If the wind blows the fire off the top of the tank for a few seconds, this can have a large effect on the local wall temperature. The wall temperatures in fire tests are only measured at selected locations and it is not guaranteed that there are thermocouples located where the peak wall temperatures are. This means the peak measured wall temperature may not be the real peak wall temperature. Therefore, we should not expect the predicted wall temperatures from the model to look exactly like the measured wall temperatures. If the model predicts higher wall temperatures than measured, this may still be perfectly valid. The key question is – does the model include the important physical processes? The key processes are the convection and radiation in the vapour space and the fire heat flux.

The tank pressure, the liquid lading temperature, and the fill level are more driven by the average fire conditions and are therefore easier to model. We should expect better agreement between model and experiment with these if we use the correct fire temperature. If we set the fire temperature so that the model predictions of peak wall temperature agree with experimental results, it may not give us the correct tank fill level. If we set the fire temperature to give us the correct fill level, the local wall temperature may be wrong. This does not mean the model is wrong – it means the fire heating was not uniform due to other effects such as wind.

Because of these uncertainties, the IDA 2.1 code has been programmed to be conservative. If the model predicts failure early by a few minutes, this can be explained as follows:

i) We assume the fire is on 100% at t = 0; this could mean we predict failure early by a few minutes because we ignore the fire buildup.
ii) We assume the tank is 100% engulfed when, in fact, the tank may not be due to wind effects or fire size effects.
iii) We assume constant fire exposure and therefore higher predicted wall temperatures when, in reality, the fire exposure in the vapour space may be intermittent due to wind effects.

6.1 Pre-validation Code Changes

The IDA 2.0 code (version before IDA 2.1) was developed based on limited data on unprotected tanks. This validated well with the RAX 201 data [6], but there was no data for tanks with thermal protection defects. The thermal protection defect part of the code was partly validated for the fire and wall heat transfer (see Birk and VanderSteen [10]) but not for the overall tank model.

The IDA 2.0 code was used to predict the outcomes of the fire tests of a 500 gal. tank with thermal protection defects conducted in the summer of 2004 [11]. The following general observations were made:

i) For the baseline unprotected tank with 25% fire engulfment, IDA overpredicted the tank pressurization rate and underpredicted the peak wall temperatures.
ii) Even with the low wall temperatures predicted, the code predicted tank failure much earlier than experienced in testing (due to poor material property data for SA 455 steel).
iii) For the 15% defect case, IDA underpredicted the tank wall temperature by over $100^{o}C$ and also overpredicted the tank pressurization rate (by more than factor of 2). The net effect was that the code did not predict tank failure.
iv) The heating of the vapour space in the defect case was much more widespread than predicted by the IDA code.

The SA 455 material property data was replaced with more appropriate data for the tanks as tested. The SA 455 steel in the tanks had a UTS of about 610 MPa, compared with the minimum properties for SA 455 of 480 MPa. This made a very large difference to the failure predictions.

The most significant issue was the underprediction of wall temperatures for the defect cases. The following are possible reasons for the wall temperature underprediction:

i) assumed fire heat flux too low (i.e., assumed fire T too low)
ii) wall conduction to cooler areas of wall overpredicted

iii) back side convection too high
iv) back side radiation losses too high

IDA was able to predict the jacket temperature very well with an assumed fire temperature of 870°C and therefore it is not believed that the fire heat flux is too low. The measured jacket temperature leveled off at about 840°C, which was also predicted by IDA. An IDA run was performed with a fire temperature of 871 + 56 = 927°C and the jacket temperature leveled off above 900°C. This suggests the fire temperature was closer to 870°C. The 870°C assumed fire temperature also gave good liquid temperature rise rates.

The wall heat conduction losses are very small in the vicinity of the peak wall temperature; therefore, conduction was not believed to be the source of the underprediction.

It is very possible that the predicted wall convection is too high. Simple correlations are used for both free and forced convection.

The one remaining source of the error is that the wall radiation heat losses are too high. A likely source of this error is that the heated wall sees large areas of unheated wall, which IDA predicts to be quite cool. The heated wall also sees the relatively cool liquid surface. The testing showed that large areas of the wall in the vapour space got very hot due to high vapour temperatures, which IDA cannot predict with its two-node thermal model.

If the liquid surface emissivity is reduced to 0.0, then the cooling effect of the liquid will be reduced to only a conduction effect at the wall. This will result in higher vapour space wall temperatures. It also means the wall temperature will be insensitive to tank fill level, which is counter to the published literature. Fire testing with water in the 500 gal. tanks suggests that there is a 50°C difference in peak wall temperature for a tank filled to 50% vs. a tank filled to 80%. This is a significant effect.

The inside tank surface reflection may be higher than currently assumed. If we reduce the inside wall emissivity, we would get higher wall temperatures. However, this would be counter to available emissivity data for rolled steel plate. The emissivity would have to be reduced to around 0.4 for this affect to correct the prediction errors.

When the PRV is not open, the vapour space can get very hot. In these tests we observed vapour space temperatures around 300°C with the PRV closed. Once the PRV opens, the vapour space temperature drops down to the saturation temperature of the PRV operating pressure. An attempt was made to see whether this was the issue. The vapour space temperature was set to 300°C and a simulation was performed. This only increased the peak wall temperature by about 25°C.

An IDA simulation was performed with the emissivity of the liquid surface set to 0.2, but this only increased the peak wall temperature to 635°C, which is still about 70°C too low. This was not the solution.

The next attempt was to reduce the emissivity of the tank inside wall from 0.9 to 0.4. This solved the wall temperature problem. The peak wall temperature rose to 707°C in excellent agreement with test 04-03. The jacket temperature was also in good agreement with the test at 850°C. This low emissivity was not considered credible until a brief literature review was conducted. New rolled steel plate was reported in various sources to have emissivity as low as 0.28, so a setting of 0.4 for a new tank is not unreasonable.

The baseline case was run again with IDA, with the inside wall emissivity set to 0.4. This resulted in too high a wall temperature and early tank failure (FOS fail at 5.2 min and SR fail at 5.9 minutes). At 8 minutes (actual failure time) IDA predicted a wall temperature of about 773°C or about 35°C too high. This is conservative but not excessively so. Also note that IDA predicted PRV pop at 6.8 minutes. In the actual test, the PRV did not pop before failure at 8 minutes. Therefore, the early failure prediction here is partly due to the overprediction of tank pressure. We should also recall that in the fire test the fire took about 2 minutes to build. In the simulation the fire is assumed on 100% at t = 0.

As a result, the code was modified as follows:

i) The SA 455 material property data was replaced with better data (the tested steel had a UTS of about 610 MPa vs. the minimum allowable for SA 455 of 480 MPa).
ii) Stress rupture data was included from the recent testing of TC 128B.
iii) The two-node vapour space radiation model was replaced with a three-node model.
iv) Vapour space convection was turned off (i.e., h = 0 W/m^2K) at the very top of the tank.
v) The fraction of the liquid in the warm boundary layer was increased from 5% to 10%.
vi) The emissivity of the tank wall on the inside surface was reduced from 0.9 to 0.4.
vii) The liquid surface emissivity was reduced from 1.0 to 0.5.

The code with these changes is IDA 2.1.

These changes reduced the pressurization rate and increased the vapour space wall temperature. They also delayed tank failure to higher wall temperatures.

The code is seen to have a flaw in that it cannot predict the vapour temperature accurately when the PRV takes a long time to open. The vapour temperature was observed to approach 300°C in the testing and the IDA code does not predict vapour temperatures above 72°C. This requires a three-node thermal model for the liquid and vapour lading (separate nodes for vapour space, liquid boundary layer and liquid core). However, even with this flaw, the code appears to be able to predict tank wall temperatures and failure in a reasonable way.

6.2 Validation Case – RAX 201

This is a benchmark case of a full-scale, non-thermally protected tank car engulfed in fire. Further details of this test can be found in [6]. The basic test conditions were as follows:

i) tank type = 112A340W
tank volume = 128,000 L (33,700 US gal.)
tank material = TC 128, minimum ultimate strength 550 MPa; typical room temperature ultimate strength 590 - 620 MPa

ii) tank ID = 3 m, wall thickness = 16 mm

iii) LPG was 98% propane and 2% ethane

iv) PRV was a Midland A-3180-N with P set = 1.93 MPa and flow rating was 39,400 scfm air at 120% of P set (PRV effective area 0.053 m^2 assuming a Cd for air = 0.8)

v) tank loaded to 96% of volume with 60,800 kg of propane at 21°C

vi) fire temperatures fluctuated between 650 and 990°C

vii) time averaged heat input to the wetted tank wall calculated to be 100 kW/m^2 (effective blackbody T = 877°C)

It should be noted that the standard DOT engulfing fire temperature is 871°C plus or minus 56°C. However, it is common that the AFFTAC code is run at the lower extreme of this range, or 816°C. This is done because the AFFTAC code was validated at this lower temperature (i.e., AFFTAC uses T = 816°C and e = 0.8; with these it predicts the correct failure time of 24.5 minutes). The 816°C fire temperature is probably a good estimate of the overall average fire temperature the tank saw in the RAX 201 fire test [6]. It may not be a conservative number for estimating possible peak wall temperatures in credible hydrocarbon pool fires.

The following results from IDA 2.1 are for the case of an unprotected tank exposed to a 100% engulfing fire at 816°C blackbody temperature. The following results are presented in Figures 6 through 11:

i) tank pressure vs. time
ii) lading temperature vs. time
iii) wall temperatures vs. time
iv) PRV mass flow vs. time
v) fill vs. time
vi) SR damage and FOS vs. time

IDA 2.1 predicts failure in just under 20 minutes. The fire test result from Townsend et al. [6] was 24 minutes. One possible reason for this difference is that the fire buildup time (a few minutes) was not accounted for in the IDA simulation. IDA predicts slightly higher wall temperatures than AFFTAC.

The fire test of an unprotected tank (RAX 201 [6]) yielded a failure time of 24 minutes. However, reasonable thermal model predictions show that this failure time could have been anywhere from 15 minutes to 30 minutes (i.e., 24 minutes + 20% and – 38%).

If we look at the FOS vs. time (Figure 11) for a non-thermally protected tank exposed to an engulfing fire, we see that the FOS drops rapidly in the first 10 minutes and then levels off. Exactly where it levels off is hard to predict because it is extremely sensitive to the peak wall temperature. An error of 20^{o}C in predicted wall temperature can make a big difference in the FOS and in the time to failure. The strength difference between a wall at 640^{o}C and 660^{o}C is more than 13%. This is for an error in prediction of wall temperature of only 3%. This 3% error in predicted wall temperature results in a 13% error in FOS, and this can mean an error in the prediction of time to failure of 20%.

It should also be pointed out that, in this case, the FOS method and the stress rupture method agree almost exactly. This is because we are dealing with a full intensity heating situation as described in section 5.

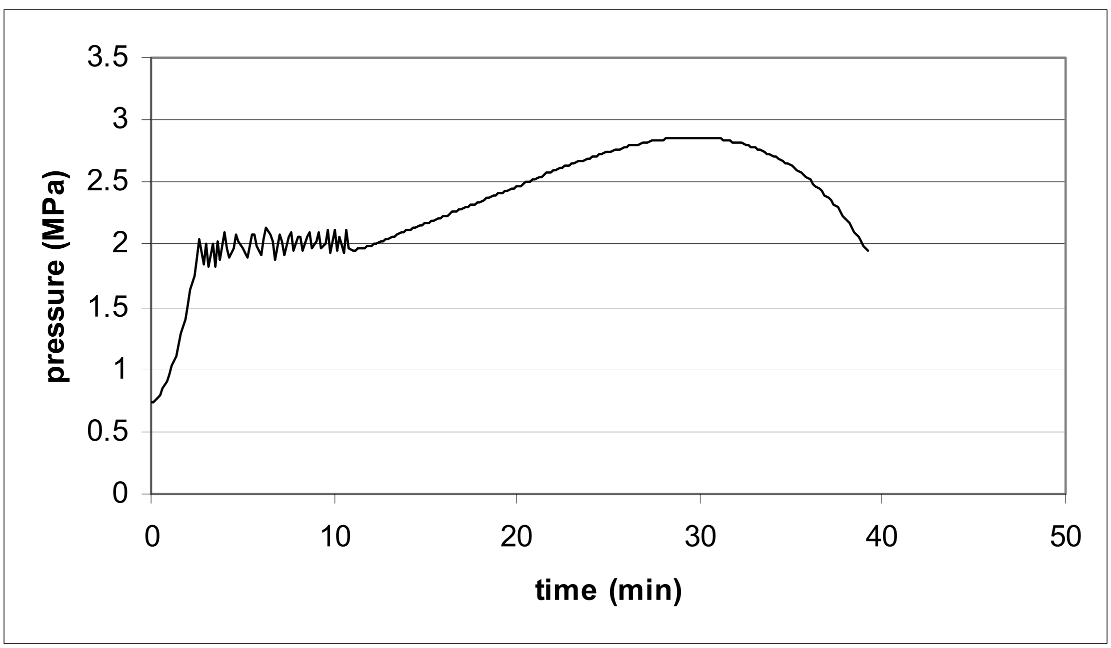

Figure 6 IDA 2.1 results – pressure vs. time for unprotected tank RAX 201 (fire T = 816°C, fill 0.94, propane, PRV set to 1.93 MPa, PRV scfm 34,000)

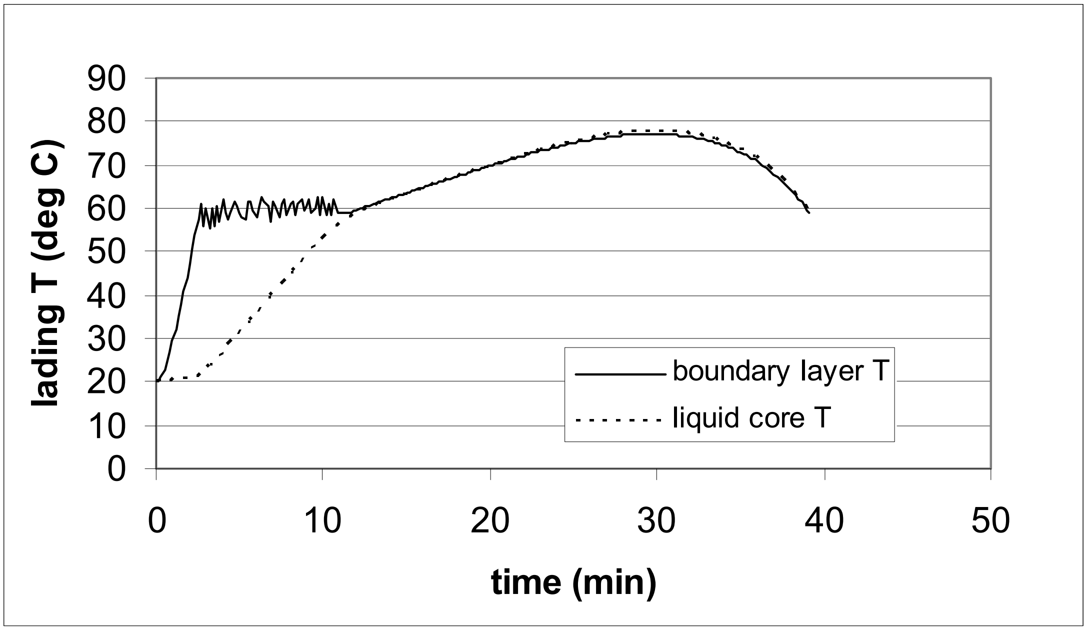

Figure 7 IDA 2.1 results – lading T vs. time for unprotected tank RAX 201 (fire T = 816^{o}C, fill 0.94, propane, PRV set to 1.93 MPa, PRV scfm 34,000)

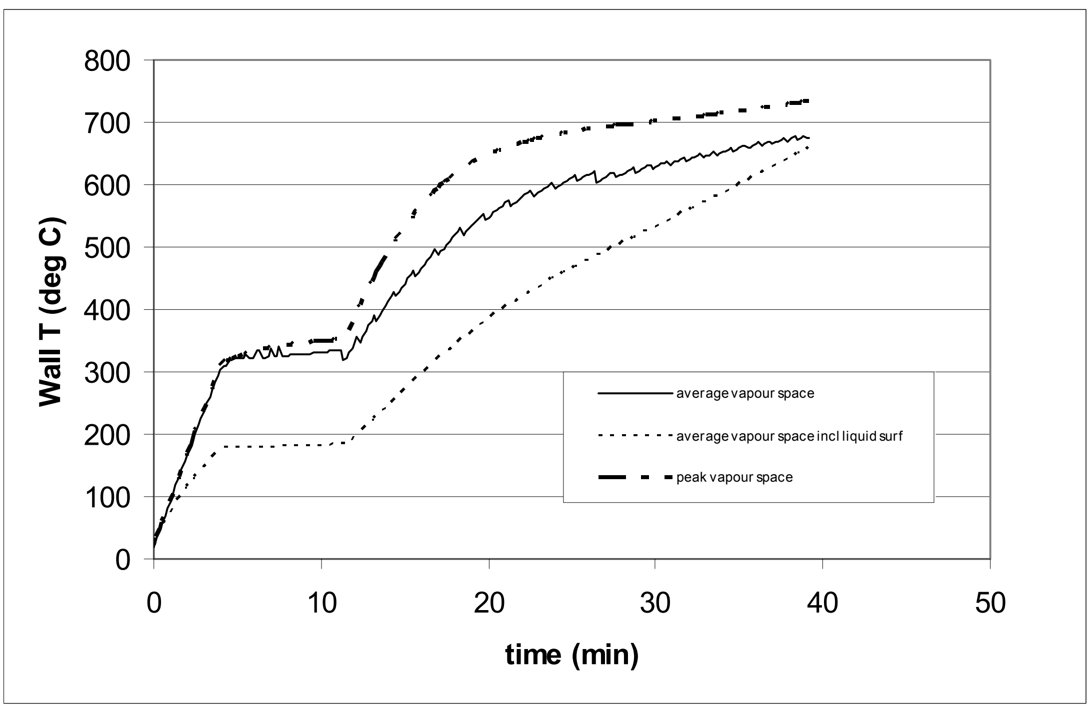

Figure 8 IDA 2.1 results – wall T vs. time for unprotected tank RAX 201 (fire T = 816°C, fill 0.94, propane, PRV set to 1.93 MPa, PRV scfm 34,000)

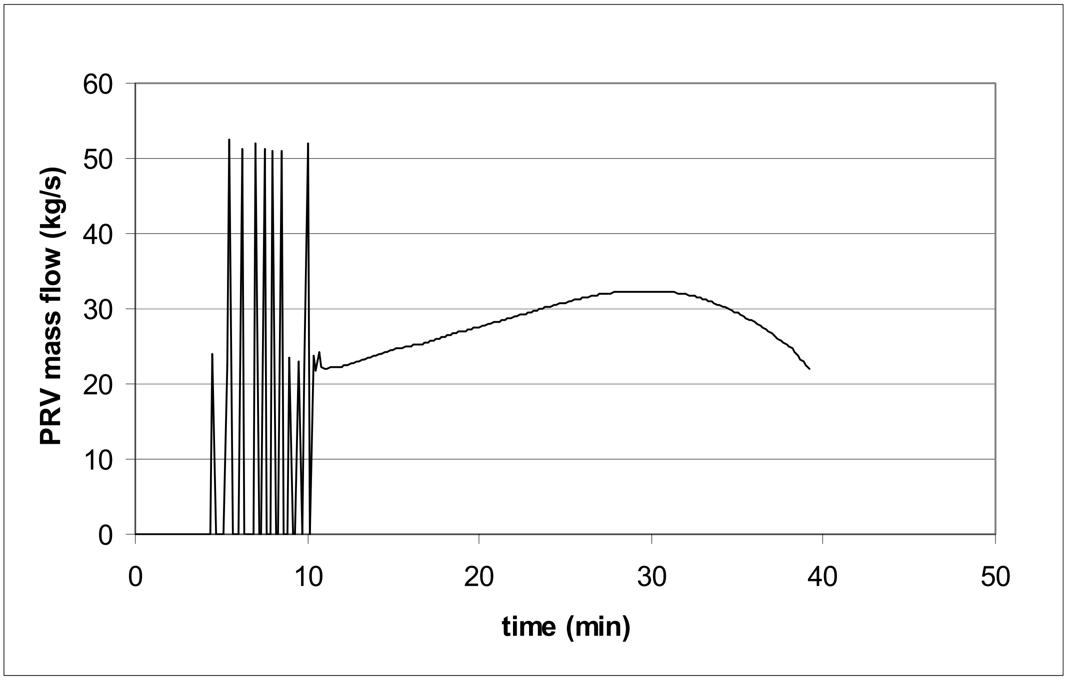

Figure 9 IDA 2.1 results – PRV mass flow vs. time for unprotected tank RAX 201 (fire T = 816°C, fill 0.94, propane, PRV set to 1.93 MPa, PRV scfm 34,000)

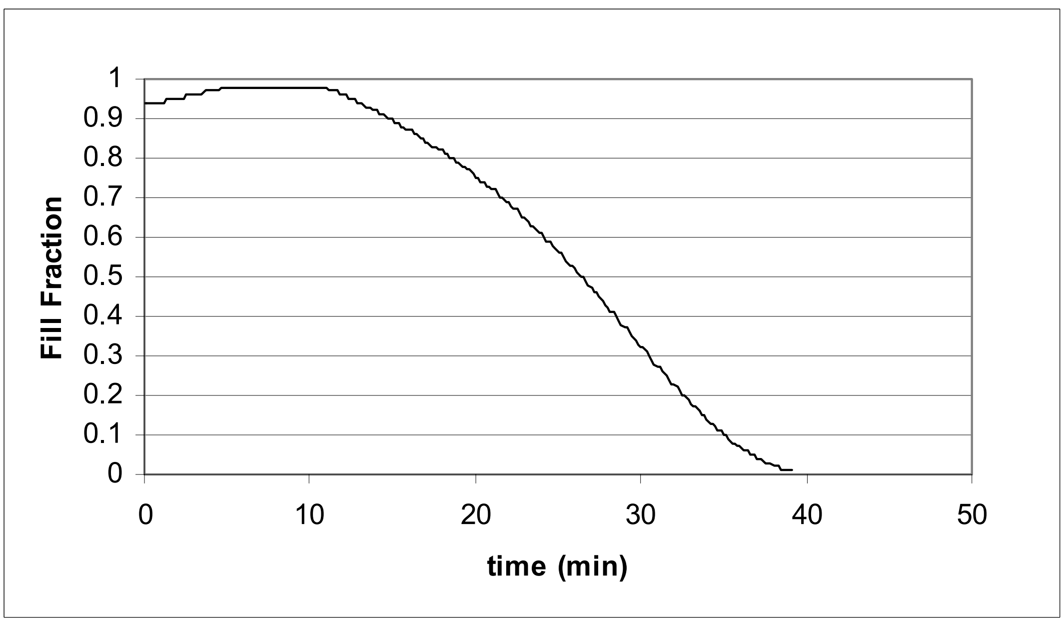

Figure 10 IDA 2.1 results – fill vs. time for unprotected tank RAX 201 (fire T = 816°C, fill 0.94, propane, PRV set to 1.93 MPa, PRV scfm 34,000)

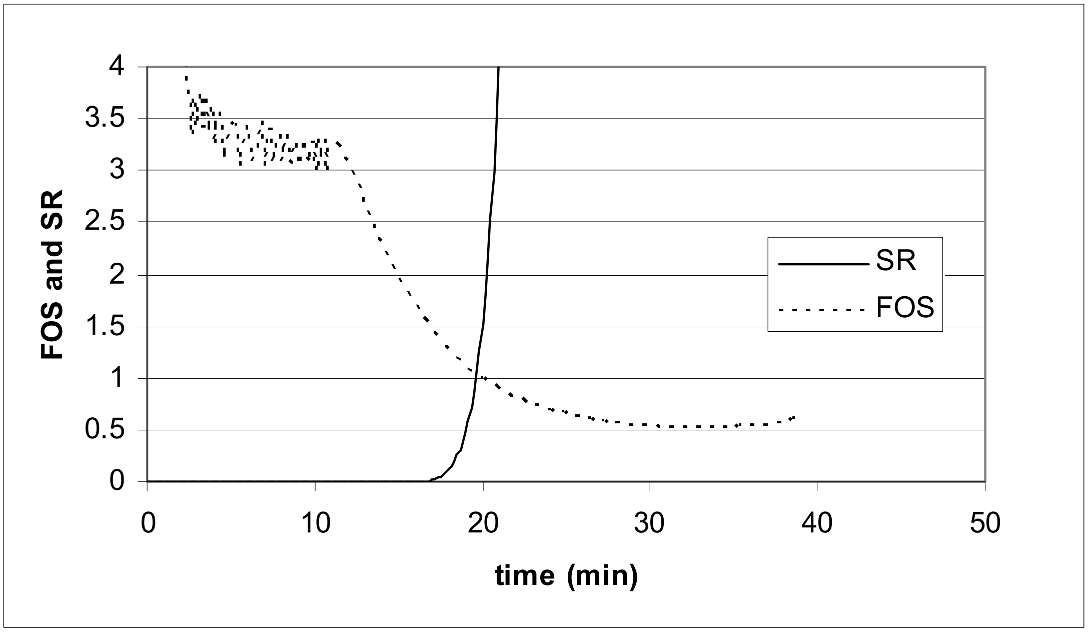

Figure 11 IDA 2.1 results – stress rupture damage (SR) and factor of safety (FOS) vs. time for unprotected tank RAX 201 (fire t = 816°C, fill 0.94, propane, PRV set to 1.93 MPa, PRV scfm 34,000)

6.3 Validation with Defect Data – 2004 Field Trial Data

The latest validation of the IDA code was done using the new fire test data of tanks with simulated thermal protection defects. Fire tests were conducted by A.M.Birk at the Department of Mechanical and Materials Engineering at Queen's University under contract to Transport Canada. The tests were conducted using 500 gal. ASME code propane tanks to simulate 33,000 gal. tank-cars. The actual tests were conducted at the Munitions Experimental Test Centre (METC) at Valcartier, Quebec.

The fire tests were set up so that the model tanks experienced the same hoop stress to ultimate strength ratio as the full-scale tank-cars. The tests were conducted because there is no data available on the behaviour of tank-cars with local thermal protection defects.

These tests involved 500 gal. ASME code tanks with thermal protection defects covering 8 and 15% of the tank total surface. Baseline tests were also conducted on a tank with no thermal protection. In all tests the tanks were 25% engulfed in fire from one side. The defects spanned from the tank top to near the bottom. Full details of this testing can be found in Birk et al. [11].

The following cases were used to validate the IDA 2.1 code for the case of tanks with thermal protection defects:

i) Case 1 – Baseline (unprotected tank, 25% fire exposure) PRV pop pressure set to 386 psig – test 04-06
ii) Case 2 – 15% defect, 25% fire exposure, PRV pop pressure set to 386 psig – test 04-03
iii) Case 3 – 8% defect, 25% fire exposure, PRV pop pressure set to 386 psig – test 04-05

In the tests the fire temperature varied because of wind effects. At times the effective fire temperature dropped below 700^{o}C. This was obvious from the measured jacket temperatures. All final simulations were done with a blackbody fire temperature of 871^{o}C.

The elevated PRV set pressure was used so that the tanks would have the same hoop stress to ultimate strength ratio as a full-scale tank-car. The validation results are shown in sections 6.3.1 to 6.3.3.

6.3.1 Case 1 – Baseline (unprotected tank with 25% fire exposure)

This was test 04-06 from the summer 2004 fire testing. The case was as follows:

i) 500 gal. ASME code tank
7.1 mm wall
SA 455 steel
80% full
initial T = 14^{o}C

ii) fire consisted of an array of 25 liquid propane burners
approximate coverage was 25% of tank surface area
fire applied to tank side
effective blackbody fire T = 870^{o} C

The IDA 2.1 code was run for this case and the following main results are shown in Figures 12 through 14:

i) tank pressure vs. time
ii) tank peak wall T vs. time
iii) tank FOS and SR

FOS = factor of safety = (material ultimate strength at peak wall T)/(hoop stress)
SR = stress rupture damage based on Von Mises stress and SR data from TC 128 steel

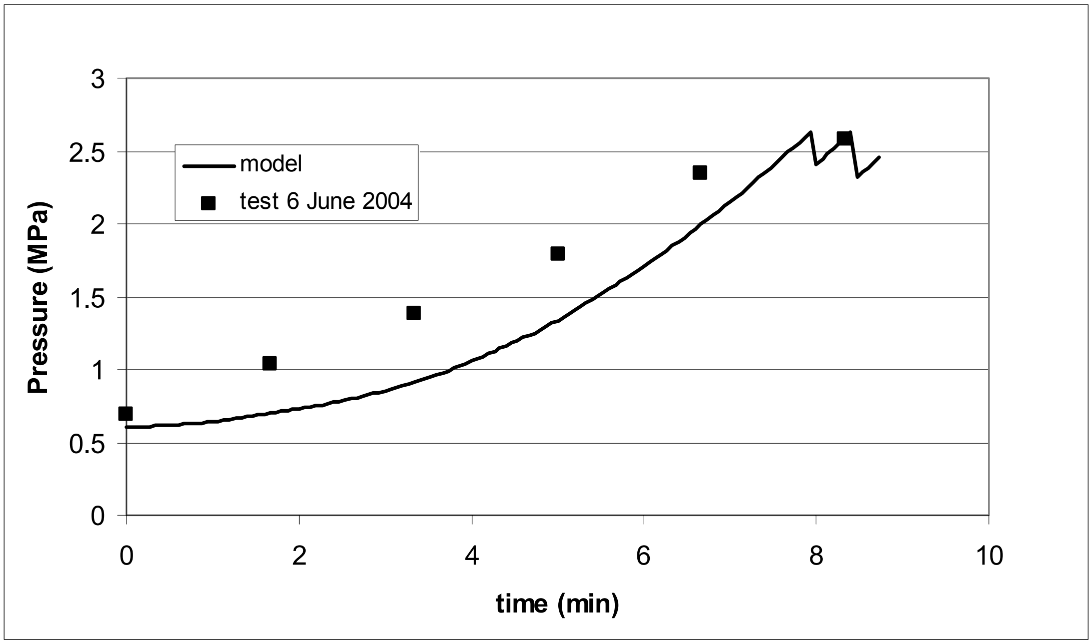

Figure 12 Predicted pressure vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

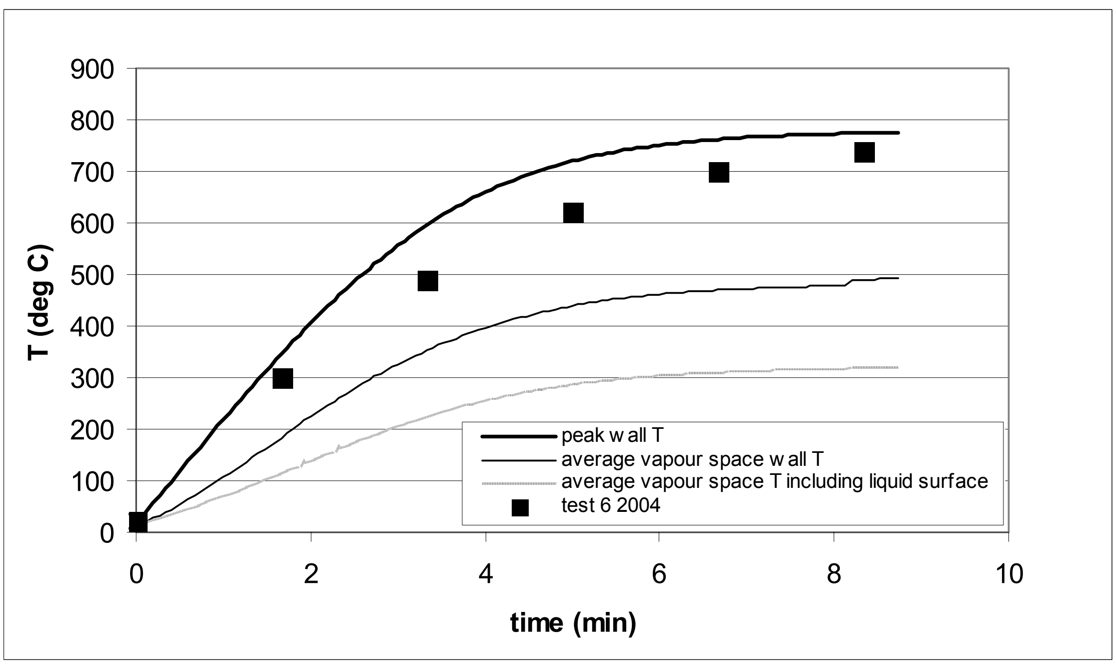

Figure 13 Predicted wall T vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

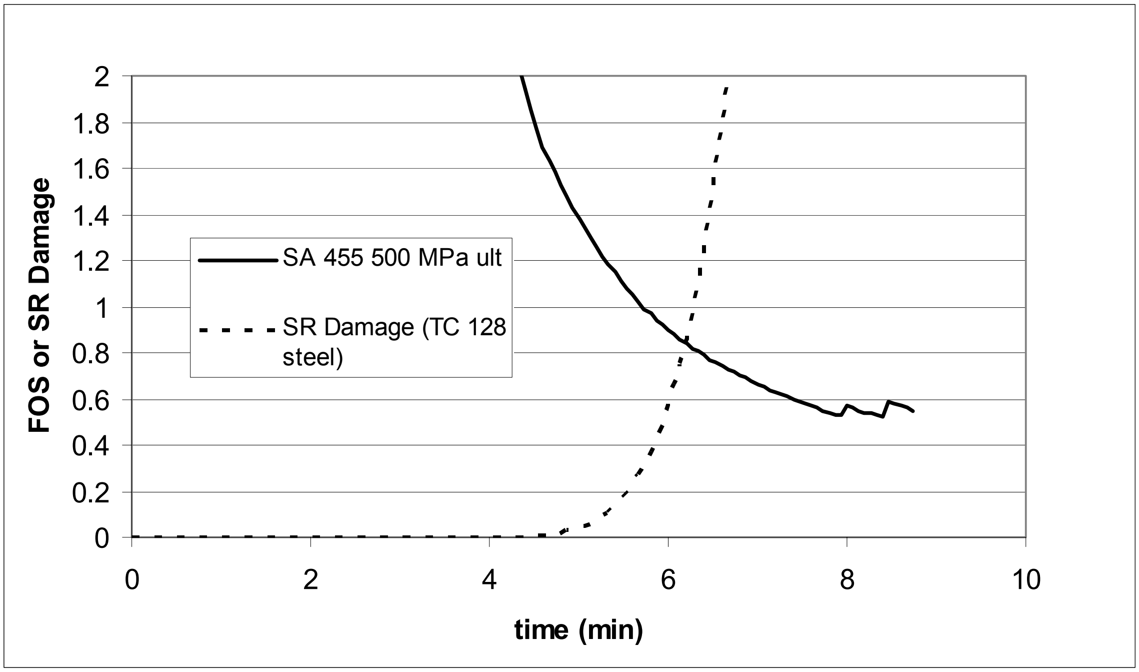

Figure 14 Predicted failure vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

The following can be seen in Figures 12 to 14:

i) The pressurization rate is reasonably well predicted. The IDA 2.1 code predicted the PRV would open at about 7.8 minutes. In reality the tank failed at 8 minutes before the PRV opened. The test data showed the PRV would have opened within a minute of the failure time.

ii) The peak wall temperature is somewhat overpredicted. The initial temperature rise rate is overpredicted by IDA (conservative) and then the peak wall achieved is overpredicted by IDA by about 25°C (conservative).

iii) The failure time is predicted when the FOS drops to 1.0 or when the SR rises to 1.0. The simple failure model (FOS = 1) predicts failure at about 5.9 minutes (conservative) while the SR method predicts failure at about 6.3 minutes (also conservative). The actual failure time was 8 minutes.

In general the model predictions are in good agreement with test results. The differences can partly be attributed to the fact that the actual test fire took 1-2 minutes to reach full intensity.

6.3.2 Case 2 – 15% defect

This case involved a tank with 15% of its area with thermal protection defect (i.e., steel jacket with no insulation under). The 15% defect was 100% covered with fire. Again, the total fire contact was about 25%.

This was test 04-03 from the summer 2004 fire testing. The case was as follows:

i) 500 gal. ASME code tank
7.1 mm wall
SA 455 steel
71% full
initial T = 14^{o}C

ii) fire consisted of an array of 25 liquid propane burners
approximate coverage was 25% of tank surface area
fire applied to tank side
effective blackbody fire T = 870^{o}C

The rest of the tank was thermally protected with 13 mm of ceramic fibre tank-car insulation. The assumed insulation conductivity for simulation purposes was 0.050 W/mK (this is appropriate because only the defect area was engulfed in fire) for the 500 gal. tank. These are believed to be the true insulation properties at low temperatures. However, it should be noted that for tank-car simulations with 100% fire engulfment the assumed k is between 0.15 and 0.3 W/mK, which is the expected range for the insulation when it is between 400 and 800°C.

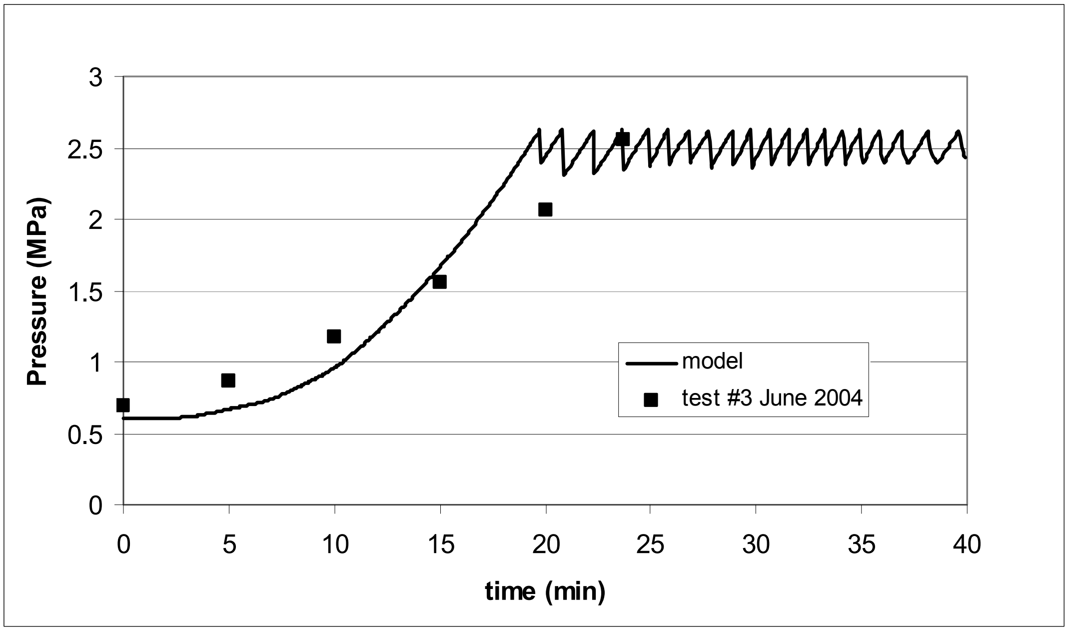

Figure 15 Predicted pressure vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

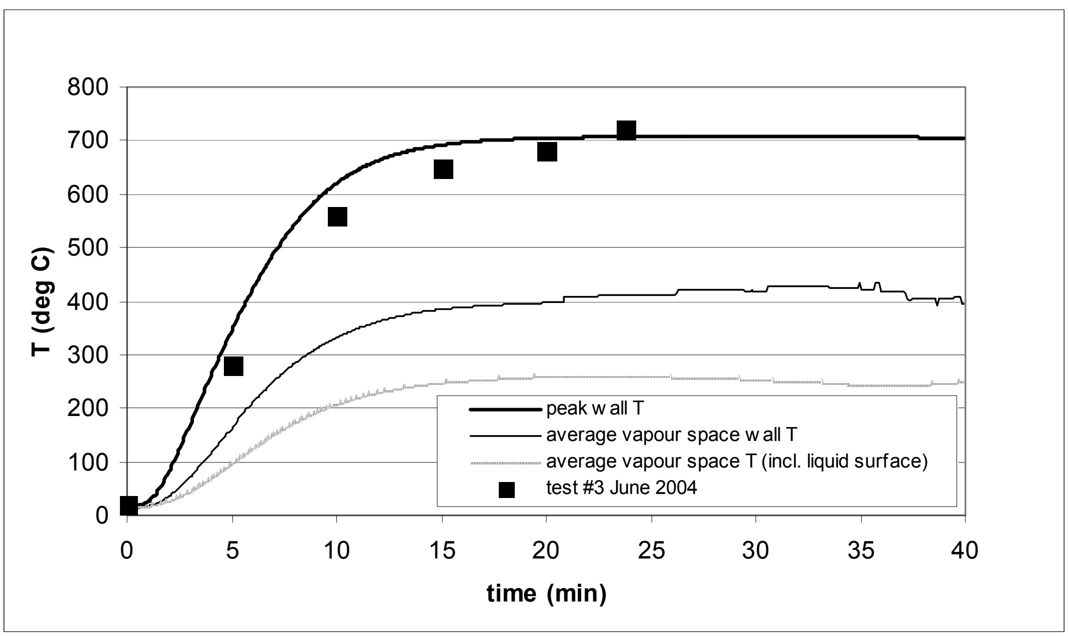

Figure 16 Predicted wall T vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

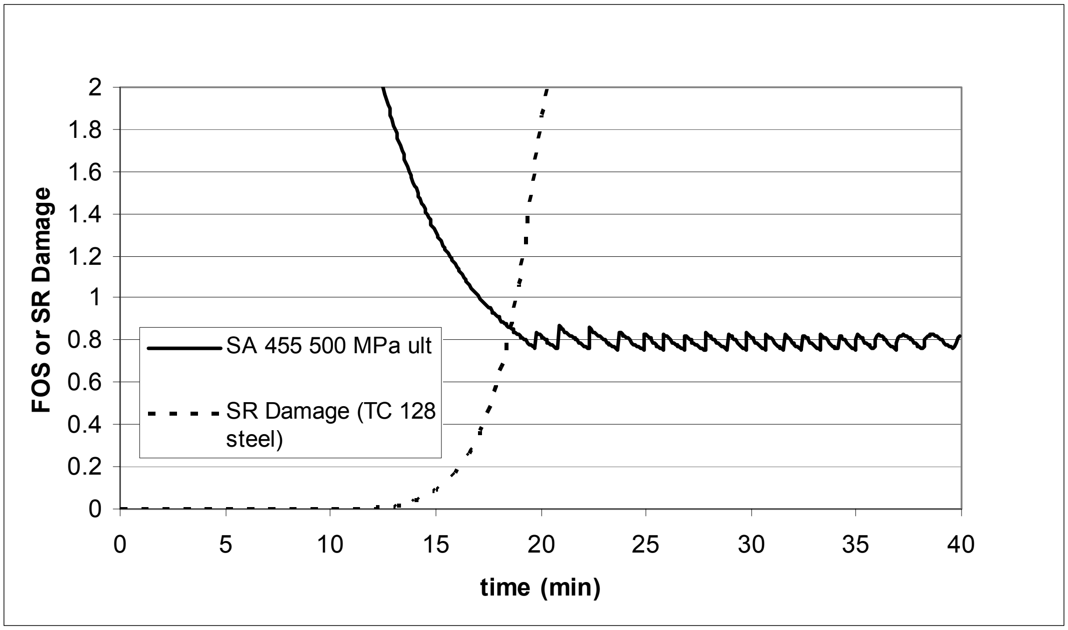

Figure 17 Predicted failure vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

The addition of the thermal protection increased all critical times (PRV pop time, failure time, etc.) by a factor of about 3. The following is observed from Figures 15 to 17:

i) IDA overpredicts the pressurization time by a factor of about 1.2 (20% early, conservative).
ii) IDA overpredicts the initial wall temperature rise rate and then underpredicts the peak temperature slightly.
iii) IDA predicts failure by stress rupture at 19 minutes vs. the actual failure time of 24 minutes.

The results are in reasonable agreement with experiment. The prediction of failure is conservative. If fire buildup time were accounted for, the predicted failure time would probably be around 21 minutes.

6.3.3 Case 3 – 8% defect

This case involved a tank with 8% of its area with thermal protection defect (i.e., steel jacket with no insulation under). The 8% defect was 100% covered with fire. Again, the total fire contact was about 25%.

This was test 04-05 from the summer 2004 fire testing. The case was as follows:

i) 500 gal. ASME code tank
   7.1 mm wall
   SA 455 steel
   71% full
   initial T = 17^{o}C

ii) fire consisted of an array of 25 liquid propane burners
   approximate coverage was 25% of tank surface area
   fire applied to tank side
   effective blackbody fire T = 871^{o}C

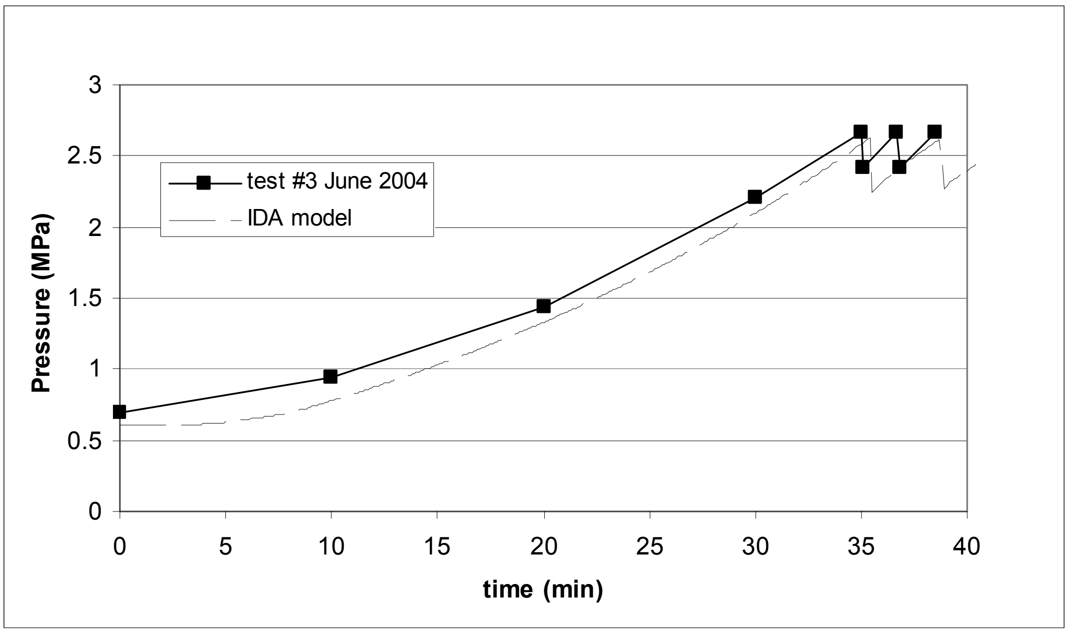

Figure 18 Predicted pressure vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

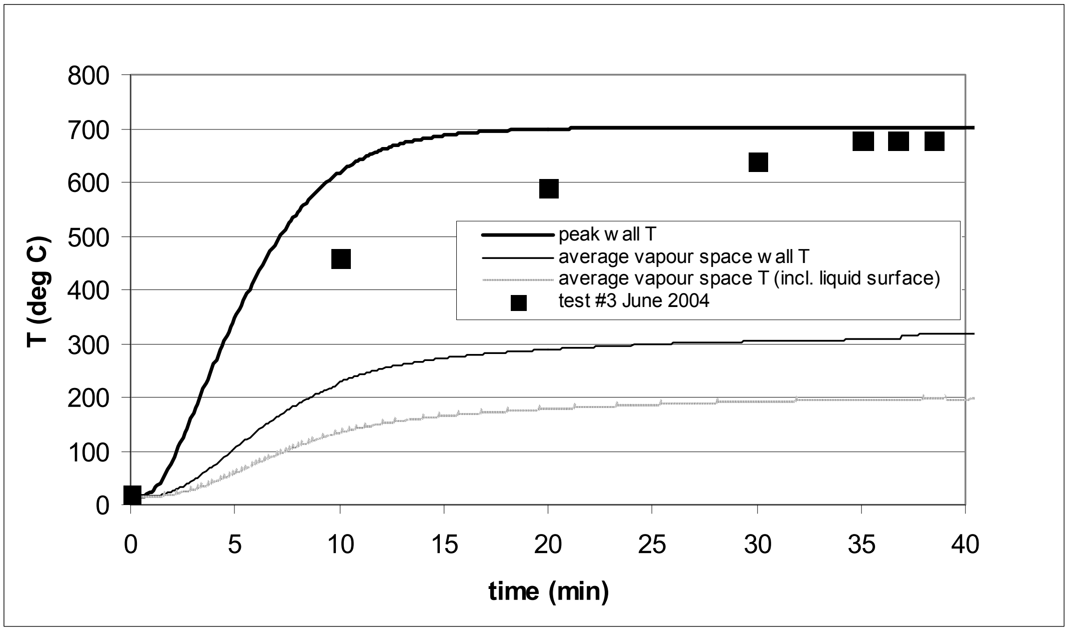

Figure 19 Predicted wall T vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

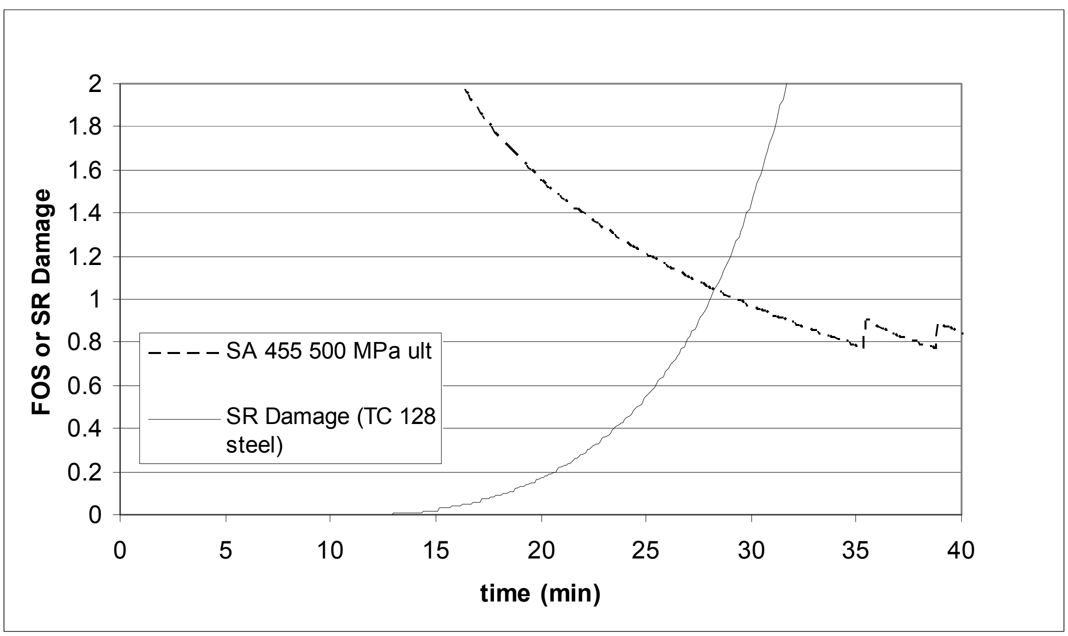

Figure 20 Predicted failure vs. time from IDA 2.1 (baseline 500 gal. ASME code tank, no thermal protection, 25% engulfed in 871°C blackbody fire, initial fill 80%, PRV set to pop at 2.66 MPa)

As can be seen from Figures 18 to 20, the pressure prediction is very good. The PRV opens around 35 minutes. For the wall temperature IDA overpredicts the temperature early in the fire. However, it should be noted that the wind affected this test significantly and it is believed that the IDA predictions are very reasonable for a fire with no wind effects.

The predicted failure time is about 28 minutes. The actual failure time was 60 minutes. However, during the test the fire intensity dropped dramatically due to the wind. It is assumed that had the wind not affected the fire, the wall temperature would have exceeded 700^{o}C at 30-36 minutes and this would have been the likely failure time. If we consider the effects of the wind on the fire then the IDA predictions are very reasonable.

Table 1 gives a summary of the IDA 2.1 validation. The table includes comparison of IDA 2.1 with the RAX 201 test of a full-scale unprotected tank-car.

Table 1 Summary of thermal model results

Result	**RAX 201 Full-scale tank**		**500 gallon no thermal protection**		**500 gallon 15% defect**		**500 gallon 8% defect**	
	test	model	test	model	test	model	test	model
PRV first pop (minutes)	2	2	8-9	7	25	21	35	31
time to fail (minutes)	24	19	8	6.1	24	19	36	28
fill at fail (%)	50	60 (see note)	75	75	71	70	74	72
peak wall T at fail (°C)	640	645	720	730	720	730	690	710

Note: The prediction of 60% fill for RAX 201 case is due to the low fire temperature of 816^{o}C. This fire temperature gives the correct wall temperature but does not represent the true average fire temperature the entire tank sees. As a result, the tank empties too slowly.

In general, IDA predicts the first PRV pop and failure early by a few minutes. This may be partly due to the test fire taking time to reach full intensity, which is not modeled. In any case, Table 1 demonstrates good validation for several cases. The model is slightly conservative on predicting time to failure.

7. Thermal Protection Defect – Thermal Modeling Results

The fire tests carried out with the 500 gal. tanks were not perfect models of the 112J type tank-car. For example, the fill levels were not the same. The 500 gal. tanks were filled to about 70 - 80% while tank-cars may be filled to above 95%. We know the liquid level is important and therefore we must correct for this. The 112J tank also has a larger L/D ratio (i.e., the 112J length is about six times as long as its diameter while the 500 gal. tank is about three times as long as its diameter). To correct for this, we need a detailed thermal model is needed of a tank in a fire like IDA 2.1.

Before we present the simulation results we should say a few things about the thermal protection system and the PRV.

7.1 Thermal Protection

Thermal protection is used to slow the rate of heating from a fire. Thermal protection involves covering the tank with a thermal insulation material. This insulating layer slows the rate of heating and this delays the pressure rise, the wall temperature rise and the tank failure. The current thermal protection systems for 112J type cars have been designed so that a tank can be expected to survive a credible hydrocarbon pool fire for 100 minutes or a jetting fire for 30 minutes.

The original design of the thermal protection system was intended to keep the tank wall temperature below 427^{o}C (see [12]) for 100 or 30 minutes depending on the fire type. At this temperature the tank is not expected to fail at all if the PRV is working to keep the tank pressure near the PRV set pressure. In the RAX 201 engulfing fire test [6], the unprotected wall reached 427^{o}C in about 13 minutes. With the thermal protection system we expect to delay this until about 100 minutes.

The most common modern thermal protection system for tank-cars involves a 13 mm thick blanket of high-temperature ceramic fibre insulation. This is then covered with a 3 mm jacket of steel. We can see the benefit of this insulation in Figure 21 from Birk and VanderSteen [10]. From this figure we see the wall temperature vs. time plot for the no thermal protection case and the full thermal protection case. The slope of the non-protected case is between 9 and 11 times as steep as that for the thermally protected case. This suggests a reduction in heat flux for the thermally protected wall of about a factor of 10. Table 2 summarizes the thermal conductivity of this type of ceramic blanket.

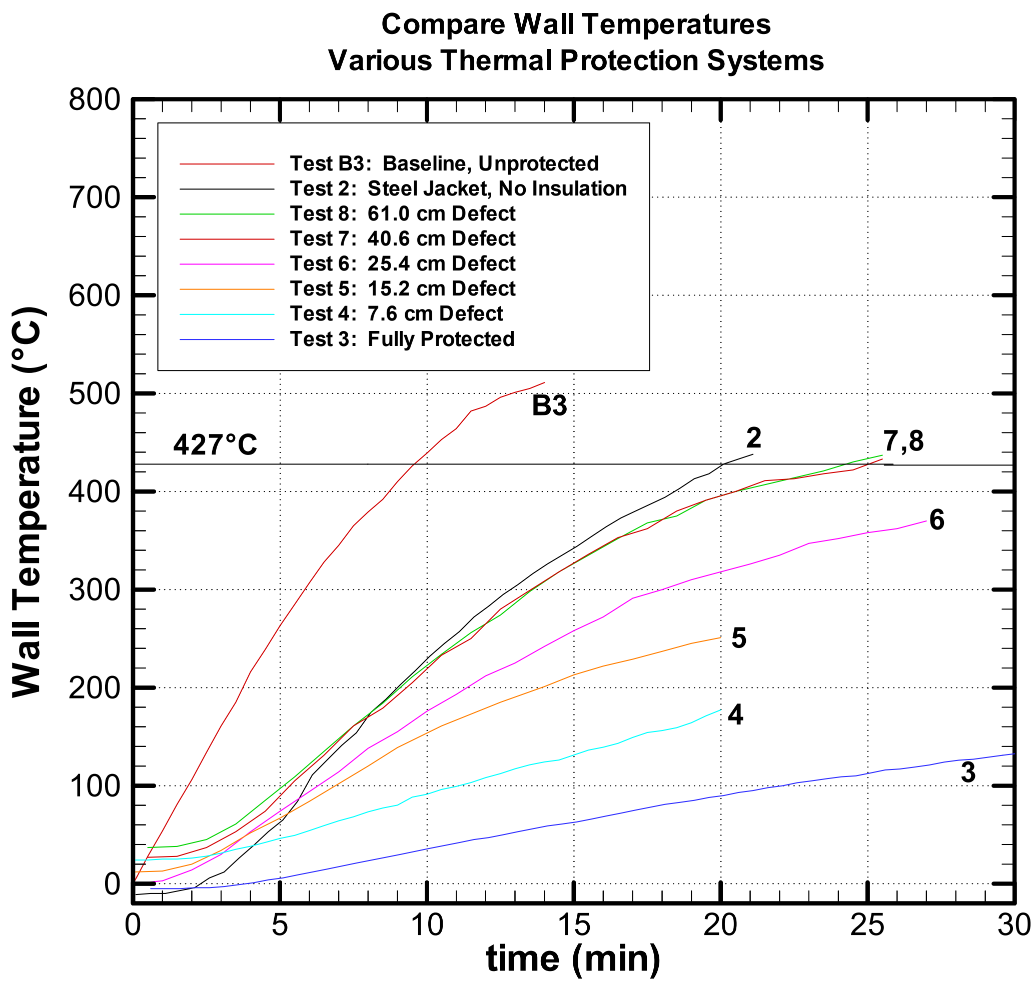

Figure 21 Measured wall temperatures for various defects (data from Birk and VanderSteen [10])

Table 2 Summary of ceramic fibre insulation properties (Unifrax, tank-car insulation, 72 kg/m^3 density, new condition)

Temperature (°C)	**Thermal conductivity *k* (W/mK)**	**Comment**
-20	**0.03**	
100	0.05	liquid wetted wall temperature
300	0.09	
500	0.15	protected vapour space wall temperature
650	0.20	
800	0.30	jacket temperature in engulfing fire

As can be seen from Table 2 the thermal conductivity increases as the temperature increases. The fire will quickly heat the jacket to near the fire temperature so the thermal conductivity of the insulation near the jacket will be around 0.3 W/mK. On the wall side of the insulation, the insulation will take up the wall temperature. In the liquid wetted regions this means the insulation will have a *k* around 0.05 W/mK. The net affect is an average *k* of about 0.17- 0.2 W/mK in wall areas cooled by the liquid. In the vapour space the k is closer to 0.3 W/mK.

The *k* also depends on the blanket density. If the insulation is crushed, the *k* increases.

When a thermally protected tank is engulfed in fire the outer jacket will rapidly approach the fire temperature. The tank wall in the liquid space will stay near the liquid temperature. This gives an average insulation temperature of about 450^{o}C. At this temperature the thermal conductivity of the insulation will be about 0.14 W/mK.

All thermal protection systems have some direct conduction links due to tank structures. This will tend to increase the average effective thermal conductivity of the thermal protection system. Local crushing of the insulation will also increase this conductivity. Johnson [4] states that the maximum allowable effective thermal conductivity of the system that can still meet the plate test standard is 0.295 W/mK for a 13 mm blanket.

It should be noted that thermal protection only delays failure. If the fire were to last long enough, then eventually the tank would empty, the wall temperature would approach the fire temperature and the tank would fail if it were still pressurized.

7.2 Pressure Relief Valves

PRVs are sized such that they can control the tank pressure in the event that the tank is engulfed in fire. The size of the PRV depends on the rate of fire heating of the tank lading. If a tank is thermally protected, then the AAR PRV sizing formula allows the size of the PRV to be reduced from that for an unprotected tank-car. For a tank-car with 13 mm of high-temperature ceramic thermal insulation, the PRV flow capacity can be reduced by a factor of about 10 from the case of an unprotected tank-car. This typically means the flow rating can be reduced from about 16 to 1.6 m^3/s of standard air (35,000 to 3500 scfm) for a 112 J type car. If a tank is equipped with such a small PRV, then the issue of thermal protection defects becomes even more critical.

7.3 Sample Results – 112J Tanks with Thermal Protection Defects

Here we present some detailed results from selected simulations. This first case is a 100% engulfed tank with 100% defect. The second is a tank with 8% of its surface covered with defect.

7.3.1 100% Defect

This is the case where the tank has a steel jacket and air gap but no thermal insulation. The basic inputs to the program were:

i. 112J type car
ii. 94% full
iii. 816°C 100% engulfing fire
iv. 100% defect
v. no direct contact between jacket and shell
vi. PRV set to 1.93 MPa, pop at 110% and reclose at 100%
vii. PRV capacity is 35,000 scfm at 120% pset

Figures 22 to 25 show the pressure, lading temperature, wall temperature and failure of this tank.

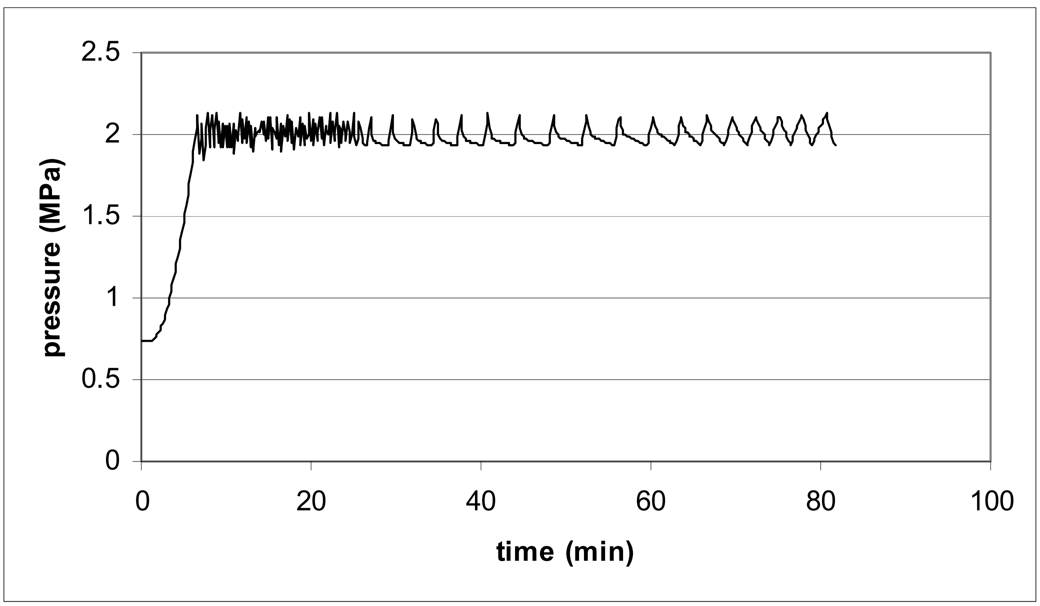

Figure 22 Predicted pressure for 112J tank with 100% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

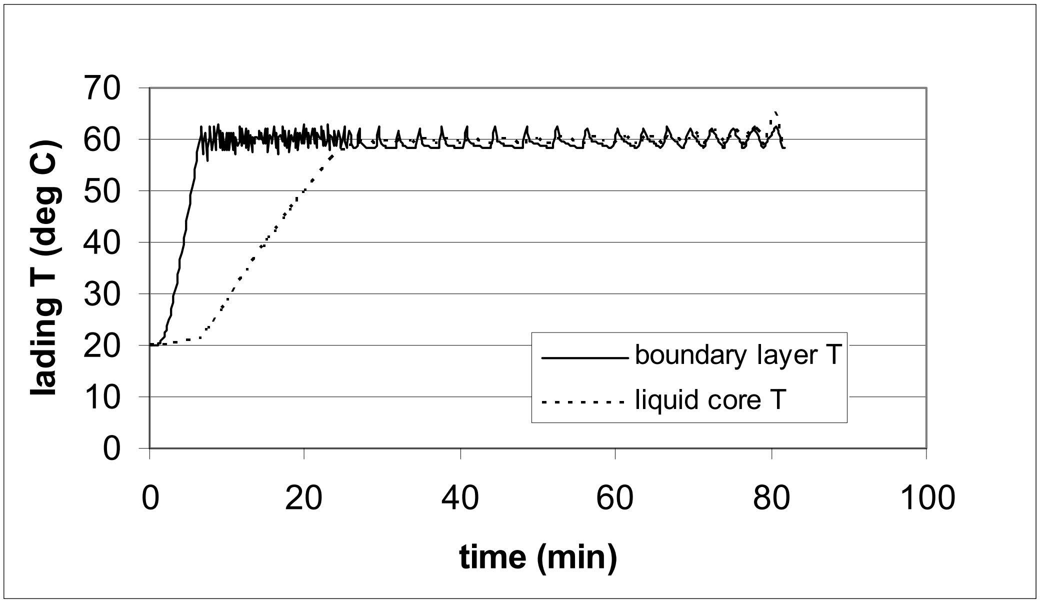

Figure 23 Predicted lading temperature for 112J tank with 100% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

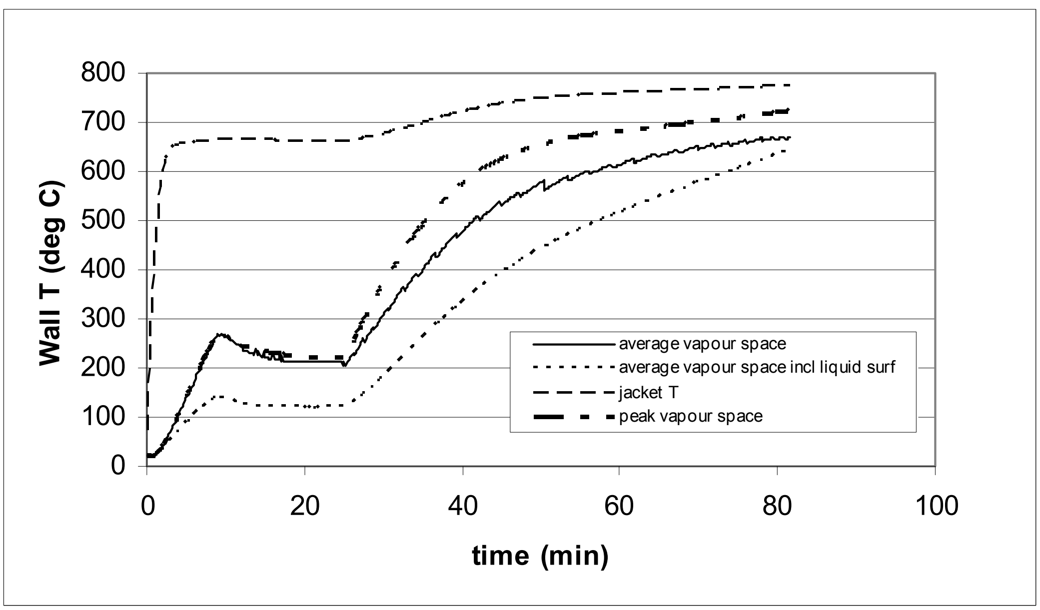

Figure 24 Predicted wall temperature for 112J tank with 100% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

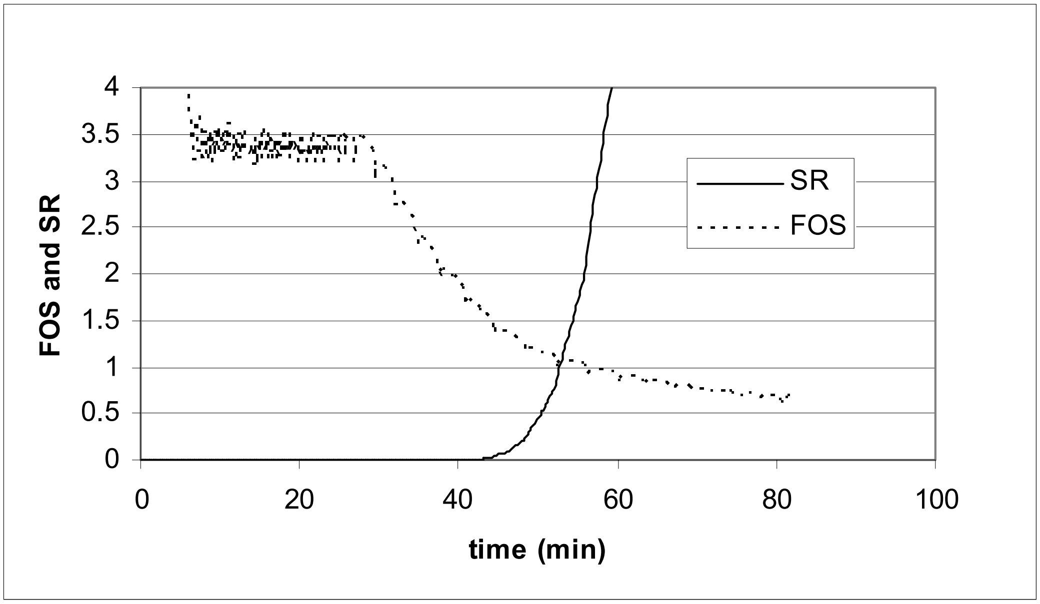

Figure 25 Predicted FOS and SR for 112J tank with 100% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

7.3.2 8% Defect

In this case the tank has 8% of its surface covered with defects. It is assumed that there is one large defect spanning the tank from top to bottom on one side with a length along the tank axis of about 3 m. The remainder of the tank is covered with 13 mm of thermal insulation with an average thermal conductivity of 0.295 W/mK. As before, the basic inputs were:

i. 112J tank
ii. 94% full
iii. 816°C fire
iv. 8% defect
v. PRV set to 1.93 MPa
vi. PRV scfm 35,000 at 120% pset
vii. insulation k = 0.295 W/mK

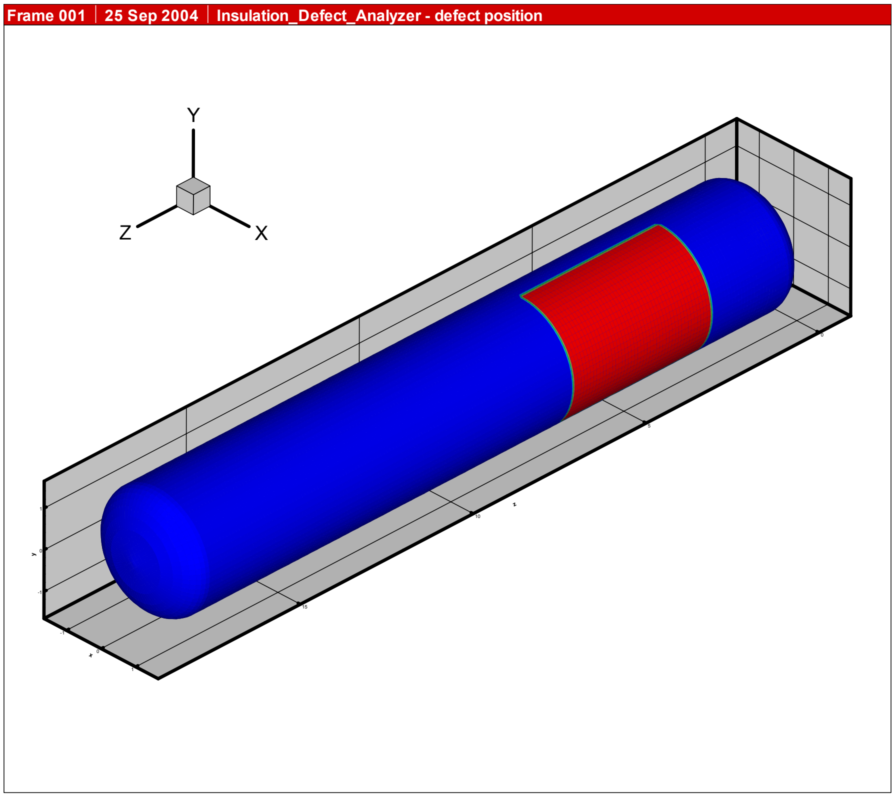

Figure 26 Graphic showing tank with one large defect covering 8% of tank surface area

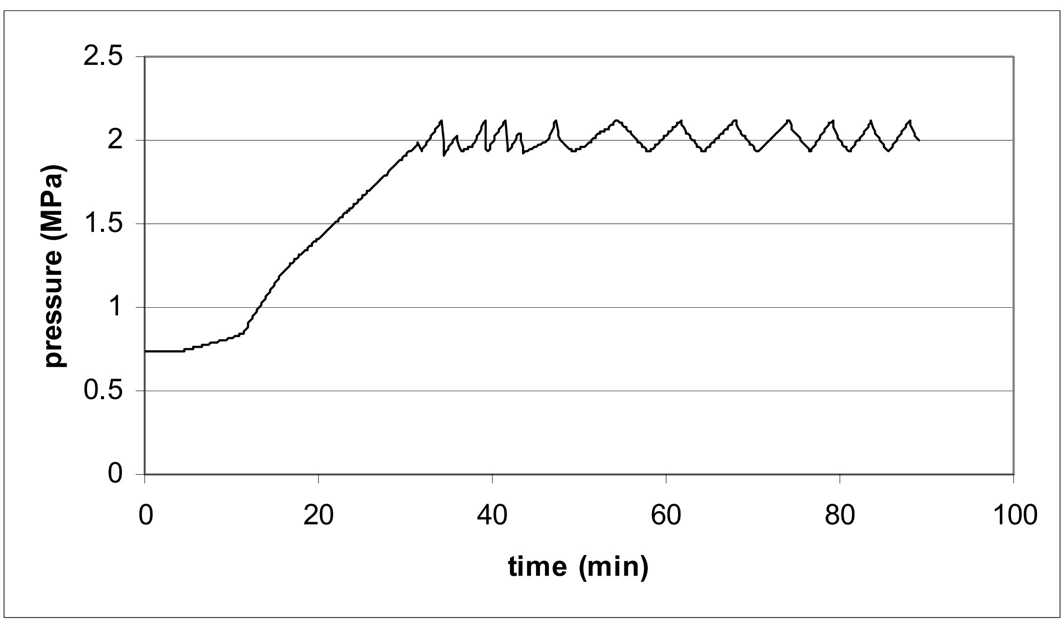

Figure 27 Predicted pressure for 112J tank with 8% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

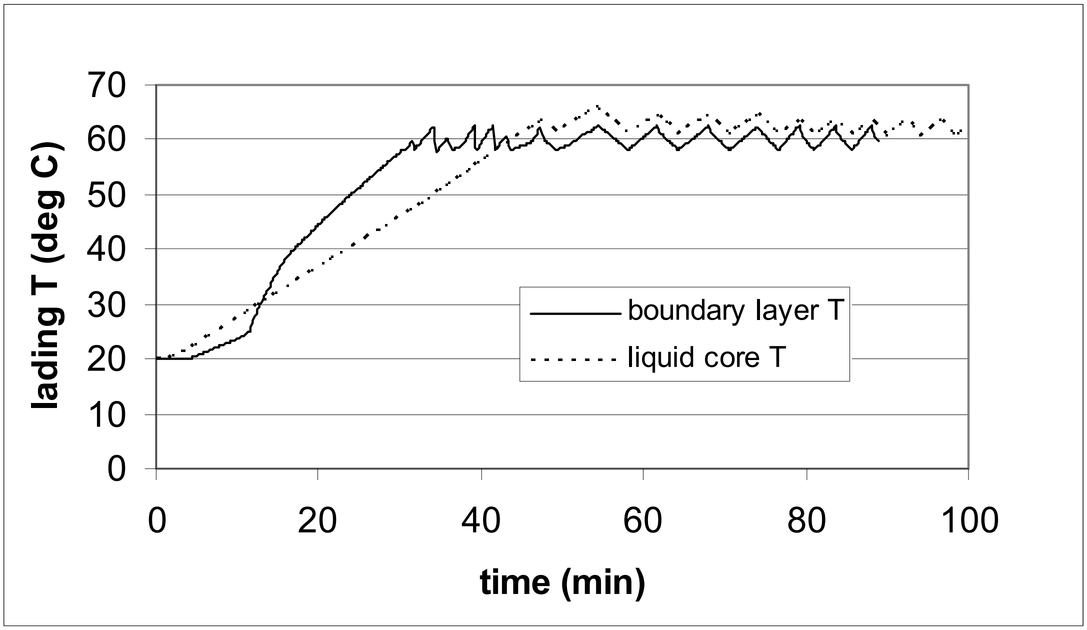

Figure 28 Predicted lading temperature for 112J tank with 8% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

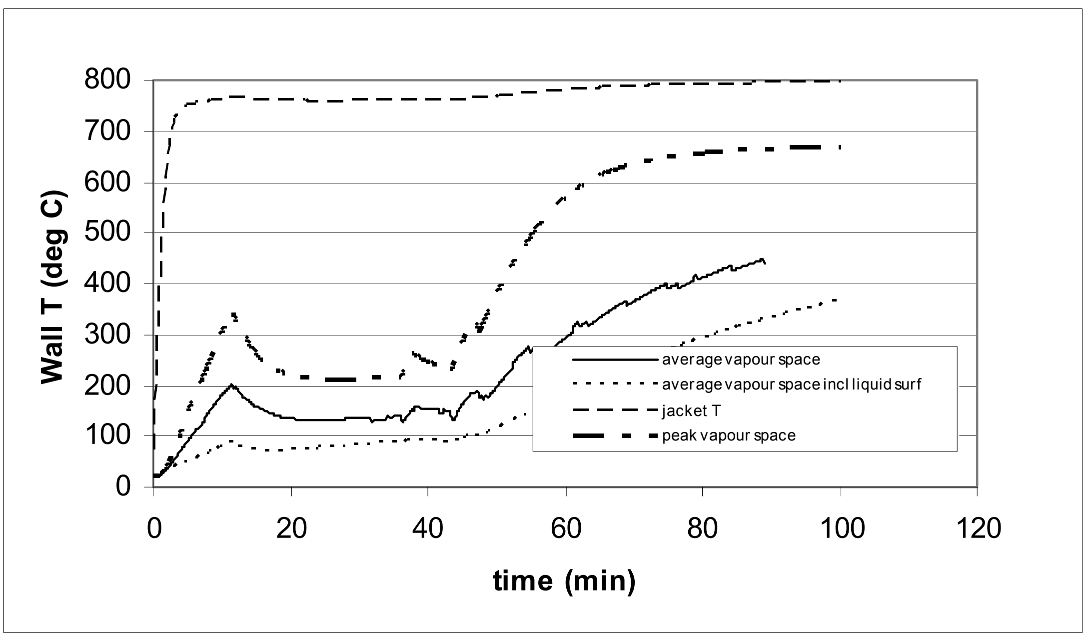

Figure 29 Predicted wall temperature for 112J tank with 8% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

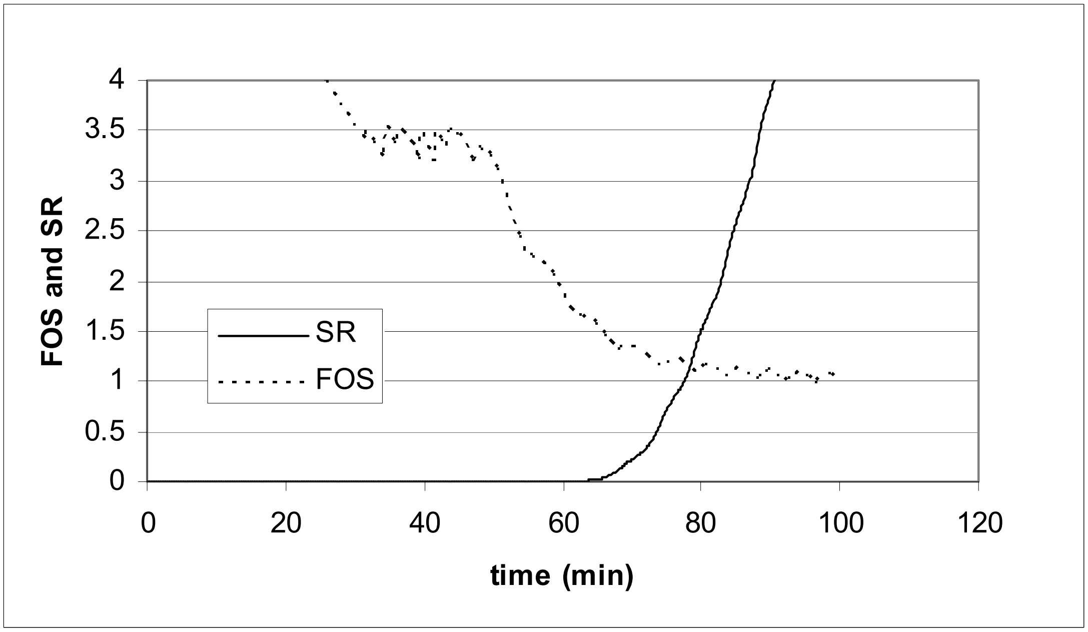

Figure 30 Predicted FOS and SR for 112J tank with 8% defect (fire T = 816°C, 94% fill, propane, PRV scfm = 35,000)

7.4 Simulation Summaries

The objective of this calculation is to determine the percentage of the tank surface that can include thermal protection defects. This is a function of the following most critical variables:

i) average thermal conduction properties of the intact thermal protection system
ii) PRV flow capacity
iii) fire effective blackbody temperature
iv) size and location of thermal protection defects

There are an almost infinite number of possibilities of defects. However, we will only get tank failure if there is at least one critical defect located near the top of the vapour space and liquid level drops below that defect.

For this set of simulations it was assumed the tank was equipped with a 35,000 scfm PRV. The fire was assumed to have an effective blackbody temperature of 816^{o}C. A critical defect is defined as one that is 1.2 m long along the tank axis or longer and is at least 0.4 m wide (based on plate fire testing [10] and 500 gal. tank testing [11]).

It should be noted that direct heat conduction links through the tank thermal protection system have not been included here. If these are significant for a specific tank then this should be accounted for in the calculation of the overall thermal conduction properties of the thermal protection system.

7.4.1 One Large Defect at Tank-Car Top

This case considered a tank-car with one large defect located at the top of the tank. The single defect covers about 1% of the tank surface. The thermal conductivity of the remaining 13 mm insulation was varied from 0.1 to 0.3 W/mK.

Table 3 summarizes the results.

Table 3 Summary of tank condition for one large defect (112J tank, propane, defect at top, 2 m long along tank axis by 0.75 m wide, about 1% of tank surface, tank initial fill 94%, tank initial temperature 20°C, fire T = 816^{o}C, PRV set = 280 psig, PRV capacity 35,000 scfm)

Insulation conductivity (13 mm insulation)	**Pressure at 100 minutes**	**Fill at failure or 100 minutes**	**Failure time**	**Comment**
k = 0.3 W/mK	2 MPa PRV cycling	50%	77 minutes	max allowable *k* to meet plate test standard
k = 0.2 W/mK	2 MPa PRV cycling	96%	no fail in 100 minutes	may be typical of "as new" system
k = 0.1 W/mK or less	1.5 MPa PRV passing liquid when shell full	>99% shell full	no fail in 100 minutes	probably not typical of real systems

From Table 3 we see that the failure of the tank depends very much on the condition of the remaining thermal protection. If the remainder of the tank has 13 mm insulation with an average k = 0.3 W/mK, then even a single defect covering 1% of the tank surface can lead to failure. It should be noted that a k = 0.295 is the upper limit of allowable thermal conductivity for a 13 mm blanket to pass the plate test standard for the thermal insulation material [4].

If the rest of the tank is protected with a thermal insulation 13 mm thick with average k = 0.2 W/mK, then the tank should survive if it only has one large (1%) defect in the vapour space. Note that the failure time is very sensitive to the *k* value in the range 0.2 to 0.3 W/mK.

The problem is that we do not know the condition of the remaining insulation unless we take samples and measure the thermal conductivity using some appropriate method (ASTM standard C-177, for example).

7.4.2 Many Small Defects Including at Least One Critical Defect (1.2 m x 0.4 m) at Tank Top

Table 4 summarizes failure times with varying levels of defects for the case where the rest of the tank is thermally protected with 13 mm insulation with an average k = 0.20 W/mK. The average insulation temperature in the liquid wetted wall areas should be about 450-500°C when the tank is engulfed in fire. Therefore a k = 0.2 W/mK should be conservative for a thermal protection system in otherwise good condition. This assumes the insulation is in direct contact with the jacket and primary shell.

Table 4 summarizes the simulation results.

Table 4 Summary of tank condition for many defects including at least one large defect at the top of the vapour space (112J tank, propane, assumed insulation k = 0.2 W/mK, tank initial fill 94%, tank initial temperature 20°C, fire T = 816°C, fill = 94%, PRV set = 280 psig, PRV capacity 35,000 scfm)

Fraction of tank surface covered with defect	Pressure at 100 minutes	Fill at failure or 100 minutes	Failure time	Comment
1%	1.5 MPa PRV passing liquid while shell full	> 99%	No fail in 100 minutes	pass
2%	2 MPa PRV cycling	> 99%	No fail in 100 minutes	pass
4%	2 MPa PRV cycling	79%	96 min	limit of allowable defect?
8%	2 MPa PRV cycling	79%	89 min	Fail
12%	2 MPa PRV cycling	75%	85 min	Fail

As can be seen from the Table 4, it only takes about 3-4% of the surface to be defective of insulation for the tank to fail within the 100 minute time frame. A 4% defect covers about 7 m^2 of the tank surface. This requires that there be at least one critical defect near the top of the tank.

Note that all of these cases involved a full-size PRV with flow capacity of 35,000 scfm at 120% of the PRV set pressure. In this case the PRV is much oversized for the heating conditions and therefore it has no difficulty keeping the pressure below 120% of the PRV set pressure.

Figure 31 gives a graphical summary of failure times for a range of percent defect and insulation *k* values. This effective *k* must also include all direct conduction links through the tank-car structure. As can be seen from the figure, the level of allowable defect is very sensitive to the condition of the remaining thermal protection system. If the system is in "as new" condition, then the average conductivity could be around 0.15 W/mK in fire conditions, which would allow a total defect area of about 8-9%. If the remaining insulation has an effective thermal conductivity of 0.3 W/mK, then one critical defect located at the top of the vapour space would theoretically be unacceptable. We do not have any data on in-service average thermal conductivity of the thermal protection systems.

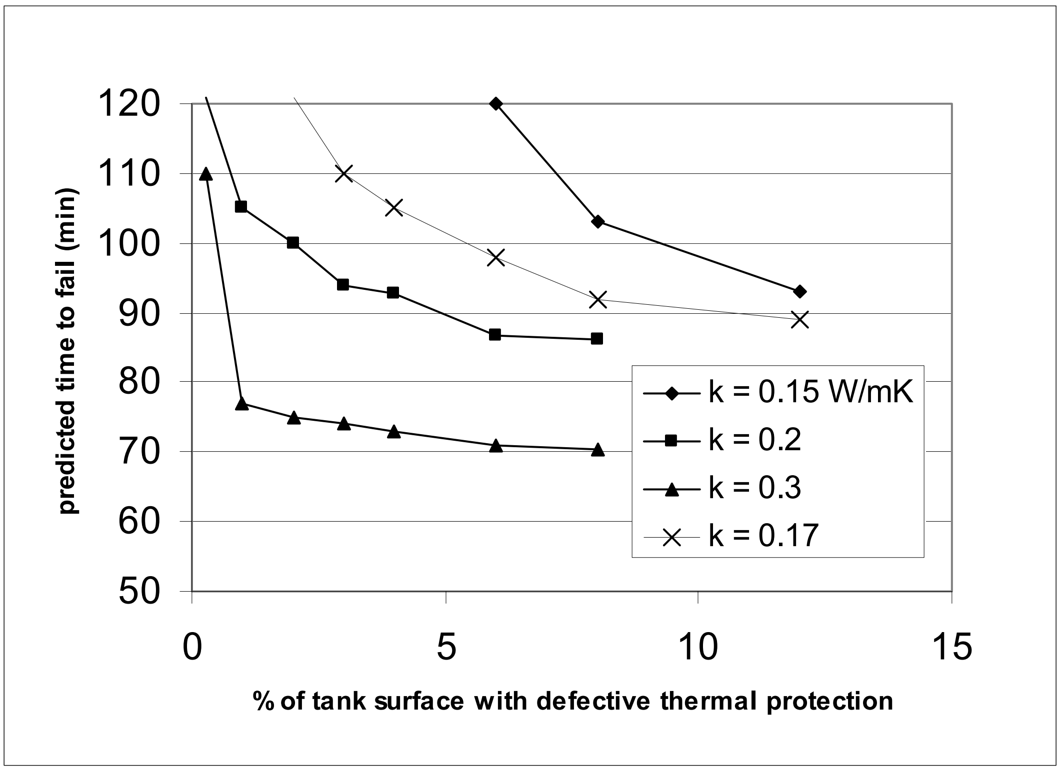

Figure 31 Failure time vs. percent defect from IDA 2.1 (for 112J Tank, 816°C fire, 94% initial fill, propane, PRV flow capacity 35,000 scfm, 13 mm ceramic insulation with thermal conductivity k (W/mK))

7.4.3 Effect of PRV Flow Capacity

Simulations were attempted for the case of a tank-car (same tank, fill, fire, initial conditions, etc.) with a PRV flow capacity of 3500 scfm, which is allowed by the AAR PRV sizing equations for thermally protected LPG tanks. The first case was for 0% defect (i.e., no defects in thermal protection) with k = 0.30 W/mK. This simulation failed to run for the 100 minute fire duration. At 80 minutes the program terminated due to internal errors. At the 80-minute time, the tank was shell full at 3.3 MPa pressure (570 psig). We are uncertain whether the shell full model (liquid and two-phase PRV flow) is working properly since we have no validation data for this case. The model result suggests that the 3500 scfm PRV may not be appropriate for the assumed heating conditions and thermal protection conductivity. Based on this outcome, we would recommend that no defects are acceptable for the 3500 scfm PRV case until further analysis is conducted.

8. Discussion

Sections 8.1 through 8.6 provide some additional discussion of specific details.

8.1 Allowable Defect

The allowable level of defects is a complex function of the following:

- i) fire conditions
- ii) tank wall material properties (stress rupture)
- iii) tank design (tank D, tank L, wall thickness W)
- iv) PRV performance (capacity, pop pressure, reclose pressure)
- v) remaining thermal protection system (overall k/w (conductance per unit area) including direct conduction links in the tank structure)
- vi) initial conditions (fill and temperature)

Table 5 shows the assumptions we have made for these variables in this study.

Table 5 Summary of main variables

	Assumed	Comment
Fire	816^{o}C Blackbody No convection 100% engulfing at time = 0	Minimum fire case – actual conditions are 871^{o}C plus or minus 56^{o}C (i.e., fire T not conservative).
Material properties	TC 128 B UTS 620 MPa As tested by Birk and Yoon [8]	Minimum for TC 128B is 550 MPa. (i.e., assumed material properties not conservative).
Tank design	D = 3 m, L = 18 m, wall thickness = 16 mm	
PRV	35,000 or 3500 scfm at 120% of Pset Pset = 1.93 MPa (280.5 psig) Pop assumed at 110% of Pset, reclose at 100%	
Remaining k/w	w = 13 mm Overall average k = 0.15, 0.175, 0.20, 0.3 W/mK	High temperature ceramic. Maximum *k* acceptable for plate test standard is 0.295 W/mK.
Initial conditions	T = 20^{o}C Fill = 94%	

The only variable that we do not know with a reasonable level of certainty is the overall effective conductance per unit area k/w. As it turns out, this is a very important variable since it can determine how quickly the liquid level drops to expose any defects located at the top of the tank. If the k/w is very good (low) then it is the total defect area that drives the rate of liquid lading loss through the PRV. If k/w is not so good (high) then it is this conductance that determines how fast the liquid level will drop and you only need one critical defect near the tank top to have failure.

8.2 IDA 2.1 and AFFTAC

IDA will generate results that are different than AFFTAC. The results from IDA are conservative, but reasonable based on comparison with fire test data.

The following differences between IDA 2.1 and AFFTAC are most significant:

i) IDA is a partially 3D model so it can model local thermal protection defects anywhere on the tank.
ii) The IDA code predicts that the tank will pressurize faster due to liquid temperature gradients (liquid is warmer near the walls and liquid surface).
iii) The PRV will cycle open and closed due to the pop action of the valve and this will allow the tank pressure (and stress) to rise and fall like it would with a real valve cycling between its pop and reclose pressure.
iv) The vapour space wall will heat up faster as the liquid level drops because convection and radiation parameters in the vapour space are more conservative.
v) The tank is less likely to go shell full in a fire situation because the PRV pops earlier due to saturation pressure (see item ii) and because the PRV entrains liquid as the liquid approaches the PRV inlet.
vi) Failure is predicted using high-temperature stress rupture data.

All of these add up to give a code that is more realistic in the prediction of time to failure.

8.3 IDA 2.1 Validation

The results generated by IDA 2.1 appear to be reasonable when they are compared to well-established benchmarks such as the RAX 201 fire test of an uninsulated tank-car [6]. We also have validation based on fire testing of a 500 gal. tank with thermal protection defects [11]. This suggests the code has the ability to predict performance over a range of realist scales.

In all cases the IDA 2.1 code predicted tank failure early by a few minutes compared to test results. In most cases the IDA code predicts high wall temperatures earlier than observed in tests and this is what causes the early prediction of failure. This difference in predicted wall temperature is most likely due to the fire buildup time in the tests. It is also known from fire testing [13] that large cool objects in fires actually cool the fire and reduce the heat flux. As the large object heats up, the cooling effect is reduced and the fire gets hotter. This is not accounted for in IDA 2.1.

All in all, we consider the predictions by IDA to be reasonable and conservative.

8.4 Location of Defect

If there is no large defect near the top of the tank (i.e., in the vapour space) then the failure will be delayed until the liquid level drops down to the defect area. As stated earlier, we must consider the fact that tanks can roll over in accidents and defects on the tank side can become defects in the vapour space when it is rolled over on its side.

We should also note that a tank rolled on its side will empty more quickly through the PRV because the PRV will be submerged in liquid. This means the liquid level will drop more rapidly, exposing more wall to a vapour space. We have not considered rolled tanks in this work.

8.5 PRV Capacity

The modeled cases all included full-sized PRVs with flow capacities of around 35,000 scfm. It was noted that defects are more critical on tanks with small PRVs (3500 scfm as allowed for thermally protected LPG tanks) because the defects could lead to pressure buildup.

The RAX 201 tank was equipped with a 34,900 scfm PRV. During the fire test the pressure reached 360 psig (128% of set). This probably means the PRV was slightly undersized for that test condition. If the tank had been covered with steel jacket with an air gap, the heat flux would have been reduced by about 50%, which means the PRV could be reduced to about 17,450 scfm. If the tank had been fully thermally protected, the heat flux would have been reduced by about 90% so the PRV could be around 3500 scfm. This assumes that the thermal insulation is in "as new" condition and has an average thermal conductivity of about 0.13 W/mK in the area covering the liquid wetted wall. This applies for ceramic fibre insulation at an average temperature of about 420^{o}C (i.e., tank wall T = 80^{o}C and jacket T = 800^{o}C).

If a tank-car has thermal protection defects in the liquid space, then the required size of the PRV would scale linearly with percentage of defect in the liquid space between these two values. For example, a tank with 10% defect in the liquid space would need a PRV of the following capacity:

$$scfm_{10\%defect} = 0.10(17450) + (1-0.1)(3500) = 4900scfm$$

These values are approximate and need further refinement.

8.6 Effect of Fill Level

A tank with thermal protection defects can fail if the defect area reaches dangerous temperatures. This can only happen if the liquid level drops below this area. The question is, how far must the level drop below this defect area for it to reach dangerous wall temperatures?

The RAX 201 tank [6] started with a fill of about 95% and failed when its fill level dropped to around 50%. The fire tests by Birk et al. [11] of 500 gal. tanks with thermal protection defects showed that tanks could fail with fill levels as high as 80% when the thermal protection defect was a the tank top. Birk et al. also did tests with 500 gal. tanks filled with water and found that a tank filled to 50% with water would have peak wall temperatures about 50^{o}C hotter than a tank filled to 80% with water. This is due to the expected cooling effect of the liquid.

With IDA 2.1, dangerous wall temperatures can be achieved at the top of the tank in defect areas when the fill level drops below about 80%.

9. Conclusions

The following conclusions were made:

i. The IDA 2.1 code has been reasonably validated against the summer 2004 fire testing of a 500 gal. propane tank (both baseline and with thermal protection defects).
ii. The IDA 2.1 code is in reasonable agreement with the RAX 201 fire test results of a full-scale unprotected rail tank-car.
iii. There are some differences between the IDA 2.1 model and test results. IDA 2.1 tends to predict a more rapid increase in wall temperatures, which leads to failure prediction a few minutes earlier than observed in tests. This can be partly explained by how the fire is modeled. Real fires take some time to build up whereas in IDA the fire is on 100% at time = 0.
iv. The model appears to be reasonable and conservative in the prediction of tank failure.

This program has not been fully validated and should therefore be used with caution.

The following conclusions were made based on the modeling reported herein. It was assumed that the critical thermal protection defect size is 1.2 m measured along the tank car (112J) axis by 0.4 m wide as determined from the fire testing conducted by Birk et al. [11].

i. A critical thermal protection defect can lead to tank rupture if it is located in the tank-car vapour space during a fire engulfment accident.
ii. The failure of a tank-car with thermal protection defects depends not only on the size and location of defects, but also on the quality of the remainder of the thermal protection system that is not defective (including all direct condition links in the tank structure). The better thermally protected the tank is, the more capable it is of surviving with local thermal protection defects. This is because the overall thermal protection system determines how fast the liquid level will drop when the tank is exposed to fire.
iii. The total allowable defect area is very strongly affected by the area average thermal conduction properties (i.e., k/w where k = thermal conductivity and w = insulation thickness) of the tank thermal protection insulation during fire conditions. It is estimated that this value of thermal conductivity is in the range of 0.15 to 0.3 W/mK for high-temperature ceramic blanket insulation under fire exposure conditions.
iv. A tank with 13 mm ceramic blanket thermal protection with an area average thermal conductivity of 0.15 W/mK (at fire conditions) can probably allow up to 8-9% of its surface to be defective of thermal protection. This assumes that there is at least one critical defect in the vapour space. This also assumes that the PRV

has a flow capacity greater than about 5000 scfm at 120% of the PRV set pressure (280.5 psig assumed here).

v. A tank with 13 mm ceramic blanket thermal protection with area average thermal conductivity of 0.20 W/mK (at fire conditions) can probably allow up to 4% of its surface to be defective of thermal protection. This assumes that there is at least one critical defect in the vapour space. This also assumes that the PRV has a flow capacity greater than about 4000 scfm at 120% of the PRV set pressure (280.5 psig assumed here).

vi. A tank with 13 mm ceramic blanket thermal protection with area average thermal conductivity of 0.30 W/mK (at fire conditions) cannot allow any critical defects (i.e., longer than 1.2 m along tank axis by 0.4 m wide). This effective thermal conductivity is the maximum allowable for a 13 mm blanket that meets the original plate test standard for thermal protection systems. If a tank has this average thermal conductivity, then a 3500 scfm PRV is probably too small for that tank.

vii. If there are no defects larger than 1.2 m x 0.4 m, then more defect area may be acceptable, but this should be determined on a case-by-case basis by running the IDA 2.1 code for the specific tank. For this case, insulation samples should be taken so actual *k* values can be measured. At least 10 samples should be taken so that a truly representative average *k* can be determined.

viii. 112J type tank cars equipped with 3500 scfm PRV should not be allowed to have any defects unless the overall thermal protection properties can be defined.

The reader is reminded that this study did not consider the following:

i) end failures
ii) defective PRVs
iii) defects in primary shell
iv) corrosion
v) impact damage
vi) torching fires
i) rolled tanks
ii) hard contact between the jacket and tank shell

10. Recommendations

The results described in this report depend very strongly on the details of the heat transfer in the vapour space when the tank is at high fill levels. We have almost no data to validate these models in any detailed way.

The following work is needed based on the analysis presented in this report:

i) Conduct fire tests of tanks at high fill levels, including thermal protection defects.
ii) Measure typical *k* valves for "as installed" insulation in thermal protection systems. This must be measured under temperature conditions that are expected in fire accidents.
iii) Determine how often there is direct contact between the tank jacket and primary wall.
iv) Determine the behaviour of PRVs during shell full conditions (i.e., how they open and close, flow capacity of liquid and two-phase, etc.).
v) Measure typical emissivities for the inside wall of old and new tanks.
vi) Measure or obtain reflection characteristics of propane liquid surface.
vii) Investigate current PRV sizing formula for thermally protected tanks. Simulations suggest that current sizing requirements may not be conservative.

The IDA 2.1 code continues to evolve. The following tasks are suggested for ongoing work:

i) Improve two- and three-node thermal models.
ii) Improve vapour space radiation model.
iii) Include two-node vapour space model for cases where late PRV action is expected.
iv) Validate shell full model assumptions (PRV flow, etc.).
v) Include other commodities in the code.
vi) Improve user interface.

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Appendix A: Overview of Modelling Approach

Wall Temperatures

The temperature at each wall zone is calculated from an energy balance at that wall zone. The wall temperature is modelled as 2D in space with the temperature gradient through the tank wall thickness ignored.

Wall temperatures are calculated using a finite difference approach in space and time. The energy balance at a wall zone considers the following:

i) heat conducted through insulation or radiated directly from steel jacket if insulation is not present
ii) heat removed from back side of wall in vapour space by thermal radiation and convection
iii) heat removed from back side of wall in liquid space by convection and boiling
iv) heat conducted to/from neighbouring wall zones

In simple equation form this is:

$$\frac{dT}{dt} = \frac{(q_{fire} - q_{backside} - q_{cond})A}{\rho c A w}$$

where

T = average wall volume temperature
q = heat flux due to fire, backside convection and radiation and conduction to neighbouring wall zones
A = surface area of wall zone
W = thickness of wall zone
C = wall material specific heat

In this analysis the temperature gradient through the wall has been ignored to speed up computation time. This gradient is usually small on the order of 10°C. Wall failure temperatures are typically in the range of 600 – 700°C.

The temperature change in a zone accounts for the mass and thermal capacity of the wall. The thermal capacity of the insulation is ignored.

The steel jacket temperature is calculated in the same way but includes the following:

i) heat radiated and convected from fire
ii) heat conducted from back side of jacket to insulation or radiated to wall if insulation is not present
iii) heat conducted to/from neighbouring jacket zones

Vapour Space Heat Transfer

Vapour space heat transfer is a critical part of the model because it must accurately predict vapour space wall temperatures for the failure analysis. A simple rule of thumb is that the tank will fail when the pressure is at the PRV set pressure and the vapour space wall reaches about 640^{o}C.

The heat transfer in the vapour space is due to convection and thermal radiation.

At high wall temperatures thermal radiation dominates. Free convection occurs when the PRV is closed. Forced convection comes into play when the PRV is open. Data from various tests suggest that the convection coefficient in the vapour space is not very significant.

Thermal radiation is very important at the high wall temperatures in the vapour space. Table A-1 gives a summary of the effects.

Table A-1: Example Distribution of Heat Transfer in Vapour Space

	Radiation E = 0.9	Free Convection H = 5 W/m^{2}K	Forced Convection H = 20 W/m^{2}K
640^{o}C wall with 60^{o}C vapour and liquid surface	35 kW/m^2	3 kW/m^2	12 kW/m^2
	70%	6%	24%

The following heat transfer assumptions are used for the vapour space.

Radiation

The vapour space is divided into three zones as follows:

i) the wall zone (zone 1)
ii) the liquid surface (zone 2)
iii) the remaining wall in the vapour space (zone 3)

The absorption of radiation by the vapour is ignored. An average vapour space wall temperature is used for the remainder of the vapour space wall.

The view factor from the wall zone to the liquid surface is calculated using Hottel's crossed strings method. It is assumed the wall zone 1 does not see itself.

Convection

Convection is modelled based on empirical correlations for free and forced convection for simple surfaces (see, for example, Holman[1]).

When the PRV is closed, the free convective *h* is calculated from the following:

$Nu = 0.58\ (Gr\ Pr)^{0.20}$

where

Nu = Nusselt number = hD/k
Gr = Grashof number = $g\rho^2\beta(Tw-Tvap)L^3/\mu^2$
Pr = Prandtl number = $C_p\mu/k$

μ = molecular viscosity
k = thermal conductivity
c_p = specific heat
h = convective heat transfer coefficient
L = characteristic length (arc length from liquid surface to tank top)

For forced convection a mean vapour velocity in the vapour space is estimated based on the vapour space cross section and PRV mass flow. When the PRV is open, the PRV is divided in half, assuming the flow approaches the PRV from both ends of the tank. The average flow velocity is determined from the vapour space cross section and the average vapour density. The forced convective coefficient is determined from correlations for pipe flow using the following relations from Holman:

$Nu = 4.0 + 0.025\ (Re\ Pr)^{0.25}$

where

$Nu = hD/k$
Re = Reynolds number = $\rho UL/\mu$
U = mean vapour space velocity
L = vapour space hydraulic diameter

The free and forced convection are combined as follows to determine an overall vapour space convection:

$$h = \sqrt{h_{free}^2 + h_{force}^2}$$

The convection coefficient is not assumed to be uniform over the entire vapour space. The convection is assumed to be weaker near the tank top where warmer vapour will rise when the PRV is closed. The free convective boundary layer is also thicker near the top. In IDA 2.1 the convection varies from 100% at the liquid level to 0% at the tank top.

Lading Thermal Model

This is the most complex model in IDA.

The thermal model for the insulation defect analyzer must be able to model the basic physics of the tank heating process. It should be able to account for the following in a reasonable way.

i) localized heating in the liquid space
ii) localized heating in the vapour space
iii) the effect of thermal gradients in the liquid on the pressurization of the tank
iv) the effect of thermal gradients in the liquid on the blowdown of the tank when the PRV is activated

If it can do the above, then it should be able to predict the tank pressure and how the liquid level will drop as the PRV opens. This means it should be possible to predict the wall temperatures with some accuracy and this will allow failure prediction with some accuracy.

The above could be done using a full field type analysis (i.e., transient 3D CFD), but this approach would require extensive computer power and would be very expensive to implement. This type of study has been done by Birk and Yoon [2]. This is not what is desired in this case.

The objective is to have a fast and efficient analysis tool that can be run in a reasonable length of time on a typical desktop computer. In this case, a run time of under 60 minutes is desirable. It is also desirable to be able to set up a run with minimal inputs from the user.

This will be accomplished using a multi-node type thermal model. This model is then solved using the laws of conservation of mass and energy.

Multi-node Thermal Model

In a multi-node type analysis, the tank contents are divided into a number of isothermal volumes or nodes. Here is a brief description of how this has been done for this version of IDA.

The tank internal space is divided into two volumes as follows:

i) vapour space and warm liquid boundary layer
ii) cool liquid core

It has been assumed that the warm layer is the heated liquid boundary layer near the tank wall and that this flows by natural convection to the top of the liquid where it forms a stable layer. The cool liquid core is the remainder of the liquid volume. This is in agreement with extensive fire test data from tank tests (see, for example 3, 4, 5].

These volumes are related by the following volume considerations:

i) total liquid space = warm liquid layer + cool liquid core
ii) tank total space = total liquid space + vapour space

In other words, the tank volume is assumed fixed and is filled by the liquid and vapour.

The cool and warm liquid volumes are determined by the heated tank surface area and a boundary layer thickness over that heated area, i.e., this boundary layer thickness will depend on the lading thermal and transport properties and the heat flux.

Heat is transferred between the heated and unheated liquid volumes by convection in the liquid space. In other words, there is a mass transfer between the heated and unheated liquid volumes.

It is assumed that the heat added to the vapour space by the heated wall in the vapour space mixes uniformly in the vapour to give an overall average vapour space condition.

It is assumed that the heated liquid volume (i.e., the warm layer) communicates directly with the vapour space. This means there is mass transfer between the heated liquid volume and the vapour space. This mass transfer between the liquid and vapour spaces is due to evaporation and/or condensation. For now it is assumed that the vapour space and warm layer are in thermal equilibrium at all times (i.e., they have the same temperature).

The above model results in two separate energy equations for the three volumes.

When the PRV opens, it is assumed the energy comes from the vapour space and the warm liquid layer. It is also assumed that the mixing between the heated and unheated liquid spaces is increased by this PRV action.

The above model results in a need to set factors to account for the heated liquid boundary layer volume and the mass transfer between the heated and unheated liquid. These have been set based on available test data. This means that this model is partially empirical and therefore not fully validated because of the limited data available. The validation cases used are shown later in this appendix.

Two-Node vs. One-Node Thermal Models

The thermal model has been formulated to account for thermal gradients in the tank lading. The current AFFTAC [6] model assumes that the entire tank contents are at the same temperature and are in saturated equilibrium. This assumption leads to slow tank pressurization and in some cases the false prediction of a shell liquid full condition.

Test results for unprotected and protected tanks show that the tank will pressurize faster than predicted by such a model (see Birk [7]) and therefore the single-node model cannot predict time to PRV action properly. For example, the RAX 201 engulfing fire test of a full-scale non-thermally protected tank car shows that the PRV would open in about 2 minutes, but a single-node thermal model would predict a PRV open time of about 10 minutes based on saturation pressure. With the single-node thermal model, the RAX 201 tank will go shell full after about 5 minutes. This demonstrates how a single-node model cannot be expected to give accurate predictions of tank pressurization.

Two-Node Thermal Model with Local Heating by Defects

There is no data for localized heating of a tank due to insulation defects; therefore, this model has been formulated based on the following assumptions. These assumptions come from limited test results that indicate trends but not detail.

i) The localized heating will result in thermal gradients that cause early opening of the PRV in a similar fashion to tanks engulfed in fire.

ii) A tank heated near the bottom of the liquid will pressurize more slowly than a tank heated near the liquid surface.

iii) The slowest possible pressurization would be assuming the tank contents are isothermal (like AFFTAC model).

iv) A tank heated only in the vapour space will pressurize slowly.

These assumptions are supported by the recent CFD analysis by Yoon and Birk [8].

Equations

This section gives a summary of the equations solved in this thermal model. The following variables have been used in the model.

m_g = vapour mass
m_{fc} = cool liquid mass
m_{fs} = saturated liquid (warm layer)

$T_{fs} = T_g$ = warm layer and vapour space temperature
T_{fc} = cool liquid temperature

The following equations are solved using a finite difference approach in time. At $t = t_o$ the conditions are known. A time step of dt is used to determine the conditions at $t = t_o + dt$. Currently the time step used in IDA is between 0.1 and 10 sec. The size of the time step is determined by the tank size (diameter) and is related to the PRV cycling time. For example an unprotected 112J tank car with D = 3 m can be modelled using a time step of around 1 second.

Conservation of Mass

The mass in the tank must equal the mass in the tank at the beginning of a time step minus the mass leaving through the PRV during the time step.

$m = m_{fs} + m_{fc} + m_g$

and

$m_2 = m_1 - dm_{prv}$

or

$m_{fs2} + m_{fc2} + m_{g2} = m_{fs1} + m_{fc1} + m_{g1} - dm_{prv}$

As mass leaves the tank, some mass moves from the cool core into the warm layer and some mass moves from the warm layer into the vapour space. It is assumed here that only vapour flows out through the PRV. The following equations apply:

$dm_{fcfs} = m_{fc1}-m_{fc2} = - dm_{fc}$

$dm_{fsg} = m_{g2} - m_{g1} = - dm_g$

$dm_{fs} = m_{fs2} – m_{fs1} = - dm_{prv} + dm_{fcfs} - dm_{fsg}$

where

dm_{fcfs} = mass transfer to warm layer from cool layer
dm_{fsg} = mass transfer from warm layer to vapour
dm_{prv} = mass transfer out of tank through the PRV

There is also a mixing mass transfer between the cool liquid and warm liquid. It is assumed this is an exact mass balance and it makes no contribution to the above equations. It does make a contribution to the energy balance.

Core Liquid and Warm Liquid Volumes

It is assumed that the partition of the liquid is a function as follows:

$V_{fs} = \lambda V_f$

V_{fs} = volume of the warm liquid layer
V_f = total liquid volume

and

λ = f(fluid, tank geometry, scale, heating conditions, PRV action)

In this case the fluid, tank geometry and scale are set. Heating conditions and PRV action are variable. Since there is no data to define this function it will be assumed for now that λ is a constant to be determined by calibration of the model.

An alternative to a constant would be to assume the warm layer volume is related to the volume of heated liquid and this is related to the heated surface area by a boundary layer thickness δ. In this case:

$\lambda = A\delta/V_f$

where

A = heated wall area wetted by liquid

In this model the δ is a constant to be calibrated with available data.

Conservation of Energy

Each node has its own energy equation as follows:

Vapour and Warm Liquid Layer

$$\dot{Q}_g \Delta t + \dot{Q}_{fs} \Delta t = m_{fs2} u_{fs2} - m_{fs1} u_{fs1} + m_{g2} u_{g2} - m_{g1} u_{g1} + \dot{m}_{prv} h_{prv} \Delta t$$
$$- \dot{m}_{fcfs} h_{fc} \Delta t + \dot{m}_{mix} (h_{fs1} - h_{fc1}) \Delta t$$

where

m_{prv} = PRV mass flow rate
m_{fcfs} = mass transferred from cool core to warm layer
m_{mix} = mixing between the cool core and warm layer due to free convection and boiling
h_{prv} = enthalpy of mass leaving through PRV

Q_g = heat transfer rate to vapour space
Q_{fs} = heat transferred to the warm liquid layer
m_{fsg} = mass flow rate from warm liquid to vapour

Core Liquid

The core liquid energy balance is as follows:

$$\dot{Q}_{fc}\Delta t = m_{fc2}u_{fc2} - m_{fc1}u_{fc1} + \dot{m}_{fcfs}h_{fc}\Delta t - \dot{m}_{mix}(h_{fs1} - h_{fc1})\Delta t$$

where

m_{fg} = mass mixing rate between cool liquid and warm liquid
Q_{fs} = heat transfer rate to heated liquid space
m_{fc} = mass mixing rate between liquid and vapour
h_{fc} = enthalpy of cool liquid
h_{fs} = enthalpy of heated liquid

These two nodes are coupled by the mass transfers between them. These mass transfers are driven by evaporation and condensation between the liquid and vapour and by mixing between the cool and warm liquid.

Mixing Mass Flow

The mixing mass flow m_{mix} between the liquid boundary and core is caused by free convection when the PRV is closed and by boiling driven convection when the PRV is open. Over time, with the PRV acting, this mixing will make the liquid isothermal. For example, we know from the RAX 201 test that it took about 8 minutes of PRV action to fully mix the liquid contents to make it near isothermal. Once the liquid is isothermal at the saturation temperature for the PRV set pressure, all of the heat added to the liquid goes into boiling the liquid for maintaining the PRV flow. Before this time, part of the heat added goes into generating the PRV flow and part goes into heating up the liquid core.

When the core liquid is cool compared to the boundary (i.e., when thermal gradients are large) the vapour bubbles generated at the wall collapse into the boundary (subcooled boiling) and this heat rises slowly towards the liquid level. This plume also adds heat to the core as discussed earlier. As the core heats up the process becomes one of saturated boiling and the bubbles no longer collapse into the cooler liquid but rise rapidly to the surface causing strong mixing.

The mixing relationships are assumed as follows:

For free convection (no PRV action):

$$\dot{m}_{mix-free} = f(A_h, \dot{q}, T_{fs}, T_{fc}, z)$$

where

A_h = heated wall area
q= local wall heat flux
T_{fs} = liquid boundary temperature
T_{fc} = liquid core temperature
Z = position of heating relative to liquid surface

For forced mixing during PRV action:

$$\dot{m}_{mix-forced} = f(\dot{m}_{prv}, x_{prv,} T_{fs}, T_{fc}, z)$$

where

m_{prv} = PRV mass flow
x_{prx} = flow quality entering PRV

Various functional relationships have been used in the code to model these mixing mass flows. Parameters used in these functions are set using experimental data where available. These are set to give agreement for the following important model outputs:

i) time to first PRV action
ii) time to isothermal liquid

This part of the code is continuously being modified and improved as data becomes available.

Fixed Volume

The total of all the mass volumes must equal the tank volume, i.e.,

$V = m_{fs}\, v_{fs} + m_{fc}\, v_{fc} + m_g\, v_g$

PRV Mass Flow

The PRV mass flow assumes isentropic homogeneous flow. For propane vapour, the PRV will choke with a pressure ratio of about 0.61. The mass flow in this case is calculated as follows for vapour flow:

i) The h_o and s_o are calculated assuming a quality of x_o =1.0 for the tank pressure = P_o.
ii) The pressure at the PRV minimum area (at the critical section) P_c is assumed to be 0.61 P_o.
iii) The new h_c and v_c are calculated at the PRV minimum area assuming the P_c = 0.61 P_o and $s_c = s_o$.
iv) The velocity at the critical section is $U_c = (2000(h_o - h_c))^{0.5}$.
v) The mass flow = $C_d A_c U_c / v_c$.

For vapour flow a C_d of 0.8 is used.

The above analysis can easily be modified to consider homogeneous two-phase flow or frozen two-phase flow. This only requires that the quality x_o be set.

PRV performance plays a critical role in the response of a tank to fire impingement. Testing of PRVs has shown their performance to be highly variable [9].

For modelling, the PRV is assumed to pop open at some fraction of the PRV set pressure (e.g., 110% of Pset) and it is assumed to reclose at some lower fraction of the set pressure (e.g., 90% of Pset). When the PRV is open, it will allow the flow of liquid or vapour or two-phase fluid depending on the valve position, tank roll and liquid level in the tank. Due to the large pressure ratio, the flow is usually choked (see, for example, [10]).

The flow is determined assuming compressible isentropic homogeneous flow. A flow coefficient is applied to account for non-isentropic behaviour. The basic flow equation is:

$$\dot{m} = \rho U A C_d$$

where

m = mass flow rate
ρ = fluid density
U = flow velocity at minimum area
A = minimum area
C_d = flow coefficient

The flow velocity is determined from the energy equation:

$$U = \sqrt{2(h_o - h_2)}$$

where

h_o = fluid enthalpy at stagnation conditions
h_2 = enthalpy at PRV minimum area

The enthalpy at the PRV throat is calculated assuming an isentropic pressure drop from stagnation conditions to the PRV throat. The pressure drop is that needed for maximum mass flow or choking. The stagnation quality x_o can be anywhere from 0 to 100%.

When the quality of the vapour entering the PRV is less than one, then the following method is used to calculate the liquid and vapour flows.

i) The flow inlet void fraction is calculated from the equation for homogeneous two-phase flow from $\alpha =$.
ii) The PRV channel area taken up by vapour = αA and the flow channel area taken by liquid is $(1-\alpha)$A.
iii) The vapour flow is calculated the same as listed above for pure vapour flow.
iv) The liquid flow is calculated assuming the liquid is frozen in its liquid state (i.e., non-equilibrium frozen flow). A Cd = 0.6 is used for the discharge coefficient.

The two-phase flow model is used when the liquid level rises near the inlet of the PRV. It is assumed that liquid is entrained in the vapour flow. This is described in more detail later in this appendix.

Overall Solution Procedure

We begin with the cool core liquid. This allows us to solve for the new core conditions from the energy equation.

$$u_{fc2} = \frac{\dot{Q}_{fc}\Delta t + m_{fc1}u_{fc1} + \dot{m}_{mix}(h_{fs1} - h_{fc1})\Delta t}{m_{fc1} - \dot{m}_{fcfs}\Delta t}$$

where the m_{fcfs} is calculated from a previous time step. At t = 0 this flow equals 0. From the u_{fc2} above we can find T_{fc2} and v_{fc2} from the property tables for propane.

The total core volume becomes $m_{fc2}v_{fc2}$. This volume is removed from the tank volume to give us the volume occupied by the vapour and the warm liquid layer.

$V_w = V_{tank} - m_{fc2}v_{fc2}$

From the energy equation for the warm region we see that:

$U_{w2} = m_{fs2}u_{fs2} + m_{g2}u_{g2}$

and from the energy balance for the warm region we see that the above is:

$$U_{w2} = \dot{Q}_g \Delta t + \dot{Q}_{fs} \Delta t + m_{fs1}u_{fs1} + m_{g1}u_{g1} - \dot{m}_{prv}h_{prv}\Delta t + \dot{m}_{fcfs}h_{fc}\Delta t - \dot{m}_{mix}(h_{fs1} - h_{fc1})\Delta t$$

To solve this we must guess a new T_{fs2}, which also equals T_{g2}. The correct T_{fs2} will satisfy the following equations for energy and volume:

$U_{w2} = m_{fs2}u_{fs2} + m_{g2}u_{g2}$

and

$V_{w2} = m_{fs2}v_{fs2} + m_{g2}v_{g2}$

From continuity we know that:

$m_{fs2} + m_{g2} = m_{fs1} + m_{g1} - m_{prv} + m_{fcfs}$

We can use the above to eliminate m_{fs2} from the energy equation, by noting:

$m_{fs2} = m_{fs1} - m_{g2} + m_{g1} - m_{prv} + m_{fcfs}$

and putting this into the volume expression gives:

$V_{w2} = m_{g2}v_{g2} + (m_{fs1} + m_{g1} - m_{g2} - m_{prv} + m_{fcfs})v_{fs2}$

From this we solve for m_{g2}:

$$m_{g2} = \frac{V_{w2} - (m_{fs1} + m_{g1} - m_{prv} + m_{fcfs})v_{fs2}}{v_{g2} - v_{fs2}}$$

We can put this into the expression for energy to find T_{fs2} and from this all other terms can be found.

Now that we have the new condition, we can calculate the new total liquid volume from:

$V_{liq} = m_{fs2}v_{fs2} + m_{fc2}v_{fc2}$

This new liquid volume determines the new liquid level. We now must reallocate the liquid masses to satisfy the model for the size of the cool core and warm liquid layer.

The final new zone volumes become:

$m_{fsnew} = KV_{liq}/v_{fs2}$

$m_{fcnew} = (1\text{-}K)V_{liq}/v_{fc2}$

and the mass moved from the core to the warm layer becomes:

$m_{fcfs} = m_{fc2} - m_{fcnew}$

Shell Full Condition

If a tank starts off with a high fill level and if it is heated by fire, the liquid will expand and may fill the tank. This is called the shell full condition. In this case it is expected that the PRV will open partially to allow a small liquid flow and this stops the tank from rupturing. This is a non-linear process and is very difficult to model in detail unless very small time steps are used, which requires large computer resources. The current version of AFFTAC allows the tank to go shell full and calculates the necessary liquid flow to maintain the pressure and the PRV pop pressure.

In the full-scale fire test of a non-thermally protected tank (RAX 201, [5]) the tank started off 96% full of propane at 21^{o}C. In this case we would not be surprised if the tank went shell full during fire engulfment. However, the test data shows that the tank approached the shell full condition, but it never went shell full. This is shown in the measured vapour space wall temperatures. If the tank were to go shell full, all of the tank wall would be cooled and we would not see high vapour space wall temperatures. RAX 201 saw high wall temperature in the vapour space during the entire fire test.

Why did this tank not go shell full? With RAX 201, the PRV opened first after about 2 minutes of fire exposure. This happened because of liquid temperature stratification (i.e., the warm liquid boundary pressurized the tank faster to cause early PRV opening). With the early opening of the PRV, the tank did not go shell full.

As the liquid level rises and approaches the PRV inlet, liquid will be entrained into the PRV when it is open. This causes a high mass flow, which tends hold back further rising of the liquid level.

With slow heating in the case of local thermal protection defects it is possible the tank will go shell full. In this case the opening of the PRV by hydrostatic pressure must be accounted for. Therefore, the following two approaches are used in IDA 2.1 for very high fill levels or shell full.

Fills > 95%

In IDA 2.1 a simple liquid entrainment model is used for an upright tank as follows:

i) If the liquid level is more than 0.05D (i.e., 0.05 x the tank diameter D = 150 mm for 3 m tank) away from the inlet of the PRV, then the quality of the vapour entering the PRV is 100% (i.e., no liquid).
ii) If the liquid level rises to within 0.05D or less from the PRV inlet, then the quality of the vapour entering the PRV is 10%.

When the flow is two-phase vapour and liquid, the flow rate is modelled using the following assumptions:

i) isentropic vapour flow
ii) frozen liquid flow
iii) pressure ratio for propane flow P throat = 0.61 P source
iv) vapour and liquid flow areas based in inlet void fraction

Shell Full

For the shell full case the following is assumed:

i) The PRV opens such that the flow area is 10% of the full open flow area.
ii) PRV mass flow is calculated based on this.
iii) Mass flow is assumed 100% frozen liquid flow.

With this mass flow the tank pressure stays near the PRV set pressure and the liquid level will drop over time. In some cases this model has numerical stability problems and the open fraction must be adjusted up or down from the 10%.

Shell Empty Condition

When all of the liquid is gone the shell is liquid empty. In this case there is no longer any liquid to be vaporized, which means the liquid cooling effect is gone. We would expect the temperature to rise in the tank even if the pressure is maintained by the PRV. Also, as the vapour gets very hot in the near-empty tank, the PRV spring may begin to relax, causing the pop pressure to decrease. This is not modelled at this time.

As the tank approaches empty, we would expect an increase in the slope of the wall temperature curve. If the tank is near failure then this increase in the slope of the wall temperature curve could result in tank failure within a few minutes of the tank going liquid empty. This suggests that the thermal protection should never allow the tank to go empty within the 100 minutes for a pool fire, or 30 minutes for a torch fire. This in itself is a limit on the allowable insulation defect area.

On the other hand, some may argue that the test is over when the tank is liquid empty. At that point in time much of the hazard is gone – only the hazard from the compressed gas remains and this is small compared to when the tank is near full. However, the CGA standard is that the tank cannot fail for 100 minutes in a pool fire and 30 minutes in a torch fire regardless of fill.

Modelling of the vapour in the tank after it goes liquid empty is done as follows:

i) Redlich-Kwong equation of state
ii) Cp and Cv stored as a function of T in data file

The energy balance on the vapour in the tank over one time step dt is:

$$Q = m_2 u_2 - m_1 u_1 + \dot{m}_{prv} h_{prv} dt$$

This assumes the vapour is isothermal, which is probably not the case in reality.

Conservation of mass requires that:

$$m_2 = m_1 - \dot{m}_{prv} dt$$

If these are combined, it is possible to determine the temperature of the remaining vapour from:

$$T_2 = \frac{\dot{Q}dt - \dot{m}_{prv} C_p T_1 dt + m_1 C_v T_1}{C_p (m_1 - \dot{m}_{prv} dt)}$$

With the known T_2 and the known total tank volume it is possible to find the tank pressure from the Redlich-Kwong equation of state.

The problem with this model is that it cannot account for thermal stratification in the vapour when the PRV is closed and this may be very considerable. As a result it is not recommended that this model be used. This model has not been validated in any way.

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Appendix B: User Guide

Program Organization

The IDA 2.1 code runs in the Microsoft Windows environment (98SE, 2000, XP).

The code is written in the FORTRAN 90 language. The overall program consists of many subroutines. A brief summary follows:

main	- main program
	- this controls all I/O and calculations
thermal_2node	- 2-node thermal model of lading
mmix	- routine to estimate mixing between liquid core and liquid boundary
PRVflow	- PRV mass flow calculator
vonmises	- Von Mises stress analysis
vapconvection	- convection in vapour space
wallt2d	- 2D wall temperature model
tankgeo	- tank geometry calculator for hemi and elliptical heads
shape	- overall tank shape in 3D
cylindershape	- determines position in x,y,z for a given zone on the tank wall
defectposition	- locates defects on tank surface in 3D space
zones	- zones for remote fire analysis (not used at present)
aftac	- AFFTAC failure model
properties	- reads in propane saturation properties from data file
tables	- propane thermal property table lookup
failuretime	- stress rupture failure prediction
areasofzones	- calculates areas of wall zones
isentropic	- isentropic pressure drop
quality	- determines quality of 2-phase mixture entering PRV
findtsat	- finds saturation temperature (T sat) from known Psat
findtfromu	- finds T for liquid when u is known
findtfromuandm	- finds T when u and m are known for a 2-phase mixture
warmbalance	- energy balance solution for warm layer and vapour space
heatinputs	- sums up heat inputs to wall zones and partitions them between the vapour space, cool core and warm layer
wallwithinsul2d	- 2D wall temperature calculator for wall with thermal protection
fwalltoliqhoriz	- radiation view factors for vapour space (horizontal tank) to liquid surface
fwalltoliq	- as above for vertical tank
vertcylandwallview	- view factors for vertical tank to burning remote wall (not used at present)
fillvertcyl	- fill level calculator for vertical cylinder
fillhorizcyl	- fill level calculator for horizontal tank

Data Files

The program reads in data files and outputs data files as follows:

Input Files

IDA_2_rawdata.dat
IDA_2_defectinputdata.dat
IDA_2_fireposition.dat

These files define the tank overall size and head shape, wall thickness, PRV size and settings, fire conditions, initial tank conditions, and defect locations. The user must input this data using a text editor. Further details of these files can be found in Appendix A.

Other Input Files

propane_si.dat
A285grdB.dat
TC128_550.dat
TC128_620.dat
SA455.dat
Etc.

These files have the propane thermal data and the tank wall material properties. These files are already generated. Further details of these files can be found in Appendix A.

Intermediate Files (generated by program as inputs)

IDA_2_input.dat
IDA_2_xyz.dat
IDA_2_defectplot.dat
IDA_2_fireplot.dat

The program reads in the user input and processes it. The output from this processing is saved in the above files for later use. These files contain the 3D data for the tank shape and defect and fire locations. These can be viewed as 3D plots in TECHPLOT 9.0 to confirm input.

Output Files

IDA_2_heatinputs.out	- thermal data output for plotting
IDA_2_summary.out	- general data summary for plotting
IDA_2_vonMisesattime.dat	- Von Mises stress vs. time (for TECHPLOT animation, not working)
IDA_2_stressrupture.dat	- stress rupture damage vs. time for each node on tank surface (for TECHPLOT animation)
IDA_2_FOSattime.dat	- FOS vs. time (for TECHPLOT animation)
IDA_2_walltattime.dat	- wall T vs. time (for TECHPLOT animation)
IDA_2_tankviewfac.dat	- tank view factor for remote fire (not used)
IDA_2_summary.out	- text summary of simulation (for plotting)

These files contain the output for the simulation. Some of these files are input files for the generation of 3D plots and animations within the Techplot 9.0 graphics code. Other files are tabulated data for plotting in programs like MS Excel.

Running IDA 2.1

The code runs in a DOS window with simple user prompts for inputs from the user. The following figures show the user window.

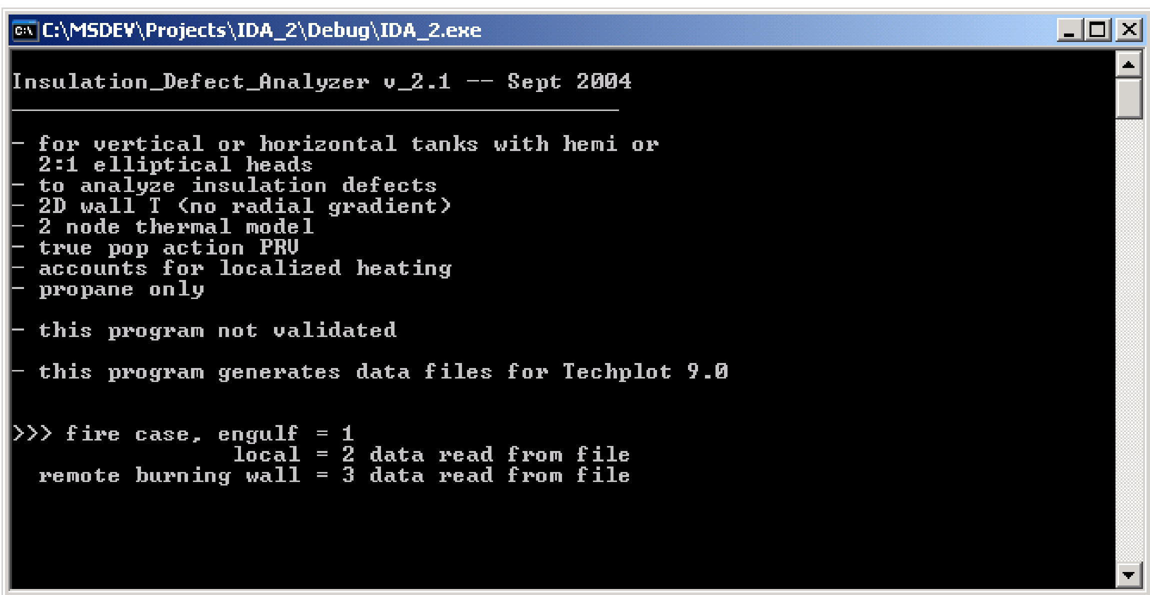

Figure B-1: IDA 2.1 in start-up in DOS Window

The user is asked about basic tank conditions including:

i) fire case
ii) defect case
iii) real-time duration

From this input the code reads in the appropriate input files for tank condition, defect locations, fire case. The following window is then displayed as the code runs.

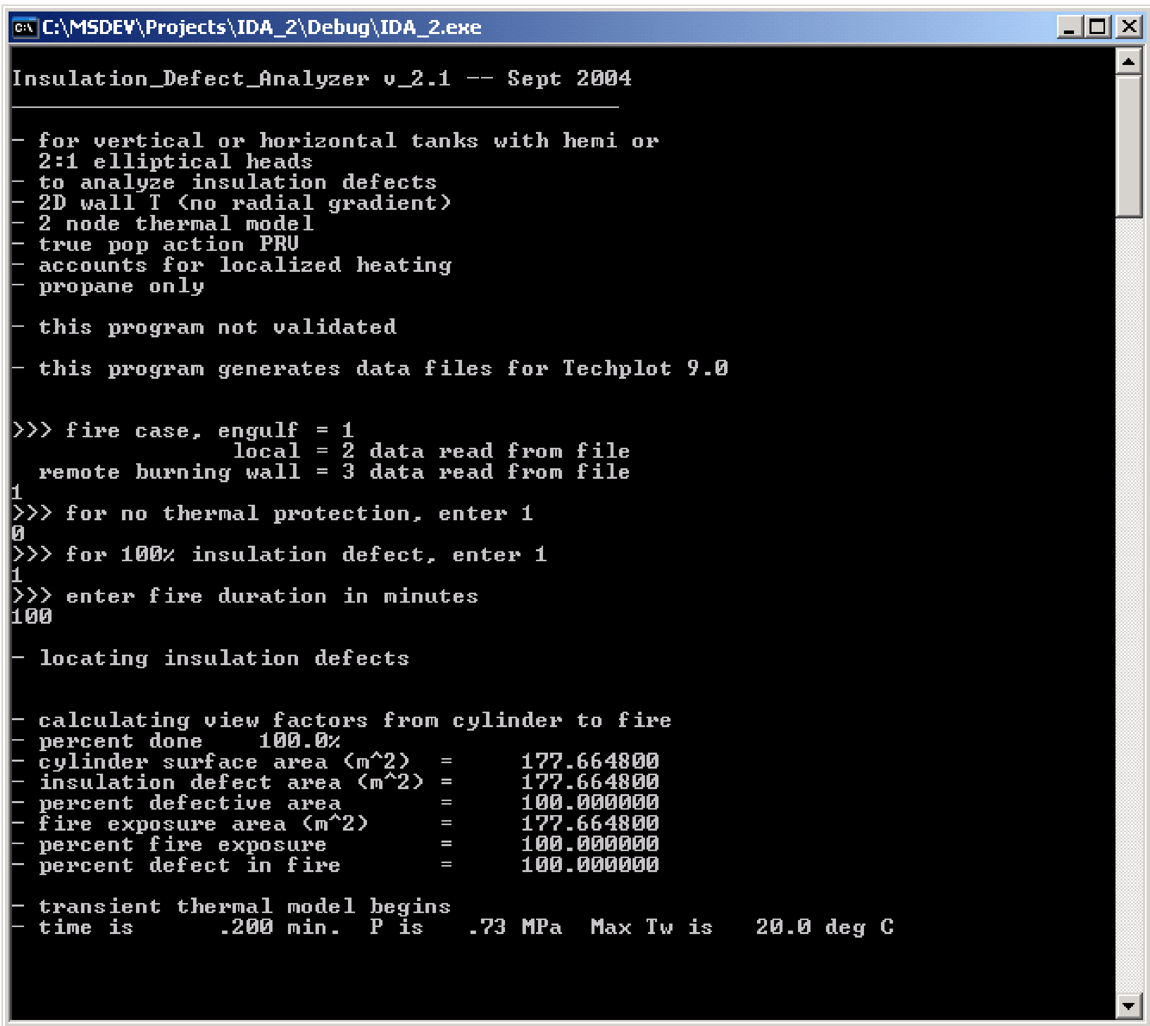

Figure B-2: IDA 2.1 Running in DOS Window

The code will run until the end of the run duration or until the tank empties. The following data is the main summary output file IDA_2_summary.out.

This file gives the following data:

i) tank 3D shape
ii) summary of main inputs
iii) summary of tank condition with time

This data can then be plotted using appropriate software. Other output files can be used to plot various program results.

IDA_2_summary.out

```
      3.000000      18.200000      17.689000      16.000000
      1.000000
     20.000000    9.400000E-01
  35000.000000      1.930000       1.100000       1.000000
  8.000000E-01    6.000000E-01
     10.000000      6.000000     816.000000
      2.000000      5.000000
   7800.000000    460.000000      45.000000    9.000000E-01
  2.950000E-01    100.000000     840.000000      13.000000
   1000.000000   9000.000000
      3.000000    9.000000E-01
    200.000000     72.000000
      .000      .000    -90.000
      .030      .239    -82.264
      .061      .327    -77.835
      .091      .386    -73.125
      .121      .429    -67.389
      .152      .460    -59.700
      .182      .481    -48.508
      .212      .494    -31.368
      .243      .500     -6.695
      .273      .500       .000
      .303      .500       .000
      .334      .500       .000
      .364      .500       .000
      .394      .500       .000
      .425      .500       .000
      .455      .500       .000
      .485      .500       .000
      .516      .500       .000
      .546      .500       .000
      .576      .500       .000
      .607      .500       .000
      .637      .500       .000
      .667      .500       .000
      .698      .500       .000
      .728      .500       .000
      .758      .500       .000
      .789      .500       .000
      .819      .500       .000
      .849      .500       .000
      .880      .500       .000
      .910      .500       .000
      .940      .500       .000
      .971      .500       .000
     1.001      .500       .000
     1.031      .500       .000
     1.062      .500       .000
     1.092      .500       .000
     1.122      .500       .000
     1.153      .500       .000
     1.183      .500       .000
     1.213      .500       .000
     1.244      .500       .000
     1.274      .500       .000
     1.304      .500       .000
     1.335      .500       .000
     1.365      .500       .000
     1.395      .500       .000
     1.426      .500       .000
     1.456      .500       .000
     1.486      .500       .000
     1.517      .500       .000
     1.547      .500       .000
     1.577      .500       .000
     1.608      .500       .000
```

1.638	.500	.000
1.668	.500	.000
1.699	.500	.000
1.729	.500	.000
1.759	.500	.000
1.790	.500	.000
1.820	.500	.000
1.850	.500	.000
1.881	.500	.000
1.911	.500	.000
1.941	.500	.000
1.972	.500	.000
2.002	.500	.000
2.032	.500	.000
2.063	.500	.000
2.093	.500	.000
2.123	.500	.000
2.154	.500	.000
2.184	.500	.000
2.214	.500	.000
2.245	.500	.000
2.275	.500	.000
2.305	.500	.000
2.336	.500	.000
2.366	.500	.000
2.396	.500	.000
2.427	.500	.000
2.457	.500	.000
2.487	.500	.000
2.518	.500	.000
2.548	.500	.000
2.578	.500	.000
2.609	.500	.000
2.639	.500	.000
2.669	.500	.000
2.700	.500	.000
2.730	.500	.000
2.760	.500	.000
2.791	.500	.000
2.821	.500	.000
2.851	.500	.000
2.882	.500	.000
2.912	.500	.000
2.942	.500	.000
2.973	.500	.000
3.003	.500	.000
3.033	.500	.000
3.064	.500	.000
3.094	.500	.000
3.124	.500	.000
3.155	.500	.000
3.185	.500	.000
3.215	.500	.000
3.246	.500	.000
3.276	.500	.000
3.306	.500	.000
3.337	.500	.000
3.367	.500	.000
3.397	.500	.000
3.428	.500	.000
3.458	.500	.000
3.488	.500	.000
3.519	.500	.000
3.549	.500	.000
3.579	.500	.000
3.610	.500	.000
3.640	.500	.000
3.670	.500	.000
3.701	.500	.000
3.731	.500	.000

3.761	.500	.000
3.792	.500	.000
3.822	.500	.000
3.852	.500	.000
3.883	.500	.000
3.913	.500	.000
3.943	.500	.000
3.974	.500	.000
4.004	.500	.000
4.034	.500	.000
4.065	.500	.000
4.095	.500	.000
4.125	.500	.000
4.156	.500	.000
4.186	.500	.000
4.216	.500	.000
4.247	.500	.000
4.277	.500	.000
4.307	.500	.000
4.338	.500	.000
4.368	.500	.000
4.398	.500	.000
4.429	.500	.000
4.459	.500	.000
4.489	.500	.000
4.520	.500	.000
4.550	.500	.000
4.580	.500	.000
4.611	.500	.000
4.641	.500	.000
4.671	.500	.000
4.702	.500	.000
4.732	.500	.000
4.762	.500	.000
4.793	.500	.000
4.823	.500	.000
4.853	.500	.000
4.884	.500	.000
4.914	.500	.000
4.944	.500	.000
4.975	.500	.000
5.005	.500	.000
5.035	.500	.000
5.066	.500	.000
5.096	.500	.000
5.126	.500	.000
5.157	.500	.000
5.187	.500	.000
5.217	.500	.000
5.248	.500	.000
5.278	.500	.000
5.308	.500	.000
5.339	.500	.000
5.369	.500	.000
5.399	.500	.000
5.430	.500	.000
5.460	.500	.000
5.490	.500	.000
5.521	.500	.000
5.551	.500	.000
5.581	.500	.000
5.612	.500	.000
5.642	.500	.000
5.672	.500	.000
5.703	.500	.000
5.733	.500	.000
5.763	.500	.000
5.794	.500	.000
5.824	.500	6.695
5.854	.494	31.368

5.885	.481	48.508
5.915	.460	59.700
5.945	.429	67.389
5.976	.386	73.125
6.006	.327	77.835
6.036	.239	82.264
6.067	.000	90.000

+++

I N S U L A T I O N   D E F E C T   A N A L Y Z E R

IDA_v2.1 Sept 2004

vertical and horizontal cylinders with hemi
or 2:1 elliptical heads

- 2 node thermal model with localized heating
- pop action PRV
- simplified internal convection and radiation
- 2D wall conduction (no gradient with R)
- 1D jacket conduction
- ignore thermal capacity of insulation
- no degradation of insulation
- propane only
- simple hoop stress (no thermal distortion accounted for)

this version outputs files for plotting
by TECHPLOT 9.0 by AMTEC Engineering Inc.

by A.M.Birk Sept 2004
A.M. Birk Engineering

for Transport Canada

>>>> this program is not fully validated <<<<<

+++

tank diameter (m) = 3.000000
tank length (m) = 17.689000
wall thickness (mm) = 16.000000
wall material type = 1.000000
percent defective insulation = 100.000000
percent fire exposure = 100.000000
percent of defect under fire = 100.000000

tank CL height (m) = 0.000000E+00
tank CL from side wall (m) = 6.000000
tank CL from wall end (m) = 15.000000
end wall position from tank end (m) = 0.000000E+00

tank volume (m^3) = 125.036100
init tank fill = 9.400000E-01
init liquid mass (kg) = 58796.350000
init vapour mass (kg) = 135.442600
init tank P (kPa) = 834.400000
init lading T (deg C) = 20.000000
PRV setting (MPa) = 1.930000
PRV area (m^2) = 4.740549E-03
PRV open (MPa) = 2.123000
PRV reclose (MPa) = 1.930000

time step (sec) = 1.000000
axial steps = 200.000000
angular steps = 72.000000

end wall h (m) = 0.000000E+00
end wall w (m) = 0.000000E+00

side wall h (m) = 30.000000
side wall w (m) = 18.000000

ground ls (m) = 0.000000E+00
ground le (m) = 0.000000E+00

end wall temperature (deg C) = 20.000000
side wall temperature (deg C) = 816.000000
ground temperature (deg C) = 20.000000
sky temperature (deg C) = 20.000000

tank surface emissivity = 9.000000E-01

zone 1 = tank surface
2 = end wall
3 = side wall
4 = ground
5 = sky

---

results from thermal model

---

time	press	T fs	T core	fill	PRVmdot	m fs	m core	mg	Tw peak	Tj peak	Tw av	Twavvap	Damage	FOS 2	H con	F liq a
.00	.733	20.00	20.00	.94	.00	5879.64	52916.7	135.4	20.0	26.6	20.0	20.0	.00	9.24	6.33	.93
.17	.733	20.00	20.00	.94	.00	5879.65	52916.7	135.4	20.0	92.5	21.0	20.0	.00	9.24	3.48	.93
.33	.733	20.00	20.00	.94	.00	5879.73	52917.0	135.4	20.2	157.3	21.2	20.1	.00	9.24	3.54	.93
.50	.733	20.00	20.00	.94	.00	5879.94	52918.0	135.4	20.5	220.2	21.4	20.4	.00	9.24	3.67	.93
.67	.733	20.00	20.00	.94	.00	5880.36	52920.4	135.3	21.0	280.6	21.9	20.9	.00	9.24	3.88	.93
.83	.733	20.01	20.00	.94	.00	5880.42	52924.9	135.1	21.9	337.3	22.5	21.7	.00	9.24	4.13	.93
1.00	.735	20.10	20.00	.94	.00	5879.45	52927.8	135.3	23.1	389.6	23.5	23.0	.00	9.22	4.38	.93
1.17	.738	20.23	20.00	.94	.00	5877.62	52933.1	135.5	24.8	436.7	24.8	24.6	.00	9.18	4.66	.93
1.33	.743	20.44	20.01	.94	.00	5874.60	52939.2	136.0	27.1	478.2	26.4	26.8	.00	9.12	4.93	.93

Sample Input Files

The following files are required to run IDA 2. The contain the basic tank , defect and fire exposure data.

IDA_2_rawdata.dat

```
3.0
0.016 1
16.7
200 72
2 0
7800.0 460.0 45.0 0.90
0.295 100.0  840.0 13.0 1000.0 9000.
3.0 0.90
20. 0.94
29000. 1.93 1.1 1.0 0.80 0.60
10. 6.  816.
2.  5.
```

OD (m)
Wallthick (m), wallmat
cyllength (m)
zsteps, angles
headtype, vertcylinder
wall rho, c, k, emm
kinsl, rhoinsl, cinsul, winsulmm, tdegrade, timedegrade
jackthickmm, emmjack
tinit, percentfill
scfmair, prvsetMPa, factoropen, factorclose, cdg, cdf
hsideovd, wsideovd, tfirec
aovd, covd

IDA_2_defectinputdata.dat

```
3
0.0 0.5 -0.50 0.50 1.0 2.50
0.0 0.5 -0.50 0.50 1.0 2.50
0.0 0.5 -0.50 0.50 1.0 2.50
```

#defects
xovd1,xod2,yod1,yod2,zod1,zod2
...
...
x = side to side
y = centre to top

z = length

right side is xovd from x1 = 0 to x2 = 0.5
left side is xovd from x1 = -0.5 to x2 = 0

IDA_2_fireexposure.dat

```
1
850. 1.00  0.0   0.5  -0.5  0.50  1.0  2.6


#fire zones
tfiredegC, Fij, xovd1,xod2,yod1,yod2,zod1,zod2
...
...
```

x = side to side
y = centre to top
z = length

right side is xovd from x1 = 0 to x2 = 0.5
left side is xovd from x1 = -0.5 to x2 = 0