TP 13518E

# Tank-Car Insulation Defect Assessment Criteria: Thermal Analysis of Defects

for

Transport Canada
Transportation Development Centre

by

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Kingston, Ontario, Canada

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The Report uses the term "defect" when addressing any level of thermal protection and insulation deficiency. The term "defect" as used in this Report is not intended to define a level of non compliance with Regulations or Standards but rather a deteriorated or degraded condition which is not yet established as constituting an unacceptable or out of Standard situation.

Un sommaire français se trouve avant la table des matières.

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16. Abstract


An inspection technique was previously developed to find thermal insulation deficiencies on rail tank-cars. This method uses a thermal imager to locate insulation gaps under the tank steel jacket, relying on a small temperature difference between the tank lading and the surroundings. Limited field tests suggest that some tanks may have insulation deficiencies, which raises the question as to whether these deficiencies are important from a safety standpoint.

This report contains an analysis using rail tank-car thermal computer models to show how insulation defects affect the tank thermal response to the environment and fire impingement. The objective of this work was to develop criteria for assessing insulation defects on tank-cars with thermal protection or with thermal insulation. These assessment criteria will be used by inspectors to decide when a tank will need insulation repairs. The analysis of thermally protected tanks was conducted using an approach similar to the U.S. Federal Railroad Administration (FRA) computer code Analysis of Fire Effects on Tank Cars (AFFTAC 3.0). All analysis used CAN/CGSB 43.147-97 as the applicable standard.


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16. Résumé

Une technique d'inspection mise au point antérieurement permet de repérer les défauts d'isolation thermique des wagons-citernes. Elle fait appel à un imageur thermique qui situe ces défauts sous la jaquette en acier de la citerne à partir des légères différences de température détectées entre le contenu de la citerne et le milieu ambiant. Des essais limités sur le terrain indiquent que certaines citernes peuvent effectivement comporter des défauts d'isolation, mais il reste à déterminer s'ils compromettent sérieusement la sécurité.

Ce rapport présente une simulation informatique du comportement au feu et de la réaction à la température ambiante de wagons-citernes comportant des défauts d'isolation. Cette simulation a pour objet de développer des critères d'évaluation des défauts d'isolation des wagons-citernes à protection thermique et des wagons-citernes isothermes. Les inspecteurs se serviront de ces critères pour décider si la protection ou l'isolation thermique d'un wagon a besoin d'être réparée. L'analyse des wagons-citernes à protection thermique a été réalisée selon une méthode proche de celle appliquée par le modèle informatique AFFTAC 3.0 (Analysis of Fire Effects on Tank Cars) de la U.S. Federal Railroad Administration (FRA). Toutes les évaluations ont été faites par rapport à la norme CAN/CGSB 43.147-97.

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## Executive Summary

Certain rail tank-cars require thermal insulation to protect them from accidental fire impingement (these systems are referred to as *thermal protection systems*), while others have insulation to keep the tank commodity thermally insulated from the surroundings (these systems are referred to as *thermal insulation systems*). In both cases, it is possible to have insulation defects that can reduce the effectiveness of the system.

A new inspection technique was developed by A.M. Birk Engineering to find thermal insulation deficiencies on rail tank-cars. This method uses a thermal imager to find insulation gaps under the tank steel jacket. The method relies on a small temperature difference existing between the tank lading and the surroundings. Solar heating can also assist in generating thermal gradients that the thermal imager can identify. Further details of this system can be found in the 1998 Transportation Development Centre report entitled *Thermographic Inspection of Tank-Car Thermal Insulation*, TP 13203E.

Limited field tests suggest that some tanks have insulation deficiencies. A few older tanks have been inspected and were found to be over 50 percent defective. However, are these deficiencies important from a safety standpoint? The objective of this work was to develop criteria for assessing insulation defects on tank-cars with thermal protection or thermal insulation. These criteria will be used by inspectors to decide when a tank will need insulation repairs.

This report contains details on an analysis using rail tank-car thermal computer models to show how insulation defects affect the tank thermal response to the environment and fire impingement. The analysis presented in this report was conducted to quantify what level of defect is acceptable. This analysis was based on the CAN/CGSB 43.147-97 standard that specifies insulation system performance.

***Thermally Protected Tanks***

A computer model was developed to represent a thermally protected tank-car with insulation defects exposed to engulfing and torch type fires. This model, called Tank Model with Insulation Defects (TMID), is based for the most part on assumptions similar to those of the U.S. Federal Railroad Administration (FRA) Analysis of Fire Effects on Tank Cars (AFFTAC) computer code.

The TMID model is two-dimensional in that it represents the tank as a long cylinder. Insulation defects are modelled as regions where no insulation exists between the steel jacket and the tank wall. Defects are assumed to run the entire tank length on one side of the tank. Defect position is specified by giving the angle from the tank top to the defect top and bottom.

The baseline case considered the following type of tank and assumptions:

112J340W propane tank
tank diameter: 3 m, length: 17.3 m
wall thickness: 17.5 mm
insulation conductance: 22.7 $W/m^2K$
insulation thickness: 13 mm
initial fill: 90 percent
initial temperature: 15.6°C
PRV assumed to hold tank pressure at maximum of 1.03 x start to discharge pressure
PRV start to discharge pressure: 1.93 MPa) (280.5 psig)
PRV capacity assumed large enough to maintain assumed pressure
tank material: TC 128 steel

Based on the above tank, the AFFTAC model showed that:

current thermal protection systems are more than adequate to protect a tank for 100 minutes from an engulfing fire with an effective radiating temperature of 816°C.

current thermal protection systems are more than adequate to protect a tank for 30 minutes from a torching fire with an effective radiating temperature of 1204°C, and an effective torch emissivity of 0.536.

the simulations suggest that the pool fire test is the determining test. However, this conclusion changes if the torch properties are adjusted to more accurately represent a credible torch fire accident.

Both the AFFTAC and TMID models showed that large defects can bring a tank to failure under engulfing fire conditions as defined by the CGSB standard within 100 minutes.

Based on the TMID model:

it was concluded that a factor of safety (FOS) should be applied in this analysis. In the case of thermal protection, the FOS is defined in this report as (tank burst pressure at 100 minutes in fire)/(tank pressure at 100 minutes in fire). The current insulation systems with no defects have FOS > 3.0. Based on uncertainties of PRV performance, fire heating, and high temperature steel properties, it was suggested that an FOS = 1.6 would be reasonable.

when an FOS of FOS = 1.6 is used, a 112 tank with propane exposed to fire for 100 minutes can have between 4 to 24 percent defective insulation provided a steel jacket is present and is not in direct contact with the tank primary wall. The 24 percent end of the range assumes the remainder of the tank is protected with perfect insulation with properties as quoted for Kaowool ceramic blanket with

72 kg/m$^3$ density. The 4 percent end of the range assumes that the remainder of the tank is protected with insulation with a conductance of 22.7 W/m$^2$K, which represents the maximum conductance that would pass the FRA plate test standard (i.e. keep steel plate sample temperature below 427°C for 100 minutes when exposed to a pool fire).

the allowable defect area was strongly affected by input variables including:

assumed FOS
wall thickness
PRV setting
tank fill
fire temperature
commodity
tank surface emissivity

detailed analysis of defects suggested that:

as defect gets small the wall stress and temperature may be reduced
several small well separated defects are safer than one large defect

defects in the vapour space are more significant than those in the liquid space. However, the possibility of tank rollover makes this difficult to apply in the field.

The above conclusions have come from computer simulation results and, therefore, are not fully validated. It is recommended that some testing be conducted to confirm these results.

***Thermally Insulated Tanks***

A computer model was developed to calculate the effects of insulation defects on the overall conductance of thermally insulated tanks. For insulated tanks, the following conclusions have been made.

Current non-defective thermal insulation systems with no insulation discontinuities just meet the required conductance, as specified in CAN/CGSB 43.147-97. Analysis suggests that less that 2 percent of the tank surface could have insulation defects and still meet the requirement. However, if existing insulation discontinuities are accounted for, no defects would be allowed.

Some commodities, such as chlorine, cannot tolerate high temperatures (i.e. chlorine will cause steel to ignite at temperatures above 215°C). Therefore, some commodities cannot tolerate any insulation defects.

Further analysis is necessary to properly quantify allowable insulation defects.

## Sommaire

Certains wagons-citernes doivent comporter une isolation thermique pour les protéger contre l'effet des flammes en cas d'accident (il s'agit de wagons-citernes *à protection thermique*) alors que d'autres ont une isolation thermique destinée à soustraire leur contenu à la température ambiante (wagons-citernes dits *isothermes*). Dans les deux cas, des défauts d'isolation peuvent apparaître avec pour effet une protection réduite du wagon ou de son contenu.

La société A.M. Birk Engineering a mis au point une nouvelle technique d'inspection pour déceler les défauts d'isolation des wagons-citernes. Elle fait appel à un imageur thermique qui repère sous la jaquette en acier des citernes les endroits où l'isolant est détérioré ou totalement absent. Cet appareil repose sur le principe de la détection des légères différences de température entre le contenu de la citerne et le milieu ambiant. Il a été établi également que les conditions ensoleillées sont de nature à engendrer des gradients thermiques détectables par l'imageur thermique. On trouvera plus de détails sur cet appareil dans le rapport TP 13203E intitulé *Thermographic Inspection of Tank-Car Thermal Insulation*, fait pour le Centre de développement des transports en 1998.

Des essais limités sur le terrain indiquent que certaines citernes peuvent effectivement comporter des défauts d'isolation. À l'inspection de quelques wagons-citernes anciens, il a été établi que leur isolation thermique était à plus de 50 p. cent défectueuse. Toutefois, on ne sait pas si cette situation compromet sérieusement la sécurité. La présente recherche avait pour objet de développer des critères d'évaluation des défauts d'isolation des wagons-citernes à protection thermique et des wagons-citernes isothermes. Les inspecteurs se serviront de ces critères pour décider si la protection ou l'isolation thermique d'un wagon a besoin d'être réparée.

Ce rapport présente une simulation informatique du comportement au feu et de la réaction à la température ambiante de wagons-citernes comportant des défauts d'isolation. Cette simulation avait pour objet de déterminer le degré de détérioration acceptable de la protection ou de l'isolation thermiques. Toutes les évaluations ont été faites par rapport aux exigences de performance d'isolation de la norme CAN/CGSB 43.147-97.

### *Wagons-citernes à protection thermique*

Un modèle informatique a été élaboré pour simuler le comportement au feu d'un wagon-citerne présentant des défauts d'isolation thermique et enveloppé de flammes ou soumis à un jet de gaz enflammé. Ce modèle, baptisé Tank Model with Insulation Defects (TMID), applique pour la plus grande part des hypothèses proches de celles retenues pour le modèle informatique AFFTAC 3.0 (Analysis of Fire Effects on Tank Cars) de la U.S. Federal Railroad Administration (FRA).

Le modèle TMID est du type bidimensionnel du fait qu'il représente le wagon comme un long cylindre. Les défauts d'isolation pris en compte dans la simulation prennent la forme de zones où il ne subsiste aucun isolant entre la jaquette en acier de la citerne et la citerne elle-même. Il est posé que ce défaut s'étend d'un côté sur toute la longueur de la citerne. Sa position verticale est définie par l'angle formé par deux droites reliant ses limites supérieure et inférieure au sommet de la citerne.

La modélisation de référence avait pour objet le wagon ci-dessous et posait les hypothèses suivantes :

wagon-citerne à propane 112J340W
diamètre citerne : 3 m; longueur : 17,3 m
épaisseur de paroi : 17,5 mm
conductance thermique de l'isolant : 22,7 $W/m^2K$
épaisseur de l'isolant : 13 mm
remplissage initial de la citerne : 90 p. cent
température initiale : 15,6 °C
la soupape de sûreté est présumée maintenir dans la citerne une pression maximale de 1,03 fois sa pression nominale d'ouverture
pression d'ouverture de la soupape de sûreté : 1,93 MPa (280,5 $lb/po^2$)
la soupape de sûreté est présumée avoir une capacité d'écoulement suffisante pour maintenir la pression à l'intérieur de la citerne à la valeur présumée
citerne en acier TC 128

Selon les simulations AFFTAC appliquées au cas de référence ci-dessus :

les protections thermiques actuelles suffisent largement pour protéger pendant 100 minutes une citerne des effets d'un bain de flammes ayant une température effective de rayonnement de 816 °C.

les protections thermiques actuelles suffisent largement pour protéger pendant 30 minutes une citerne des effets d'un jet de gaz enflammé dont la température effective de rayonnement atteint 1 204 °C et l'émissivité 0,536.

l'essai déterminant est celui en bain de flammes enveloppantes. Par contre, lorsque les propriétés du jet de gaz enflammé sont modifiées pour représenter plus fidèlement les conditions d'un accident provoquant un jet de gaz enflammé, cette conclusion change.

Les deux modèles AFFTAC et TMID montrent que des défauts d'isolation étendus peuvent entraîner en moins de 100 minutes la rupture d'une citerne plongée dans un bain de flammes ayant les caractéristiques définies dans la norme CAN/CGSB 43.147-97.

Selon la simulation TMID :

il faut appliquer un facteur de sécurité (FS) dans cette analyse. Dans le cas d'une citerne à protection thermique, ce facteur est défini comme suit dans le rapport : (pression de rupture après 100 minutes dans les flammes)/(pression interne de la citerne après 100 minutes dans les flammes). Les systèmes isolants actuels exempts de défaut ont un facteur de sécurité supérieur à 3. Compte tenu des incertitudes concernant la performance des soupapes de sûreté, les températures en jeu et les propriétés des aciers haute température, un facteur de sécurité de 1,6 semblerait raisonnable.

avec un facteur de sécurité de 1,6, une citerne à propane de type 112 exposée aux flammes pendant 100 minutes peut présenter un défaut d'isolation couvrant entre 4 et 24 p. cent de la surface totale de la citerne dans la mesure où celle-ci est entourée d'une jaquette en acier sans contact direct aucun avec sa paroi. La valeur admissible supérieure (24 p. cent) suppose que le reste de l'isolant est en parfait état et assure la protection thermique d'une doublure céramique Kaowool (masse volumique de 72 $kg/m^3$). La valeur admissible inférieure (4 p. cent) vaut pour les cas où l'isolant de protection a une conductance de 22,7 $W/m^2K$, soit la conductance maximale permettant de réussir à l'essai standard de résistance de la FRA (maintenir la température de l'éprouvette sous 427 °C pendant 100 minutes d'exposition à un bain de flammes enveloppantes).

l'étendue admissible du défaut variait considérablement selon la valeur attribuée aux variables de la modélisation :

FS présumé
épaisseur de paroi
pression d'ouverture de la soupape de sûreté
niveau de remplissage de la citerne
température des flammes
produit transporté
émissivité pariétale de la citerne

l'analyse détaillée des défauts laisse penser que :

moins le défaut est étendu moins la paroi de la citerne risque d'être sollicitée et réchauffée
plusieurs petits défauts bien séparés présentent moins de risques qu'un seul défaut de grande superficie

les défauts d'isolation au-dessus de la marge de remplissage ont plus de conséquences que ceux donnant au-dessous de cette marge. Cette conclusion ne vaut cependant plus dès qu'il y a eu renversement du wagon-citerne.

Les conclusions ci-dessus sont fondées sur les seuls résultats de simulations informatiques; elles sont donc sujettes à caution. Aussi, est-il recommandé de mener certains essais de validation.

***Réservoirs isothermes***

Un modèle informatique a été élaboré pour calculer les effets des défauts d'isolation sur la conductance globale des citernes isothermes. Voici les conclusions tirées de la simulation :

Les systèmes d'isolation actuels en parfait état qui couvrent intégralement la surface intérieure de la citerne offrent tout juste l'isolation prescrite, en termes de conductance, par la norme CAN/CGSB 43.147-97. L'analyse suggère qu'une citerne ayant un défaut d'isolation sur moins de 2 p. cent de sa surface totale offrira encore l'isolation thermique prescrite. Par contre, avec une isolation non intégrale aucun défaut d'isolation ne serait admissible.

Certains produits tels que le chlore n'admettent aucune élévation marquée de la température. Le chlore, en l'occurrence, rend l'acier combustible dès que la température dépasse 215 °C. Les citernes servant à transporter de tels produits ne peuvent présenter aucun défaut d'isolation.

D'autres analyses sont nécessaires pour bien quantifier l'étendue admissible des défauts d'isolation.

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## Figures

# 1 Introduction

## *1.1 Background*

An inspection technique was developed by A. M. Birk Engineering to find thermal insulation deficiencies on rail tank-cars. This method uses a thermal imager to find insulation gaps under the tank steel jacket. The method relies on a small temperature difference existing between the tank lading and the surroundings. Also, solar heating can assist in the generation of thermal gradients that the thermal imager can identify. Further details of this system can be found in the Transportation Development Centre report "Thermographic Inspection of Tank-Car Thermal Insulation", TP 13203E, dated March 1998.

Limited field tests suggest that significant numbers of tanks may have insulation deficiencies. Now the question must be asked – are these deficiencies important from a safety standpoint?

This report contains an analysis using rail tank-car thermal computer models to show how insulation defects affect the tank thermal response to the environment and fire impingement.

## *1.2 Objectives and Scope*

The objective of this work was to develop criteria for assessing insulation defects on tank-cars with thermal protection or with thermal insulation. These assessment criteria will be used by inspectors to decide when a tank will need insulation repairs.

As requested by Transport Canada, the analysis of thermally protected tanks was to be conducted using an approach similar to the US Federal Railroad Administration (FRA) computer code AFFTAC 3.0.

All analysis was to use CAN/CGSB 43.147-97 as the applicable standard.

# 2 Tank Insulation Systems

The system considered in this report is shown in Figure 2-1. It consists of a horizontal cylindrical pressure vessel covered with a layer of thermal insulation with a steel jacket cover over the insulation. In this report, two different tank-car insulation systems will be considered – thermal protection and thermal insulation.

## 2.1 Thermal Protection Systems

Thermal protection systems are designed to protect tanks from accidental fire exposure. These systems were introduced in the late 1970s and early 1980s as a result of an extensive research program to identify ways of reducing tank-car accidents.

### 2.1.1 Requirements for Pressure Cars with Thermal Protection

The requirement for thermal protection systems for pressure tank-cars is specified in CGSB 79.18 (see Appendix A, Appendix B). Basically the requirement is:

*thermal protection on a tank car, shall have sufficient thermal resistance so that there will be no release of any lading from within the tank car, except release through the pressure relief device, when subjected to:*

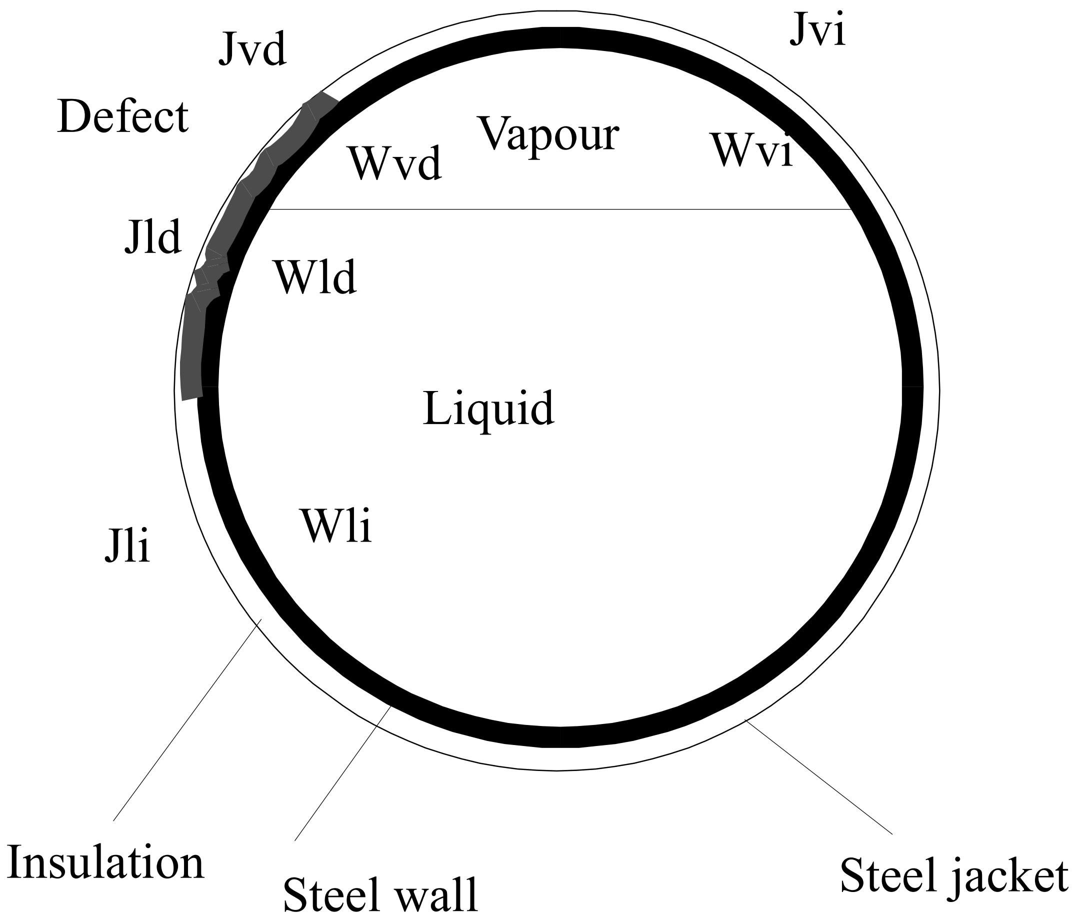
*(1) A pool fire for 100 min, and*
*(2) A torch fire for 30 min.*

This specification is a change from a previous standard that involved a fire test of a steel plate sample covered with insulation. However, the older standard pass/fail criteria was not plate failure but rather a measured temperature of 427$^oC$ within the time allowed (i.e. with pool fire plate temperature must stay under 427$^oC$ for 100 minutes, for torch fire plate temperature must stay under 427$^oC$ for 30 minutes). To pass the old requirement an insulating material had to have an overall conductance of less than 22.7 $W/m^2K$ (4 Btu/hr $ft^2$ $^oF$) (see Johnson (1998) ).

The DOT standard pool fire must have a temperature of 871$^oC$ plus or minus 56$^oC$. As per the CGSB plate test standard, the engulfing pool fire must heat a 16 mm steel plate to 427$^oC$ within 13 min (plus or minus 30 seconds). This time requirement is based on the full scale fire test of a tank-car (Townsend et al (1974) ).

### 2.1.2 Effect of Thermal Protection

Thermal protection reduces the rate of heat addition to the tank. This means the tank will heat up more slowly and it will take longer to empty through the pressure relief valve (PRV). If a thermally protected tank has insulation defects then the increase in heating rate will cause the tank to empty more rapidly through the PRV. Where there are gaps in the thermal insulation, vapour space wall temperatures will be higher than in places where there is intact thermal insulation.




Jvd - jacket vapour defect
Jvi - jacket vapour insulated
Jld - jacket liquid defect
Jli - jacket liquid insulated
Wvd - wall vapour defect
Wvi - wall vapour insulated
Wld - wall liquid defect
Wli - wall liquid insulated

**Figure 2-1: Sketch Showing Insulation System with Defect**

If a tank is protected just enough to pass the pool fire requirement, then theoretically any defect will make it fail. If a tank exceeds the requirement by say 10%, then defects would be acceptable, until this 10% margin is lost.

A 112 type tank-car with no thermal protection was exposed to an engulfing JP4 fuelled pool fire in the early 1970’s and it failed after about 24 minutes (see Townsend et al (1974). Had it not failed it would have emptied through its relief valve in about 40 minutes. A thermally protected tank must survive a pool fire for 100 minutes without failing (i.e. releasing its contents other than through the pressure relief valve).

A 112 type tank car with just a steel jacket (no thermal insulation in the gap) has some thermal protection from the steel jacket because the jacket behaves like a thermal radiation shield (see Birk (1983) ) and this reduces the heat flow by about half. With steel jacket alone a tank exposed to an engulfing pool fire will empty after about 85 minutes. If the fire is severe enough it may fail before it is empty. Once it is empty of liquid, the tank lading and wall temperatures may continue to rise and if the tank pressure is not reduced the tank could fail.

A 112 type tank car with 13 mm of high temperature thermal insulation (such as Kaowool – see Appendix F) covered with steel jacket will survive a pool fire for well over 100 minutes. Further details of this will be presented later in this report. The idea being studied here is that, if the tank exceeds the requirements of the pool fire test then it should be acceptable to have some insulation defects.

## *2.2 Thermal Insulation*

Thermal insulation is generally used for tanks that carry commodities that must be thermally insulated from the surroundings. Insulation may or may not be a tank specification requirement. This thermal insulation will keep the commodity warm or cool relative to the surroundings. Some tanks that require thermal insulation may also require thermal protection.

### 2.2.1 Requirements for Thermal Insulation of Pressure Tank-Cars

In general, if a pressure tank-car requires thermal insulation it must meet the following requirement:

*If insulation is a specification requirement, it shall be of sufficient thickness so that the thermal conductance at 15.5 C (60 F) is not more than 1.533 kJ/h $m^2$ C (0.075 Btu/h $ft^2$ F) temperature differential. If exterior heaters are attached to the tank, the thickness of the insulation over each heater element may be reduced to one-half that required for the shell.*

It should be noted that for some commodities (for example $CO_2$) the thermal conductance must be even lower.

For tanks that need both thermal insulation and thermal protection, it can be seen this requirement of a conductance of 0.075 far exceeds (by about a factor of 50, if we use 4 Btu/hr $ft^{2o}F$ as the limit to pass the thermal protection requirement) that needed for the thermal protection requirement. However, for the insulation to pass the thermal protection requirement it must be a high temperature insulation.

Common insulation systems include 5 cm of fibreglass insulation (low temperature) and 5 cm of ceramic blanket (high temperature insulation).

### *2.3 Insulation Discontinuities*

One issue that has not been discussed so far is insulation discontinuities that exist in a normal tank-car. These discontinuities are direct metal conduction paths that are present due to tank design features.

Discontinuities and associated U values were suggested by Johnson (1998) in the AFFTAC 3.0 manual. This data is shown in **Table 2-1**. It is not clear how these U values were obtained or whether they assume a certain convective film coefficient on the tank outer and inner surface. For now it has been assumed that these U values are based only on thermal conductivities of the tank wall materials.

As can be seen from the table the discontinuities are not very significant for a thermally protected tank with 13 mm of insulation (insulation discontinuities increase overall U by 1%). In this case discontinuities will be ignored.

However, discontinuities are significant for thermally insulated tank where the overall wall U coefficient is increased by 43%. This result is somewhat misleading because the 43% increase represents a small overall increase in heat transfer when compared to what a large insulation defect would produce.

Let us consider a 1 m x 1 m portion of the tank with steel jacket but all the insulation is removed. The U value for this area depends on the heat transfer conditions. Table 2-3 shows a summary of the U value associated with this 1 square meter of defect.

The U value for the wall area with insulation is calculated from the following expression:

Table 2-1: Summary of Insulation Discontinuity U values (from Johnson (1995) )

| **location** | **U (BTU/hr $^oF$)** | **U (W/K)** |
|---|---|---|
| | | |
| manway nozzle and cover | 14.6 | 2.4 |
| siphon and air vent nozzle | 6.1 | 1.0 |
| safety relief valve nozzle | 3.2 | 0.52 |
| jacket spacers | 15.5 | 2.5 |
| bottom outlet saddle | 13.1 | 2.1 |
| draft sills | 72.9 | 11.9 |
| body bolsters | 65 | 10.6 |
| brake cylinder support | 4.6 | 0.75 |
| brake rod lever support | 3.2 | 0.52 |
| | | |
| total U for discontinuities (sum of individual U's) | 198.2 | 32.3 |
| | | |
| total U for thermally insulated tank (105 car assuming conductance of 0.426 $W/m^2K$) | 456 | 74.3 |
| total U for thermally protected tank (112 car assuming conductance of 22.7 $W/m^2K$) | 24300 | 3960 |

1 Btu/hr $^oR$ = 0.163 W/K

$$U = \frac{1}{R} = \frac{1}{\frac{1}{h_{outside}} + \frac{w_{wall}}{k_{wall}} + \frac{w_{insul}}{k_{insul}} + \frac{1}{h_{inside}}}$$

where,

w = wall or insulation thickness
k = thermal conductivity (W/mK)
h = convective heat transfer coefficient ($W/m^2K$)

The resulting U value depends on the insulating properties of the insulation and also on the convective coefficients. On the tank inside surface it is reasonable to assume the h is large if the wall is wetted by liquid. However, on the outside the h value depends on the ambient conditions. Table 2-2 shows some typical convective heat transfer coefficients for a wind blowing over a cylinder. At low wind speeds the convective coefficient is quite low and this can reduce the overall U value for a tank significantly (i.e. the outside h become a resistance to heat flow).

**Table 2-2: Convective Film Coefficient for Wind Blowing over Tank-Car**

| Wind Speed (km/hr) | h ($W/m^2K$) |
|---|---|
| | |
| 3 | 3 |
| 6 | 6 |
| 16 | 12 |
| 32 | 21 |

The U value for the wall area with no insulation present must include thermal radiation in from the jacket to the wall where the insulation is gone. Table 2-3 shows how the tank U value is affected by the discontinuities and by defects.

As can be seen from the table, the importance of insulation defects and discontinuities depends on the specific heat transfer conditions. This is because the outside surface convective coefficient is itself a barrier to heat transfer. A change in the outside h will also change the overall U for the tank system. As the temperature of the exterior heat source increases thermal radiation takes over as the dominant mode of heat transfer. This change in temperature also changes the overall U value.

The results show that 1 $m^2$ of defect can be more important than all the discontinuities. The worst case defect is when the insulation is gone and the steel jacket is put in direct contact with the tank wall. One square metre of such a defect is more important than the tank discontinuities.

If an air space is left where there is no insulation then the U value depends on the temperature difference. As can be seen, a 1 m x 1 m defect becomes just as important as all the insulation discontinuities as soon as the heat source temperature exceeds the lading temperature by 600 degrees. However, at ambient temperatures (say 100$^o$C difference) it takes between 3 and 5 $m^2$ of defect to equal the effect of the insulation discontinuities

From this point on, insulation discontinuities will not be discussed.

**Table 2-3: Summary of U Value for 1 m x 1 m Insulation Defect (assuming h = 5 $W/m^2K$ in the annulus if no insulation is present)**

| | **Thermal Insulation** 102 mm nominal conductance 0.426 $W/m^2$ K | | **Thermal Protection** 13 mm insulation nominal conductance 22.7 $W/m^2$ K | |
|---|---|---|---|---|
| approximate tank U with no defects or discontinuities (outside h = 50 $W/m^2K$) | 74 W/K | | 2700 W/K | |
| approximate tank U with no defects or discontinuities (outside h = 5 $W/m^2K$) | 69 | | 715 W/K | |
| increase in U of tank due to discontinuities (from Johnston (1998) ) h outside not specified | 32.3 | | 32.3 | |
| | | | | |
| increase in U for 1 $m^2$ steel jacket in direct contact with tank shell (h outside = 50 $W/m^2K$) | 48 W/K | | 48 W/K | |
| increase in U for 1 $m^2$ steel jacket in direct contact with tank shell (h outside = 5 $W/m^2K$) | 5.0 | | 5.0 | |
| | | | | |
| increase in tank U for 1 $m^2$ defect (no insulation) including thermal radiation vs temperature difference | | | | |
| | h outside = 50 $W/m^2K$ | h outside = 5 $W/m^2K$ | h outside = 50 $W/m^2K$ | h outside = 5 $W/m^2K$ |
| | | | | |
| with 1°C temp difference between surroundings and liquid lading | 8.2 W/K for 1 $m^2$ surface | 4.9 | 8.2 W/K for 1 $m^2$ surface | 4.9 |
| 10°C difference | 8.3 | 5.1 | 8.3 | 5.1 |
| 100 | 10 | 6.5 | 10 | 6.5 |
| 200 | 13 | 8.8 | 13 | 8.8 |
| 400 | 20 | 16 | 20 | 16 |
| 500 | 25 | 21 | 25 | 21 |
| 600 | 32 | 28 | 32 | 28 |
| 800 | 49 | 45 | 49 | 45 |

note: all surfaces assumed to have emissivity of 0.9

### 2.4 Defect Assessment Criteria

In this work it is necessary to consider insulation defects and to determine whether these defects are severe enough to require repair. The test criteria will be the CGSB specifications for tank-car thermal insulation and thermal protection.

Computer models were used to simulate heat transfer conditions in a tank-car with thermal insulation defects. These models are based on good engineering principles and where there is some uncertainty in methods, the methods as described in the AAR document SD-053 (TCFIRE: A Model for Prediction of Fire Effects on Tank Cars) will be used. It should be noted that the computer program once called TCFIRE is now called AFFTAC.

The following sections briefly describe the assessment criteria for the insulation defects.

### 2.5 Commodities and Tank Types

In this report we wish to analyse the impact of insulation defects on tank-cars that carry dangerous goods. The output of this analysis will be a simple assessment criterion that can be applied by tank-car inspectors in the field.

A detailed analysis would include many variables such as tank size, wall thickness, tank materials, commodity properties, PRV set pressures and flow capacities, initial lading temperatures, initial lading fill levels, etc. It would not be reasonable to attempt to simulate all of these possibilities and therefore a baseline case will be examined for each system type (i.e. thermal protection and thermal insulation) and then a sensitivity analysis will be performed to show that these baselines reasonably cover all other possible cases.

The baseline for thermal protection will be a 112J340W tank filled with propane. For thermal insulation the baseline will be the 105A300W type tank.

# 3 Analysis of Thermal Protection Systems with Defects

In this section we will only consider tanks that have thermal protection systems. The evaluation of defects will be based on how the defective systems respond to pool and torch fire environments.

## *3.1 Thermal Protection Defect Assessment*

Thermal protection defect assessment will be based on how a tank responds to the two fire test cases – pool fire and torch fire. This determination will be done using computer models of tanks exposed to fire.

In this project we are analysing deficiencies in a critical safety component. If our analysis is wrong then the consequences may be severe. For this reason great care has been taken to ensure model predictions are reasonable.

It is important to understand that computer models are not perfect. They are not the real world and they usually cannot account for all possible factors. It is reasonable to select the best model available as a "standard" model to be used for comparison purposes. However, modelling uncertainties should be identified and quantified. It also makes good sense to include a reasonable factor of safety in any analysis that includes uncertainty.

This section will go into the details of the computer modelling process because it is critical that the computer models represent real-world behaviour of tanks. If an insulation defect can lead to premature tank failure in a fire we want to be able to show this using a suitable model.

## *3.2 Tank Failure by Fire Exposure*

A tank will fail when the stress in the tank wall exceeds the tank strength. Tank wall strength is usually reduced by high wall temperatures in the vapour space of a tank exposed to fire.

The tank wall is put under stress by the tank internal pressure and by temperature gradients that introduce thermal stresses. These hoop stresses and thermal stresses may lead to vessel failure if they are severe enough and they have been applied long enough.

The tank internal pressure causes the well-known hoop and longitudinal stresses. The high wall temperature in the vapour space and the low temperature in the liquid space cause large thermal gradients, which cause the top of the tank to expand and bend the tank into a curve. This results in compressive loading on the tank top and tensile loading on the tank bottom.

These various stresses may lead to tank failure if the tank wall strength is reduced by high temperature.

### 3.2.1 Plastic Failure

With plastic failure, the tank wall is usually heated to a point where the combined stress exceeds the yield strength of the material. Over time, plastic strain results in thinning of the wall locally and this results in elevated stress and more yielding. At some point the wall has thinned enough for a hole or tear to form in the wall, usually starting as a small hole near a local defect.

The tear will normally grow perpendicular to the hoop stress direction. If the stress is high enough and if the surrounding steel is weak enough the tear will run the full length of the tank and the tank will fail catastrophically, leading to a boiling liquid expanding vapour explosion (BLEVE). If the tear stops in stronger material before it runs the full length of the tank then the failure will be limited in size and the outcome will be a massive jet release.

These kinds of failure modes were observed by Birk et al (1997) in their fire tests of 400 litre ASME propane tanks.

### 3.2.2 Brittle Failure

In this case the tank fails with little or no plastic deformation. The pressure and temperature loading of the vessel results in the local failure at a weld or plate defect. This type of failure can also happen if the tank is made from brittle steel. It should be noted that this type of failure is very difficult to predict because it can happen well below the normal ultimate strength of the material.

This type of failure should not happen in a well manufactured tank using ductile steel plate. This type of failure will not be considered in this report.

## *3.3 Tank Thermal Modelling with AFFTAC*

The US DOT and Transport Canada have accepted the computer code AFFTAC as the standard for thermal modelling of tanks in fires. The following section provides an overview and critique of this code and comments on its use in this study of insulation defects. Further details about AFFTAC can be found in Johnson (1998).

Important factors in fire exposure simulation include the following:

- fire heat flux (temperature and radiating properties)
- tank exposure (fraction of tank involved in fire)
- tank thermodynamic response

- PRV operating characteristics
- vapour space heat transfer and wall temperature prediction
- tank material properties and failure criteria

All of the above have uncertainty associated with them. Let us consider each in turn.

### 3.3.1 Pool Fire Heat Flux

In AFFTAC the pool fire is modelled as a black body radiating at 816$^{o}$C. Fire convection is not accounted for separately. Based on pool fire data (see Rew et al (1997) ) these are reasonable and credible assumptions for a kerosene fuelled pool fire. However, other pool fire fuels (such as propane) would require higher fire temperatures.

In the CGSB standard the pool fire test specification requires a pool fire temperature of 871$^{o}$C plus or minus 56$^{o}$C. Therefore AFFTAC is using the lowest possible fire temperature to satisfy this specification. This approach has also been used in this study.

### 3.3.2 Torch Fire Heat Flux

For the torch fire it is usually assumed the heat transfer is by convection due to the jet impingement and by thermal radiation. In AFFTAC the torch is modelled considering radiation only.

The DOT standard torch must have a temperature of 1204$^{o}$C plus or minus 56$^{o}$C and must heat a 16 mm steel plate from ambient temperature to 427$^{o}$C in 4 minutes (plus or minus 30 seconds). In the AFFTAC code this heat up time is achieved if the torch black body radiation is multiplied by a factor of 0.536 (assuming a tank surface emissivity of 0.8 which gives a total factor of 0.536 x 0.8 = 0.43). This factor accounts for the effective emissivity and view factor for the torch used in the 1.2 m x 1.2 m plate tests.

This basic approach will also be used in this study with some modification (as described later).

### 3.3.3 Tank Exposure

The fraction of the tank exposed to the fire is an important factor because it affects how the tank pressurizes and how it empties through the PRV.

### 3.3.3.1 *Pool Fire*

In AFFTAC it is assumed the tank is 100% engulfed in fire. Real pool fires may not engulf a tank 100% depending on the fire size and the wind conditions. This suggests that the AFFTAC assumption is conservative with a factor of safety. However, the AFFTAC model may not be modelling a fully engulfing fire exactly.

The AFFTAC computer model has been developed to agree with the full scale fire test results (see Townsend et al (1974) ) of protected and unprotected propane tanks (assuming a fire temperature of 816$^{o}$C). It is very likely that these tanks were not fully engulfed in fire because of wind effects. This means that the vapour space wall temperatures in the fire test were not as high as they could have been with a full engulfing fire. This means then that the wall temperatures predicted in AFFTAC may not be conservative.

When an analysis has uncertainty, then a factor of safety should be applied. This will be discussed further later in this report.

### 3.3.3.2 *Torch Fire*

AFFTAC allows the user to specify the area of tank surface exposed to a torch. In the original DOT steel plate tests, the plate measured 1.2 m x 1.2 m. This exposure area has now been applied in the AFFTAC computer model as a default area for modelling a tank-car exposed to a torch fire.

If the AFFTAC model is being used to model the real world tank-car system then it should be modelling a credible torch environment. A 1.2 m by 1.2 m torch area (i.e. 1.5 m$^2$) covers less than 1% of the surface of a 112 type tank-car and this in no way represents a credible torch fire accident. In a credible torch fire impingement case where one tank's PRV flare is impinging on another tank, the area of coverage could be more like 20-40% of the tank surface.

If it is assumed that the torch covers 1% of the tank then the tank will pressurize very, very slowly. In the CGSB standard a thermally protected tank must survive a torch for 30 minutes. If the tank does not pressurize in that 30 minutes it has a much better chance of surviving the test. Therefore assuming such a small torch area is not a conservative or credible assumption.

In this study, two torch cases will be considered. One where the torch area is 1.5 m$^2$ and one where the coverage is 30% of the tank surface.

### 3.3.4 Tank Thermodynamic Response

In AFFTAC it is assumed that the tank pressure is dictated by the liquid thermal properties. It is also assumed the liquid is isothermal, which means the heat added by the fire is distributed equally throughout the liquid. This assumption is incorrect based on numerous test results of tanks in fires (see Townsend et al (1974), Appleyard (1980), Birk (1983), Moodie et al (1988), Hadjisophocleous et al (1990), Birk and Cunningham (1997) ). In reality the liquid temperature is stratified in the tank during fire exposure (hot near the top and cooler near the bottom). This is due to free convection within the tank.

Fire tests of tank-cars have shown that the tank usually pressurizes much faster than predicted by uniform liquid thermal models. In real tanks, the liquid is not heated uniformly and this results in a hot layer of liquid which dictates the tank internal pressure. The full scale uninsulated tank tests of Townsend et al (1974) showed that the PRV opens after about 2 minutes and because of this early opening of the PRV the tank never went shell full of liquid. A uniform liquid thermal model will not predict this behaviour. With a uniform liquid thermal model the tank pressurizes about five times slower, which means the tank goes shell full of liquid and this causes the PRV to open.

This detail is very important to remember because when the tank goes shell full it means there is no vapour space and therefore there is no hot wall that can lead to tank failure. If a uniform liquid thermal model predicts a shell full condition in a tank exposed to fire then it is very likely the model will give poor predictions of wall temperature.

The uniform liquid temperature assumption in AFFTAC results in the following simulation errors:

- tank pressurization is too slow
- first opening of PRV is late
- late opening of PRV may result in tank going liquid full
- liquid full causes elimination of vapour space and erroneous wall temperature estimates

These errors mean that AFFTAC results may not be conservative. The AFFTAC assumption of isothermal liquid is not very important if the tank is well insulated and therefore the fire heat is added very slowly to the tank. However, if the tank thermal protection system is damaged then the AFFTAC assumptions may lead to significant errors.

For example,

- With torch exposure the AFFTAC model predicts very little pressure build-up in the tank due to the small torch exposure area. However, it is very likely that an actual tank system would experience pressure build-up due to liquid temperature stratification.

- In a pool fire test of an unprotected tank (see Townsend et al (1974) ) filled 95% full of propane the PRV opened after about 2 minutes due to liquid temperature stratification. With AFFTAC this is not predicted. With AFFTAC the tank goes shell full of liquid before the PRV opens and this results in complete wall cooling. This was not observed in the field.

The conclusion here is that AFFTAC may not be conservative in the prediction of pressure build-up and vapour space wall temperatures. This is especially true for tanks with damaged or inadequate thermal protection systems. Once again this adds uncertainty to the analysis and therefore a factor of safety should be used.

### 3.3.5 Pressure Relief Valve Operation

The PRV is a critical part of this analysis. In AFFTAC the PRV opening and closing are modelled using four reference pressures.

- Start-to-discharge pressure (P std).
- PRV is fully open when the pressure is 103% of the std pressure (P fo).
- The valve is assumed to reclose fully at 82% of the std pressure (P fc).
- The closing stroke is assumed to start when the pressure has dropped to 88% of the std pressure (P c).

The blowdown of the valve is typically = pop pressure – reclose pressure. In this case AFFTAC is assuming a 21% blowdown. It should be noted that the maximum allowable blowdown is 20% (see Appendix K for PRV for CGSB requirements ) but PRVs may not have a blowdown this large.

In AFFTAC the open area of the PRV is assumed to be linearly interpolated between these reference pressures (for details see Johnson (1998) ). It is not clear where these linear PRV operating characteristics have been obtained. It is very likely this method was used because it was numerically stable to do it this way in a computer model. In our opinion this is not a conservative way to model PRV action.

Real pop action relief valves do not generally fit the model described above. Typical relief valves have the following reference pressures:

- start-to-discharge pressure (simmer pressure, very little flow)
- pop pressure (pop to full open)
- reclose pressure (PRV closes to full close)
- reseal pressure (PRV

With real valves the open area is not linearly related to these pressures. With a pop action PRV the valve is supposed to pop full open at some pressure and reclose fully when the pressure drops by some amount. A pop action PRV normally does not sit partially open (although they do sometimes, but not with great consistency) unless the valves flow capacity is being approached (see Pierorazio and Birk (1998) ). In pop action PRVs the valves cycle open and closed if the tank is generating vapour at a rate less than the PRV flow capacity. If the valves flow capacity is exceeded this results in the PRV being held full open. If the PRV is undersized for a given tank then the pressure will continue to rise even if the PRV is full open.

A cycling PRV exposes the tank to a pressure range between a high of the pop pressure and a low of the reclose pressure. This range may vary during a fire exposure event. The first few pops may be at higher pressures due to valve seat sticking (see Pierorazio and Birk (1998) ). Later in the fire the pop and reclose pressures may decrease due to PRV spring softening due to high spring temperatures. However, this spring softening may not be significant until the tank is nearly empty.

In the simulation of a 112J340W tank (3 m diameter, 33000 gal capacity, 19 mm wall, surface emissivity = 0.9, 95% full, initial T = 20$^{o}$C) with propane and 100% insulation defect (i.e. steel jacket but no insulation in gap), the peak PRV flow rate was estimated by AFFTAC to be about 17.9 kg/s (2360 lb/min) at about 24 minutes into the fire when the tank is 94% full of liquid. This is the full PRV flow capacity and by AFFTAC's PRV analysis requires the tank pressure to be 103% of the std pressure (i.e. 103% of 255 psi = 263 psi). However, at 60 minutes when the tank is only 30% full of liquid the PRV flow is 11.4 kg/s (1500 lb/min). This is well below the PRV flow capacity and therefore by AFFTAC's PRV model the tank pressure was only 222 psig. However, in reality the PRV probably would have been cycling between 210 and 263 psi. The reduced pressure assumed in AFFTAC results in lower calculated hoop stress and reduced chance of tank failure, which may be unrealistic and not conservative.

On the other hand a conservative analysis would assume that the PRV would cycle between its pop and reclose pressure when the full flow capacity of the PRV is not being used. For example, with a 112J340W tank the PRV std (start to discharge) pressure is 255 psig. At 100 min AFFTAC predicts tank pressure at 210 psi, while the tank burst pressure was 214 psi. This is a very close pass result. However, if the PRV was cycling between 103 and 82% of the std (i.e. 262 and 210 psi), then the tank would have failed during the high pressure part of the PRV cycle.

This is another example where the AFFTAC result may not be conservative. In this study it will be assumed that the tank is exposed to a hoop stress associated with a tank internal pressure = 103% of the PRV start-to-discharge pressure. In other words, the high side of the PRV cycle pressure will be used to calculate the hoop stress.

### 3.3.6 Prediction of Wall Temperatures

The wall temperatures in the liquid filled part of the vessel exposed to fire will be effectively cooled by the liquid. As a result the liquid wetted wall temperatures will be close to the liquid temperature. The vapour space wall temperatures will get very hot because of the poor cooling effect of the vapour. A thermal model must be able to accurately predict vapour space wall temperatures, to predict tank failure.

The vapour space wall temperature is affected by the vapour, and the liquid. The hot vapour space wall convects heat to the vapour and radiates heat to the vapour, the liquid surface and to itself (because the wall is concave). The liquid surface will be relatively cool (its temperature is limited by the PRV set pressure). Therefore, the vapour space wall will be cooler for high fill levels than for low fill levels. Also, heat will be conducted from the hot vapour space wall to the wall cooled by liquid. This also results in lower vapour space wall temperatures when the fill level is high.

When the PRV opens there will be swelling of the liquid due to bubble formation in the liquid. This causes the liquid level to rise and cause the vapour space wall area to shrink. Of course if the tank goes liquid full then there is no vapour space wall and all of the tank wall is cooled. As long as the tank wall is cool and the pressure in the tank is near the PRV set pressure the tank is not expected to fail.

The following assumptions are used to model the vapour space heat transfer in AFFTAC:

- vapour space cooling is dominated by thermal radiation to the liquid surface
- absorption of radiation by the vapour is ignored
- the view factor from the vapour space wall to the liquid surface is equal to the ratio of (liquid surface area/ vapour space wall area)

With these assumptions the vapour space wall temperature will continue to increase as the liquid level drops. When the liquid is emptied from the tank the vapour space wall temperature will increase more rapidly. With this assumption it is very likely that a tank will fail soon after it is empty if it does not fail earlier.

In other words, with AFFTAC a critical time for the tank is the time to empty. With AFFTAC, a tank deficient of insulation will probably fail soon after it is empty if it does not fail earlier. This will be demonstrated later in this report.

### 3.3.7 Effect of Liquid Fill

If a tank is exposed to fire then heat will be added to the tank and lading. If a tank starts off 90% full of liquid and if heat is added to the liquid uniformly (as assumed in AFFTAC) then the tank will go liquid full before the PRV opens and vents material. This is a possible scenario for a well insulated tank where heat is added slowly. However, it is known from fire tests of uninsulated tanks (see Townsend et al (1974) ) that the heat is not added uniformly to the liquid and therefore the PRV usually opens before the tank is shell full of liquid.

In AFFTAC the temperature increase will cause the liquid lading to expand and at some point the vessel will become liquid full. Once the tank is liquid full, AFFTAC calculates the necessary PRV liquid flow to allow for liquid thermal expansion.

Table 3-1 illustrates how propane expands with increasing temperature.

**Table 3-1: Propane Liquid Expansion with Temperature**

| **Propane T ($^oC$)** | **Expansion Factor e** | **1/e** |
|---|---|---|
| | | |
| 20 | 1 | 1 |
| 25 | 1.016 | 0.984 |
| 30 | 1.032 | 0.969 |
| 35 | 1.05 | 0.952 |
| 40 | 1.07 | 0.935 |
| 43 | 1.082 | 0.924 |
| 45 | 1.091 | 0.917 |
| 50 | 1.114 | 0.898 |
| 55 | 1.14 | 0.877 |
| 60 | 1.169 | 0.855 |

If a typical propane PRV is set at around 2 Mpa (290 psi) then this equates to a propane temperature of about $60^oC$. In this case any tank starting at $20^oC$ that is filled more than 85% is likely to go liquid full before the PRV opens (this ignores liquid temperature stratification which can open the PRV earlier).

If the tank goes liquid full the entire tank wall will be cooled and as long as the PRV limits the pressure, the tank will not fail due to high wall temperatures. For high wall temperatures to be established the liquid level must drop to expose vapour space wall. The erroneous calculation of shell full condition leads to incorrect vapour space wall cooling. This feature of AFFTAC may result in the erroneous calculation of low vapour space wall temperatures. Once again this is an example of where AFFTAC is not conservative.

In a typical scenario where a tank starts 95% full the tank will take several minutes to reach a liquid full state. In the AFFTAC model, if the tank does not fail before the tank goes liquid full then the rising liquid will cool the wall and save the tank. Failure of the tank must now wait until the liquid begins to drop so that a vapour space can be reformed.

Consider the following scenarios:

i) tank engulfed or partially engulfed in pool fire
-- rapid vapour space wall heating
-- rapid liquid heating

-- tank goes liquid full before critical wall temperature achieved
-- probable failure after liquid level drops from liquid full condition

ii) tank exposed to local torch fire
-- very rapid vapour space wall heating
-- lower liquid heating
-- tank may fail before it goes liquid full

The time to failure in these different cases depends on how quickly the vapour space wall is heated and how quickly the total liquid is heated. The wall heating causes it to weaken. The overall heating causes the liquid level to rise to cool the wall. It is a race between the wall heating and the rising liquid to determine whether the tank will fail early or later in the AFFTAC model.

The above process is not so true for tanks that are only filled to say 80%. These tanks will not go liquid full before the PRV opens.

### 3.3.8 Tank Material Properties and Failure Criteria

The tank material properties used in AFFTAC use the minimum allowable tensile strength based on the steel type. Therefore, there is some factor of safety here because the steel properties are usually better than the minimum allowed. However, the failure model used in AFFTAC is very simplistic and is not conservative.

The AFFTAC failure model assumes tank failure when the tank hoop stress exceeds the material tensile strength at the elevated wall temperature. However, at elevated temperatures steels can creep resulting in creep ruptures at stress levels well below the material tensile strength. Stress rupture properties of steels at elevated temperatures have some scatter and therefore there is some uncertainty here. As a result, AFFTAC failure estimates may not be conservative.

### 3.3.9 Tank Failure Analysis Methods

In this study the tank failure analysis assumes a ductile type failure. Ductile tank failure can be predicted in a number of ways. The following list includes methods increasing in complexity.

i) when hoop stress exceeds ultimate tensile strength based on peak wall temperature (or when tank pressure exceeds burst pressure based on peak wall temperature).
ii) when hoop stress exceeds yield stress based on peak wall temperature. The yielding accumulates as a function of stress, wall temperature and time until tank failure is indicated.

iii) detailed shell temperature and stress is calculated using finite element methods and plastic deformation is calculated as a function of time.

The method described in i) above is the simplest method to implement because it requires the least amount of data input. The data needed is simply steel ultimate strength vs temperature. The AFFTAC code uses the method described in i) above. This method will be used in this study as the baseline method to predict failure. This method is simple but it may not be a conservative approach. With this method, if the tank stress does not exceed the ultimate stress then the tank will not fail. However, in practice it is known that steel failure can happen with stress below the ultimate tensile strength of the material if the load is applied long enough to allow accumulation of creep.

In the second method, if the stress exceeds the yield but is less than the ultimate the tank can still fail if the stress is applied for a long enough period of time. With this method more property data is needed. The data must include time to tensile failure as a function of steel temperature and stress. This model is more difficult to implement because of the added data requirements. Very limited data is available for the steels of interest in this study.

The third method noted above is the most complex and most difficult to perform. It requires lengthy computer simulations. In this study we are interested in hot spots resulting from insulation defects. These hot spots will not be subjected to the same stress as the rest of the tank and therefore this method may be required to give the most accurate picture of the effects of insulation defects.

### 3.3.10 Tank Pass or Fail

For the CGSB standard a thermally protected tank must not release material (other than through the PRV) for 100 minutes when exposed to an 816$^{o}$C pool fire. In this study the simulation results will be shown for both time-to-empty and time to failure.

A very valid question is raised here – will a tank fail after it is liquid empty? The AFFTAC model contains heat transfer assumptions that may result in tank failure shortly after it is liquid empty.

The full scale fire test of an insulated tank (see Townsend et al (1974) ) resulted in tank failure when the tank was 2% full. However, other tests of tanks (see Appleyard (1980), Birk et al (1997) ) have shown that once the tank is empty of liquid the test is basically over and the tank will not fail. This is usually because once the liquid is gone the tank usually looses pressure through the PRV. The high temperatures of the vapour only environment usually result in overheating of the PRV spring which causes spring relaxation. This spring weakening causes the blowdown of the tank to atmospheric pressure. Once the pressure is gone the tank cannot fail.

Is it safe to assume the tank will not fail after it is empty? Probably not. It is conservative to assume that the wall will continue to get hotter after the tank empties and this may result in failure. Therefore it probably is not safe to say that a tank passes the test if it empties before the 100 min time limit for pool fires.

## *3.4 Thermal Modelling of Defects with TMID Code*

A program has been developed to model a tank with part of the tank surface covered by insulation defects. This program has been given the name TMID for Tank Model with Insulation Defects. The TMID thermal model was used to simulate the effects of torch or pool fire impingement on a rail tank-car equipped with defective thermal protection. A complete listing of TMID is in Appendix C.

The thermal model developed for this study has used assumptions similar to those used in the US FRA computer code AFFTAC (Ver 3.0, Nov 1998). This was done as directed by Transport Canada. It has already been suggested that AFFTAC has some modelling deficiencies that can affect the results of this study.

### 3.4.1 TMID Assumptions

TMID is based on assumptions similar to the AFFTAC code but with the necessary features to model insulation defects that cover only parts of the tank. These include:

- thermal radiation in gap between steel jacket and wall
- vapour space radiation with hot spot (defect) cool zone (insulated wall) and liquid surface
- effect of liquid level on defect temperatures (i.e. as liquid level drops it exposes defective area to vapour space, etc).
- defect is modelled as a strip that runs the entire length of the tank on one side (strip defined by angle from tank top to defect top and bottom)

In TMID it is assumed that the defect temperature and stress are uniform. In other words, two- and three-dimensional effects such as circumferential or axial heat conduction or thermal expansion are ignored. These effects will be discussed later in this report when local effects are considered.

The maximum normal stress in the defect is assumed to equal the tank hoop stress. The tank material strength in the defect is assumed to equal the strength based on the material properties (i.e. ultimate tensile strength) for the defect temperature.

The code is not fully validated (because data is not available for validation). The code does appear to be producing reasonable results. Where possible, simulation results have been compared with AFFTAC results with reasonable agreement (see Appendix D).

The code is not exactly the same as AFFTAC. The most significant difference is associated with how the PRV is modelled. In AFFTAC the PRV is modelled in the following way:

- PRV opens at 103% of PRV start-to-discharge pressure (std)
- PRV recloses at 82% of std
- PRV flow capacity based on PRV rating (SCFM air), rating pressure (usually 110% of start-to-discharge pressure) and assumed flow coefficients (0.8 for vapour, 0.6 for liquid).
- PRV flow is linearly related to opening and closing pressures

In a thermally protected tank with partial insulation defect, the PRV will most likely cycle open and closed between the PRV open pressure and close pressure. This assumes the PRV has been sized properly (as per AAR PRV sizing formulas) and it therefore provides more than enough flow capacity for a thermally protected tank. If a PRV std pressure of 280 psig is used (i.e. for 112J340W tank) then this means the tank pressure cycles between about 1.6 and 2.0 MPa (230 and 288 psig). In TMID it is assumed that the tank will fail at the high point of this cycle, therefore in TMID it is assumed the tank pressure is held at 103% of the std pressure and the PRV mass flow is calculated based on energy considerations to maintain this pressure in the tank.

In TMID the tank is modelled as a long cylinder (i.e. it is a two-dimensional model), filled with liquid and its vapour. The model includes the following basic elements.
The following sections briefly identify differences between TMID and AFFTAC.

### 3.4.2 Fire Heat Transfer

The fire heat transfer is the driver in this analysis. Two different fire types are considered, a pool fire and a torch fire.

#### *3.4.2.1 Pool Fire*

In TMID it will be assumed that the fire radiates as a black body at 816$^{o}$C. However the tank outer surface emissivity will be set at 0.9 (AFFTAC uses 0.8), which is considered to be more valid for tank-car finishes (see Appendix I).

#### *3.4.2.2 Torch Fire*

For the torch fire the temperature will be 1204$^{o}$C as in AFFTAC. In AFFTAC a factor of 0.536 is applied like an emissivity to the torch radiation to satisfy the temperature vs time requirement. However this factor is than multiplied by the tank emissivity in AFFTAC (0.8) to get the final heat input (0.536 x .8 = 0.43).

In TMID the factor is 0.45 and the surface emissivity is 0.9 (0.9 x .45 = 0.41).

### 3.4.3 Thermodynamic Model

The tank interior volume is divided into two basic regions, the vapour space and the liquid space. The thermodynamic model is needed to predict how these regions change as heat is added to the tank from the fire.

In the AFFTAC model these two regions are assumed to be of uniform temperature and in saturated thermodynamic equilibrium. Heat enters the liquid space through the liquid wetted wall by convection, and at the liquid surface by radiation from the vapour space wall. Heat enters the vapour by convection from the vapour space wall.

This same model is used in TMID. This model ignores liquid temperature stratification, which can have an important impact on the predicted results.

A more accurate thermal model would consider the liquid temperature stratification. Birk (1983) used a two-node thermal model (i.e. saturated vapour space with saturated liquid boundary and sub-cooled liquid core). With this model it was possible to predict the correct time for first PRV action and the correct liquid level. With the correct liquid level it is possible to accurately predict the wall temperatures in the vapour space.

Since AFFTAC is the standard computer code used by the FRA and Transport Canada, its basic methods will be used in this study. Therefore a single node thermal model has been used in this study. Where possible the report will point out where results may not be accurate. In addition a factor of safety will be applied to offset the simulation uncertainties.

### 3.4.4 Wall Heat Transfer Models

The wall, thermal barrier and jacket heat transfer is by pure conduction. If no insulation is present in the gap between the tank and jacket then the heat transfer in the gap is by thermal radiation and convection. For defects it has been assumed that air is in the gap and the defect is closed so that the air is stagnant (i.e. convection is ignored).

The time varying wall temperatures are calculated using an explicit finite difference technique. The following wall temperatures are calculated for this study.

- vapour space wall under insulation defect
- vapour space wall in insulated area
- liquid space wall under insulation defect
- liquid space wall in insulated area

Temperature gradients through the steel are ignored; circumferential conduction of heat around the tank wall is also ignored. These assumptions are consistent with the AFFTAC code and are conservative.

### 3.4.5 Thermal Radiation in Vapour Space

The interior heat transfer is by thermal radiation and convection. The tank vapour space interior surfaces are divided into the following regions:

| | |
|---|---|
| 1g | wall area of defect in vapour space |
| 1f | wall area of defect in liquid space |
| 2g | protected/insulated wall area in vapour space |
| 2f | protected/insulated wall area in liquid space |
| 3 | liquid surface |

The vapour space wall areas will be hot due to the poor cooling effects of the vapour. The defect area in the vapour space will be hotter than the insulated part of the vapour space wall. The liquid wetted wall will be cool due to the strong cooling effects of the liquid.

In the vapour space heat is lost from the wall by thermal radiation and convection. When the wall gets hot thermal radiation will dominate. The wall areas will radiate to the other visible wall areas and liquid surface. The vapour will also absorb thermal radiation.

Absorption of radiation by the vapour has been ignored in the AFFTAC model. This assumption is probably conservative. It should be noted that the vapour will in fact absorb thermal radiation significantly. This absorption of radiation by the vapour will reduce the cooling influence of the liquid surface. In other words, if vapour absorption is included then the wall temperature in the vapour space will not rise as rapidly with dropping liquid level.

The radiation view factors in the vapour space were calculated using Hottel's crossed strings method (see Hottel and Sarofim (1967) ). Three zones are considered – 1g, 2g and 3. This method applies for two-dimensional systems (i.e. very long cylinders) ) and therefore ignores the effects of the tank ends. It also means the model is limited to defects that run the entire length of the tank.

The tank inside surface is assumed to have an emissivity of 0.9 (AFFTAC assumes 0.8). The liquid surface is assumed to have an emissivity of 0.9 (as per AFFTAC).

### 3.4.6 Convection in Vapour Space

For convection, AFFTAC assumes the following:

i) when tank is more than 0.4 full but less than 0.9 full, and the PRV is venting

$$h = 0.05 \text{ Btu/hr ft}^{2\,o}\text{F} = 0.284 \text{ W/m}^2 \text{ K}$$

A larger value is used when the PRV is venting more than 1500 lb/min. It is not clear what the code uses in this case. In any case, the above convective coefficient is so small it can be ignored compared to the thermal radiation.

ii) when tank is empty of liquid AFFTAC assumes

$$h = 1.0 \text{ Btu/hr ft}^{2\,o}\text{F} = 5.7 \text{ W/m}^2 \text{ K}$$

It is not clear why this is used. Why should there be a 20 x increase in h suddenly when the liquid is gone? In fact, the heat transfer to the vapour will be primarily by thermal radiation, but this has been ignored in the AFFTAC model.

In the present model, convection has been ignored in the vapour space (the AFFTAC assumed h is negligibly small). The simulation ends when the tank is empty. The AFFTAC assumptions normally lead to tank failure shortly after the tank is empty (i.e. if it does not fail earlier).

### 3.4.7 Convection in Liquid Space

Thermal radiation is ignored for the liquid wetted wall. The heat transfer is by liquid convection or boiling. In the liquid space the wall is strongly cooled by convection and boiling. In this case h is assumed to be 1000 W/m$^2$ K. This forces the tank wall temperature to remain close to the liquid temperature which it does based on observations from various fire tests of tank-cars.

### 3.4.8 Properties of Thermal Insulation

The thermal properties of the high temperature insulation material is an important factor in this analysis. Fiberfrax or Kaowool ceramic blanket has the following thermal conductivity (k with units W/mK) as a function of temperature (see data sheets in Appendix F).

Table 3-2: Summary of Thermal Data vs Temperature for Fibrefrax Insulation (72 $kg/m^3$ density)

| **Temperature ($^oC$)** | **Thermal Conductivity (W/mK)** | **Thermal Conductivity (Btu in/hr/$ft^2$/F)** |
|---|---|---|
| | | |
| 20 | 0.03 | 0.2 |
| 200 | 0.06 | 0.4 |
| 600 | 0.17 | 1.2 |
| 700 | 0.23 | 1.6 |

If we assume a 13 mm layer of blanket (as used in thermal protection system) at $600^oC$ we expect a conductance of 0.17/0.013 = 13.1 $W/m^2K$. At $650^oC$ we expect 0.202 W/mK conductivity and 15.5 $W/m^2K$ conductance for 13 mm thickness. If this 13 mm layer is compressed to half of this (6.5 mm) the insulation density increases to about 140 $W/m^2K$ and this reduces the k to about 0.13 W/mK (0.85 Btu in/ft $hr^oF$) at around $600^oC$. This translates to a conductance of 20 $W/m^2$ K (3.4 Btu/hr $ft^{2o}F$).

In an earlier report, Johnson (1995) stated that a thermal protection system that satisfies the pool fire requirement (CFR 179.105-4) would have at most a conductance of 22. 7 $W/m^2$ K (4.0 Btu/hr $ft^{2o}F$). In this report it will be assumed that the thermal protection system has a thermal conductance of 22.7 $W/m^2$ K (4.0 Btu/hr $ft^{2o}F$) except where there are defects. A sensitivity analysis will also be performed to show the effects of ideal insulation properties as described above.

### 3.4.9 Ultimate Tensile Strength

In the AFFTAC model, the following equation is used to account for the high temperature degradation of TC-128. AFFTAC assumes that all low to moderate alloy carbon steels have similar high temperature properties.

for T wall < 1260 R

$$FCTR = 1.0 - 0.54(TTNV - 0.46)^4$$

for T wall > 1260 R

$$FCTR = 1.74 - 1.17(TTNV - 0.46)$$

where

TTNV = vapour space wall temperature ($^o$ R/1000)
FCTR = ratio of ultimate stress at T to ultimate stress at ambient temperature

The ultimate stress of the material can then be determined by multiplying FCTR times the material ultimate strength at ambient temperature. The tank burst strength is determined by multiplying the FCTR times the ambient temperature burst strength.

TC 128 pressure vessel steel has a minimum tensile strength of 560 MPa (81000 psi) at ambient temperature. This ultimate strength has been used in this study.

### 3.4.10 Stress, Temperature and Time to Failure Data

Creep rupture data for TC 128 was reported by Anderson and Norris (1974) This data is needed for predicting the time to failure if the tank stress is below the ultimate tensile strength, but it is high enough to cause plastic deformation over time. This data will be shown later when failure prediction uncertainty is discussed.

### 3.4.11 Factor of Safety (FOS)

As stated earlier it seems prudent to include a factor of safety (FOS) to account for some uncertainties in the analysis of a thermally protected tank with insulation defects exposed to a pool or torch fire.

For a pool fire simulation, one way to define FOS is as follows:

$FOS_{bp}$ = (tank burst pressure at 100 minutes)/(tank pressure at 100 minutes)

The subscript bp is used to indicate that the FOS is based on burst pressure. Another way to define FOS is based on time to failure.

$FOS_t$ = (time to predicted failure)/100 min

It is somewhat arbitrary which FOS we use. In this report we will use $FOS_{bp}$ and from this point on, it will be designated only as FOS.

The reader is cautioned on how to interpret FOS. An FOS = 2 means that the tank is two times as strong as it needs to be at the 100 minute time. However, at high wall temperatures this FOS = 2 may only provide 5 or 10 minutes of extra time before failure.

In AFFTAC the tank fails when FOS = 1.0. Therefore, we want the FOS to be greater than 1.0 to account for analysis uncertainties. The question is how much greater than 1.0 is prudent? We should set an FOS that accounts for the uncertainty in the following:

- fire exposure and temperature
- PRV behaviour (open and closing pressures)
- tank fill level effects (shell full, etc.)
- tank material properties and failure criteria

Let us recall the DOT insulation test standard where a sample of insulation was tested by fire exposure on a 1.2 m x 1.2 m steel plate. The limit of acceptance for insulation materials was as follows:

| | |
|---|---|
| pool fire test | test plate T < 427°C for 100 minutes |
| torch fire test | test plate T < 427°C for 30 minutes |

If we compare the temperature limit of 427°C to the current AFFTAC simulation standard of tank failure we can derive an FOS from the test standard. The current simulation standard (AFFTAC) will predict tank failure (for 112 type car, 95% full of propane, just steel jacket) from pool fire exposure, if the tank wall reaches about 640° – 700°C (depending on wall thickness and PRV pressure) within 100 minutes. At 640°C TC 128 steel has a strength of about 200 Mpa. At 427°C TC 128 has a strength of about 450 Mpa. In other words, the FOS from the plate tests was 450/200 = 2.3.

If we want the same FOS to apply to a tank with insulation defects, we must set the allowable wall temperature to 427°C for the pool and torch fire simulations. Unfortunately if we do this all insulation defects will fail the simulation tests. This is not an unreasonable outcome because there is no insulation in the defective areas!

Let us try to select some intermediate FOS between 2.3 and 1.0. Let us consider the following uncertainties.

#### *3.4.11.1 Uncertainty in Fire Temperature*

The current pool fire simulation condition assumes a fire temperature of 816°C. However the actual plate test specification requires 871°C plus or minus 55°C. A fire at 871°C produces 1.22 times as much radiant heat flux as a fire at 816°C. This increased heat flux causes higher wall temperatures and a reduced tank burst pressure.

The plate test simulates a kerosene or JP pool fire as used in pool fire tests of propane tanks (see for example Townsend et al (1974) ). However, in the real-world scenario the pool fire fuel could be other fuels such as propane, pentane, etc. These fuels can burn hotter than kerosene. Actual flame temperatures depend on many factors including the fire scale, fuel, and wind conditions. **Table 3-3** shows how different pool fire fuels have different effective radiating temperatures.

As shown above pool fire temperatures can be much hotter than 816°C. Nakos and Keltner (1989) reported very detailed heat transfer measurements in a JP4 pool fire (9m x 18m rectangle). They recorded average total heat fluxes of between 120 and 128 kW/m$^2$ , which is in good agreement with the above table. Therefore, it is clear that the AFFTAC assumption of 816°C is not conservative for a large JP4 pool fire.

Table 3-3: Maximum Pool Fire Radiating Temperatures (data from Rew et al (1997) )

| | **Maximum Pool Fire Heat Flux to Cool Surface** | **Maximum Effective Radiating Temperature (assuming flame emissivity of 1.0 and flux is 100% radiation)** |
|---|---|---|
| AFFTAC | 79 kW/m$^2$ | 816$^o$C |
| plate test | 97 | 871 |
| | | |
| butane | 225 | 1140 |
| diesel oil | 130 | 960 |
| gasoline | 130 | 960 |
| JP 4 | 130 | 960 |
| kerosene | 130 | 960 |
| propane | 250 | 1180 |
| octane | 200 | 1100 |
| LNG | 265 | 1200 |
| pentane | 200 | 1100 |
| toluene | 130 | 960 |

Let us for now consider the case of a 871$^o$C fire (which is still not conservative). Let us consider the following example:

112J340W tank fill 90% with propane
17.5 mm wall with TC 128
insulation defect from 0-90$^o$ around tank side (25% defect)
initial temperature 16$^o$C
PRV limited tank pressure P = 2 MPa

Table 3-4: TMID Simulation Showing Effect of Fire Temperature

| **Conditions at 100 minutes** | **Fire T = 816$^o$C** | **Fire T = 871$^o$C** |
|---|---|---|
| | | |
| Wall Temperature ($^o$C) | 597 | 649 |
| Tank Pressure (MPa) | 2 | 2 |
| Burst Pressure (MPa) | 2.9 | 2.2 |
| FOS | 1.45 | 1.1 |
| Fill | 37% | 25% |

It can be shown that a 22% increase in heat flux will result in a 9% increase in wall temperature (under jacket) and this leads to a 25% decrease in burst pressure. Therefore a factor of safety of 1.25 is suggested here to account for possible variations in fire conditions.

#### *3.4.11.2 Uncertainty in PRV Behaviour*

Another example of uncertainty is the PRV open and close conditions. In AFFTAC it is assumed that the PRV pops open at 103% of the start-to-discharge (STD) pressure and it closes at 0.82 of the STD. In TMID it is assumed the tank pressure is maintained at 103% of STD pressure and the PRV mass flow is whatever is needed to satisfy the energy balance on the tank.

For a 112J340W tank with a nominal 5.9 MPag (850 psig) burst pressure the maximum STD pressure = 0.33 x 5.9 = 1.93 MPag (280.5 psig). There is a 3% tolerance on this pressure and therefore the pressure could go up to 1.99 MPag (289 psig). The flow rating pressure is 110% of the STD pressure or 2.19 MPag (318 psig).

The main question is when will the PRV open fully or pop to relieve pressure? The STD pressure is when flow begins in the PRV. This could be when the PRV begins to leak. Full flow is not obtained until the PRV pops open. A full pop must happen at or before the flow rating pressure which is 110% of the STD pressure, or it may happen at the STD pressure. Therefore, there is an uncertainty of 10% here.

Smaller scale PRVs (1-2 inch) have been shown to have variability in the pop pressure (see Pierorazio and Birk (1998) ). No data exists that can be used to describe the variability of tank-car scale PRVs. Therefore an FOS of 1.1 will be applied to the tank burst pressure to account for PRV uncertainty.

#### *3.4.11.3 Tank Initial Fill*

Tank initial fill uncertainty is harder to quantify because it has several effects. Lower fills mean higher wall temperatures and faster emptying of the tank. Low fills also affect whether the tank will go shell full during the fire exposure event. In this study we were told to assume the tank was filled to its maximum allowable level for propane (i.e. when tank heated from 16 C to 43 C tank goes 99% full). For this we have assumed 90% initial fill. Let us now consider the case with a 80% fill.

Let us again consider the following example:

112J340W with propane
17.5 mm wall with TC 128
insulation defect from 0-90$^o$ around tank side
initial temperature 16$^o$C
PRV limited tank pressure P = 2 MPa

**Table 3-5: Simulation Showing Effect of Initial Fill**

| Conditions | Initial fill 90% | 80% |
|---|---|---|
| | | |
| Wall Temperature ($^oC$) at 100 min | 597 | 603 |
| Tank Pressure (MPa) at 100 min | 2 | 2 |
| Burst Pressure (MPa) at 100 min | 2.9 | 2.8 |
| Fill at 100 min | 37% | 30% |
| FOS at 100 min | 1.45 | 1.40 |
| | | |
| Time for wall to T = 427$^oC$ | 16 min | 16 min |
| Time to liquid full | 29 min | not filled |
| Time for wall to recover to 427$^oC$ | 54 min | na |
| Fail time with FOS = 1.6 | 75 min | 56 min |

This uncertainty in fill can affect the overall calculation results significantly because it affects whether the tank will go liquid full or not. The uncertainty here is that AFFTAC predicts a shell full condition when in the real world, a shell full condition may not occur. As can be seen from the table a small change in initial fill can have a dramatic effect on how the vapour space wall heats up.

However, the fill level change shown above does not appear to change the overall outcome of the simulation at the 100 minute mark. Therefore, no factor will be applied here to account for fill level uncertainty.

#### *3.4.11.4 Uncertainty in Tank Material Properties*

In AFFTAC the tank is assumed to fail immediately when the tank pressure equals 100% of the tank burst pressure. However, if the tank pressure were say 99% of the burst pressure then, in reality, the tank would probably still fail after some time is accumulated. This is because of the stress-temperature-time rupture characteristics of steel. This is not accounted for in AFFTAC. Therefore an FOS should be applied to the basic failure criteria to account for this uncertainty.

Anderson and Norris (1974) show stress-temperature-time to rupture data for TC 128 steel samples. This data is shown in Table 3-6.

**Table 3-6: Stress-Temperature-Time to Rupture Data for TC 128 Steel (from Anderson and Norris (1974) )**

| Sample Temperature ($^{o}$C) | Stress in Sample (000's of psi) | % of Ultimate Strength at Temperature | Time to Failure (min) |
|---|---|---|---|
| | | | |
| 482 | 77 | 100 | 0 tensile test |
| 482 | 65 | 84 | 16 |
| 482 | 62 | 81 | 244 |
| | | | |
| 566 | 60 | 100 | 0 tensile test |
| 566 | 49 | 82 | 52 |
| 566 | 42 | 70 | 138 |
| | | | |
| 649 | 49 | 100 | 0 tensile test |
| 649 | 32 | 65 | 6 |
| 649 | 30 | 61 | 36 |
| 649 | 20 | 41 | 115 |
| | | | |

The above data clearly shows how stress and temperature affect the time to failure of a test sample. The data shows that it does not take tensile stress at 100% of ultimate strength to cause failure. For example, at 649$^{o}$C steel temperature it took only 6 minutes to fail a sample even though it was only stressed to 65% of its ultimate strength. However, at 482$^{o}$C it took 244 minutes to fail a sample that was stressed to 81% of the sample ultimate strength. Now we must try to select an uncertainty factor to account for this behaviour.

With TC 128 at 649$^{o}$C the samples failed even at low stress levels (65% of ultimate). We should never let the tank steel achieve such temperatures. At 566$^{o}$C a stress at 82% of the ultimate resulted in failure after 52 minutes. This time fits well within the pool fire 100 minute requirement. Therefore, let us use this condition to draw the line. We will therefore assume that if the stress reaches 80% of the material ultimate strength the tank will fail within minutes. Based on the above it is suggested that a factor of 100/80 = 1.25 be applied to the tank burst pressure to account for stress-temperature-time to failure uncertainty.

Now, let us add up all these uncertainties:

i) fire heat flux uncertainty factor = 1.25
ii) tank pressure uncertainty factor = 1.1
iii) Tank fill uncertainty factor = 1.0
iv) failure criteria uncertainty 1.25

Overall uncertainty in the burst pressure then becomes 0.25 + 0.1 + 0.25 = 0.60, which gives an FOS = 1.6.

Let us now proceed with this as an FOS. (i.e. FOS = 1.6). If we do this then we will find that all defects will pass the torch test (as they already do with AFFTAC), but large defects will fail the pool fire test.

### *3.5 Validation*

This section shows a brief comparison between the AFFTAC and TMID computer models. The comparison is based on the case of an unprotected tank car exposed to an engulfing fire. This simulation is used as a benchmark because good data is available for this case (see Townsend et al (1974) ). In this case the tank failed at about 24 minutes in the engulfing fire.

The following inputs were used:

| | |
|---|---|
| no thermal protection | |
| propane | |
| TC 128 tank steel | |
| tank diameter | 3 m |
| tank length | 18 m |
| initial fill | 95% |
| initial temperature | 16$^{o}$C |
| fire temperature | 816$^{o}$C |
| fire emissivity | 1.0 |
| 100% engulfment | |

The PRV limited tank operating pressure was set at 2.5 MPa for TMID. In AFFTAC, the PRV std was 1.9 MPa and the PRV flow capacity was 12.3 $m^3$/s (26000 SCFM). Surface emissivity was 0.8 for AFFTAC and 0.9 for TMID.

The results are shown in Figure 3-1 to **Figure 3-10**. Five plots are shown:

1. P and P burst vs time
2. vapour space and liquid space wall T vs Time
3. volume fill vs time
4. PRV mass flow vs time
5. FOS vs time

The following can be seen in the figures.

- Both models predict shell full at about 4 minutes.
- There is a slight difference between the models for the time when the fill drops below 100% -- time is 10 minutes for AFFTAC and 13 minutes for TMID. This is probably due to differences in PRV action.
- Both models predict tank failure at about 24-25 minutes.
- Both models predicted a fill of about 45% at tank failure.
- Both models predict vapour space wall temperatures of 650$^{o}$C at about 24 minutes.
- PRV mass flow rates are of similar magnitude but the shape of the curve over time is quite different. This is due to the different PRV assumptions between AFFTAC and TMID. Note however that both have nominal flow rates of between 20 – 35 kg/s when the PRV is in full action.

The plot of FOS vs time deserves comment. The two models have very similar FOS curves. Both models predict an FOS of 2.0 at about 20 minutes into the simulation. This means that at the 20 minute mark the tanks were two times as strong as needed to survive the pressure. However, only 4-5 minutes later the FOS has dropped to 1.0 for indicated failure. In other words this FOS = 2 only provided 4-5 minutes of time before failure. This clearly shows that we must be very careful how we determine tank failure times. In this report, an FOS of 1.6 was suggested as reasonable. The TMID and AFFTAC codes predict FOS = 1.6 for an unprotected tank at about 21-22 minutes. This FOS of 1.6 only provided a 2-3 minute cushion. Clearly an FOS of 1.6 is not excessive.

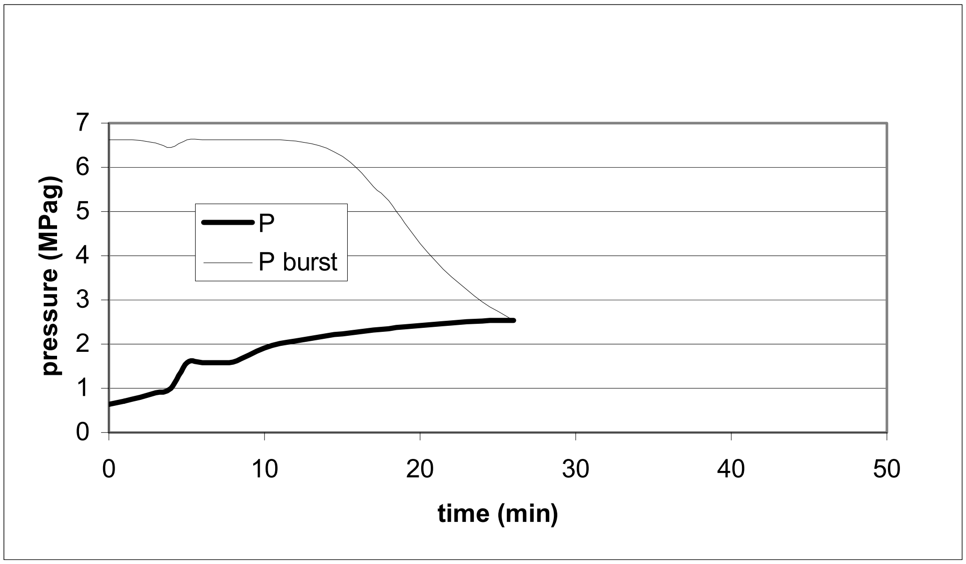



Figure 3-1: Predicted Tank Pressure vs Time (AFFTAC)

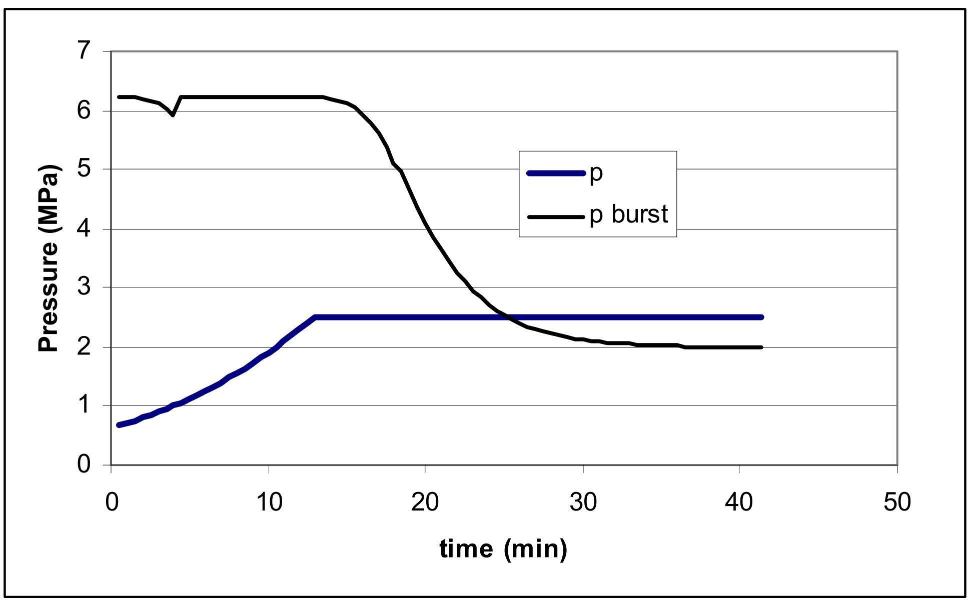



Figure 3-2: Predicted Tank Pressure vs Time (TMID)

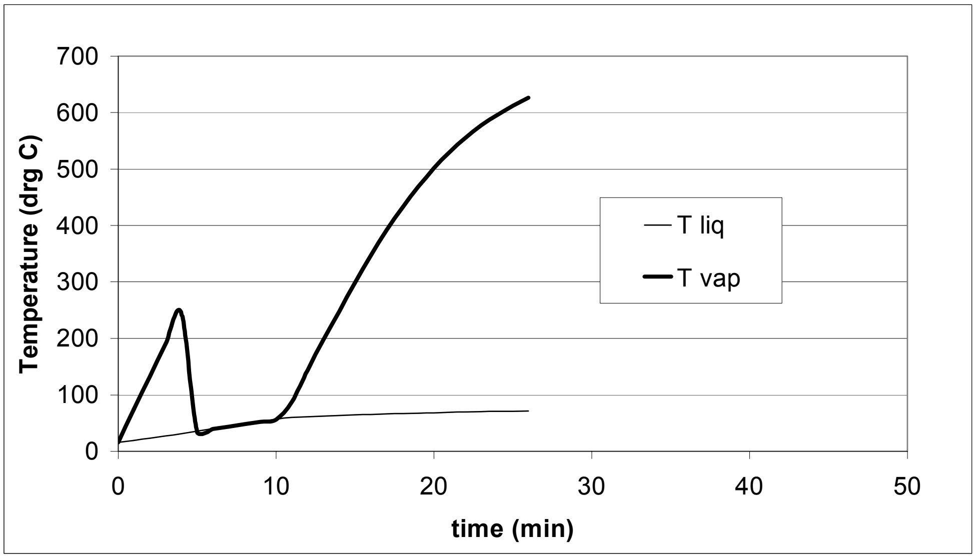



Figure 3-3: Predicted Wall Temperature vs Time (AFFTAC)

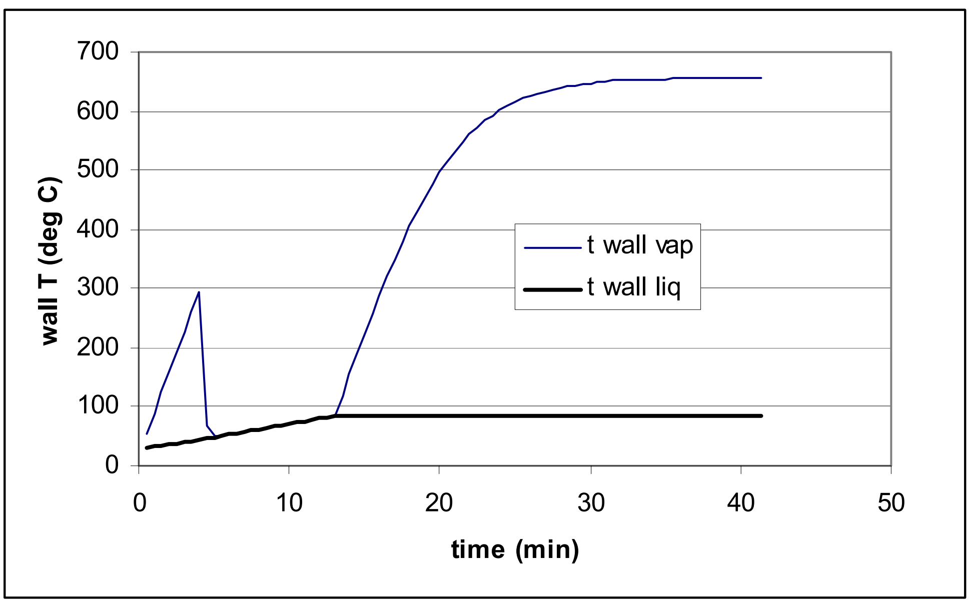



Figure 3-4: Predicted Wall Temperature vs Time (TMID)

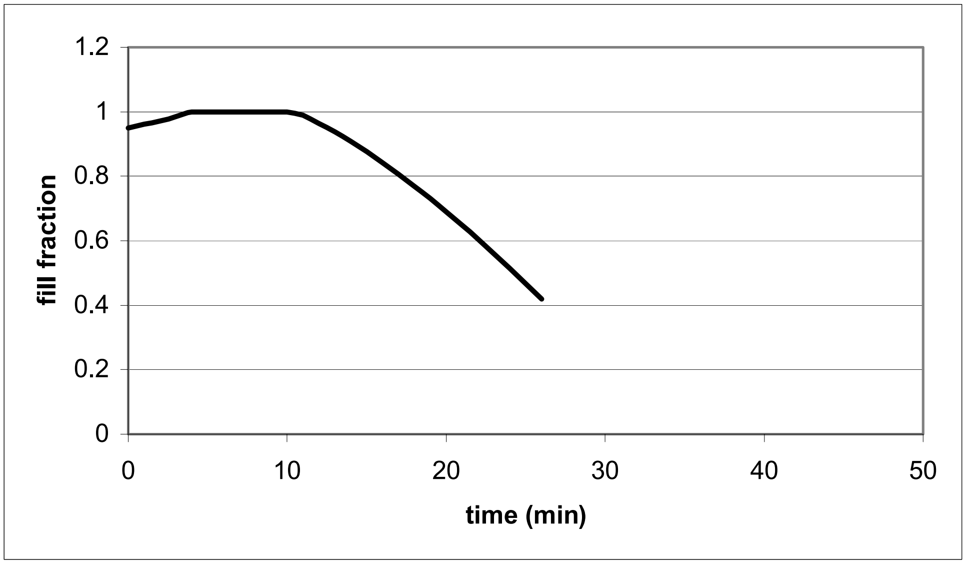



Figure 3-5: Predicted Fill vs Time (AFFTAC)

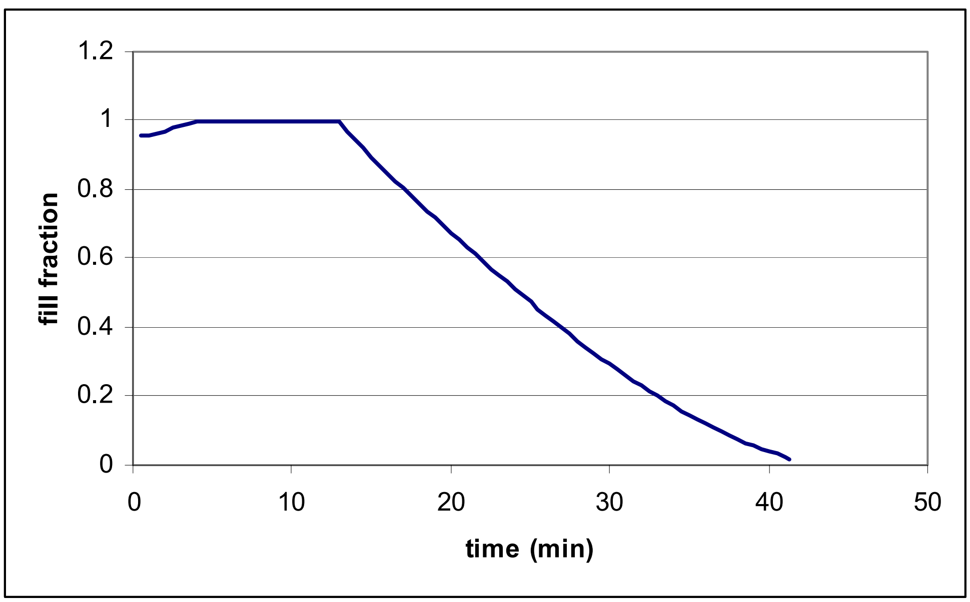



Figure 3-6: Predicted Fill vs Time (TMID)

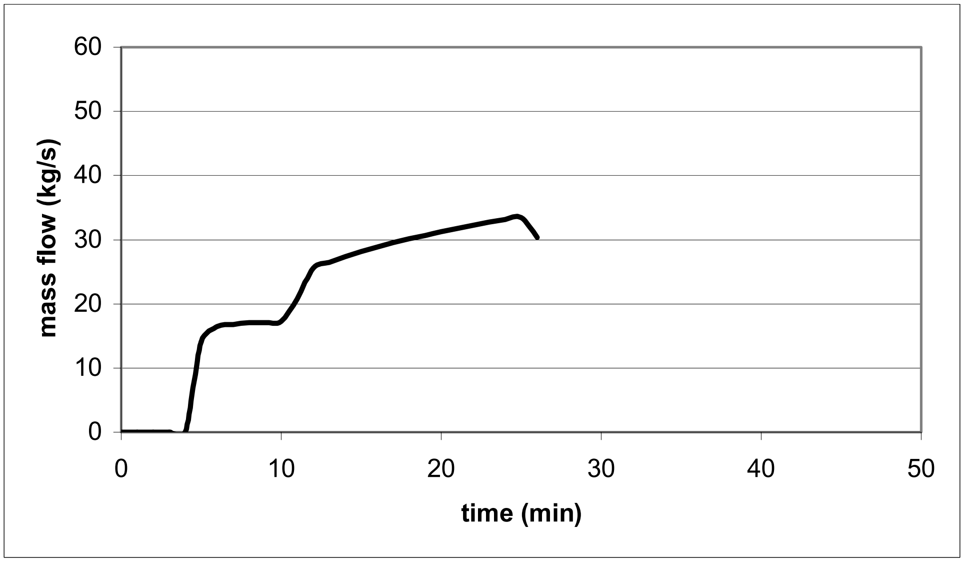



Figure 3-7 Predicted PRV Flow vs Time (AFFTAC)

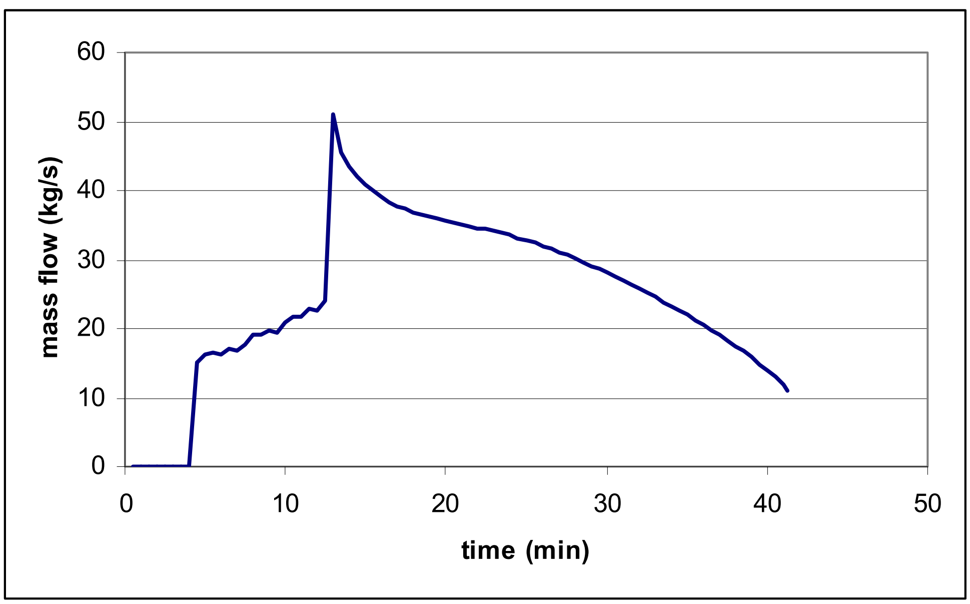



Figure 3-8: Predicted PRV Flow vs Time (TMID)

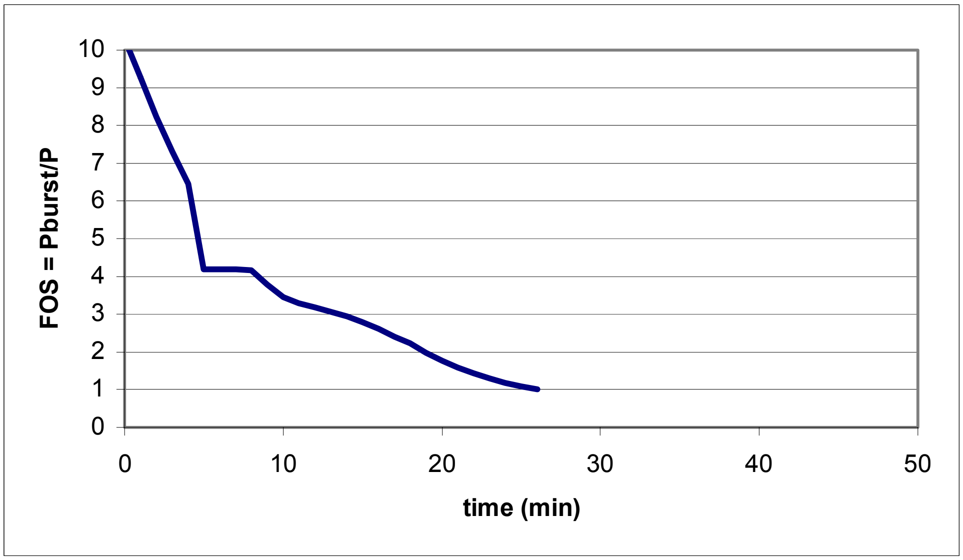



Figure 3-9: Predicted FOS vs Time (AFFTAC)

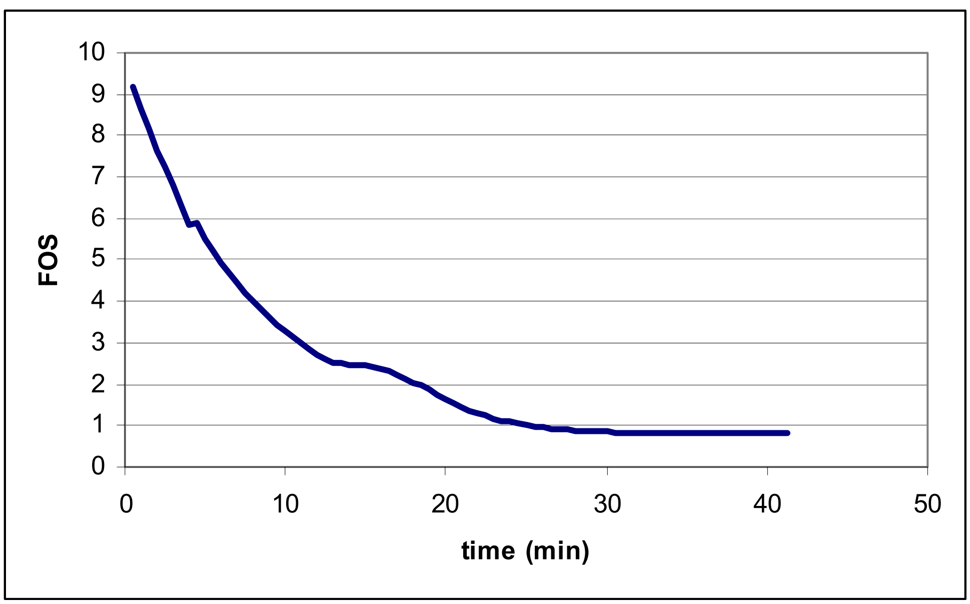



Figure 3-10: Predicted FOS vs Time (TMID)

# 4 Thermal Protection Defect Analysis Results

This section shows the results of computer simulations of tanks with thermal insulation defects exposed to pool and torch fires. Two different models were used. The first, AFFTAC, is the FRA standard computer model for evaluating thermal protection systems. The second code, TMID, was developed for this study to account for partial insulation defects.

The results about to be presented are preliminary. Further validation is necessary to confirm the accuracy of these predictions.

## *4.1 AFFTAC Simulations*

The AFFTAC model is described in detail by Johnson (1998).

The AFFTAC model cannot be used to study localized insulation defects as it currently exists. However, it can be used to study the limiting cases of a well protected tank and a thermally protected tank where all of the insulation has been lost under the steel jacket. In other words, 0% and 100% insulation defect.

### 4.1.1 0% Insulation Defect

In this case we will consider a tank with insulation that just meets the DOT plate test criteria. Johnson (1995) stated that such an insulation would have a conductance of about 4 Btu/hr ft$^2$ $^o$F (i.e. 22.7 W/m$^2$ K).

The test case has the following inputs:

- 112J340W tank car
- propane
- thermal protection with constant conductance of 22.7 W/m$^2$ K (4 Btu/hr ft$^2$ $^o$R)
- tank capacity 33000 gal
- tank diameter 119 inches
- wall thickness 0.69 inches (17.5 mm)
- 90% full of propane
- 60$^o$F initial temperature (15.6$^o$C)
- 25800 scfm PRV with start-to-discharge pressure at 280.5 psig with 308.6 psig flow rating pressure
- surface emissivity 0.9
- TC 128 steel assumed

The results of the pool fire simulation are as follows:

Table 4-1: Results of AFFTAC Simulation for Pool Fire Exposure of Thermally Protected Tank with 0% Defect (22.7 $W/m^2$ K average conductance).

| **Time (min)** | **P (psi)** | **P burst (psi)** | **T wall ($^oC$)** | **Fill (%)** |
|---|---|---|---|---|
| | | | | |
| 100 | 258 | 780 | 398 | 47 |
| 150 | 243 | 625 | 492 | 11 |
| 174 empty | 234 | 517 | 547 | 0 |
| 200 | 232 | 326 | 643 | liquid empty |
| 218 failure | 232 | 232 | 691 | liquid empty |

As can be seen from the above, the tank survives well after the 100 minute requirement. This is because the assumed insulation properties meet the plate test standard of keeping the wall temperature below 427$^o$C (800$^o$F) for 100 minutes.

The reader should note the following:

- The pressure is decreasing in the tank because as the liquid level drops the required PRV flow is decreasing and in the AFFTAC model this results in a lower tank pressure and lower hoop stress. This may not happen in reality.
- After the tank empties of liquid the vapour wall temperature rises much more rapidly resulting in tank failure at 218 minutes.

The factor of safety (based on burst pressure) in this simulation at 100 minutes is FOS = 780/258 = 3.0. In other words, even with insulation that just meets the standard, the tank was three times as strong as needed to pass the 100 minute pool fire simulation. This factor of safety is appropriate to cover some of the uncertainties of the simulation. If this tank were protected with specification ceramic blanket insulation such as Kaowool (see Appendix F) the FOS would have been even higher, in the range of 4.0 to 6.0.

### 4.1.2 100% Insulation Defect

In this case it is assumed there is no insulation under the steel jacket. Both pool fire and torch fire cases have been considered here. In AFFTAC this case is modelled by choosing a degrading insulation and specifying the time to degrade to be 0.1 minutes. With AFFTAC the model assumes a char (i.e. degraded insulation) residue conductance of 40 Btu/hr $ft^2$ $^oR$ after the insulation is gone. This, therefore, is not really the same as having no insulation in the gap between the tank wall and the jacket.

#### *4.1.2.1 Pool Fire Simulation*

With the same tank conditions as noted above (except now the insulation degrades away in 0.1 minutes), and with the tank exposed to an engulfing pool fire with T = 816°C, the tank empties of liquid in about 88.5 minutes and fails at 94.3 min. The results are summarized in the following table.

Table 4-2: Results of AFFTAC simulation for pool fire exposure of thermally protected tank with 100% defect).

| **Time (min)** | **P (psi)** | **P burst (psi)** | **T wall (°C)** | **Fill (%)** |
|---|---|---|---|---|
| | | | | |
| 88.5 (empty) | 233 | 308 | 653 | 0 |
| 94.3 (failure) | 231 | 231 | 692 | 0 |

By the letter of the standard this tank just fails the 100 minute pool fire test (it failed by about 6%). This means that if there was only 90% defect this model would result in a pass conclusion. This conclusion leaves no factor of safety.

#### *4.1.2.2 Torch Fire Simulation*

The above case was repeated but with a 1204°C torch fire applied to a 1.2 m x 1.2 m portion of the tank (as per AFFTAC assumptions). The results of this simulation are as follows:

at 30 minutes

P = 94 psig
P burst = 371 psig
FOS = 3.9
fill = 90%
T wall = 620°C (1149°F)

As can be seen, AFFTAC predicts that this tank does not even come close to failure even with 100% insulation defect (FOS = 371/94 = 3.9). The reader should note that the small torch exposure does not result in a pressure rise in this simulation. However, even if there was a pressure rise to the PRV std pressure the tank would still not fail this simulated 30 minute torch test.

### 4.1.3 AFFTAC Conclusions

Based on the AFFTAC model, a thermally protected tank (type 112J340W with 17.5 mm wall, 280.5 psig std pressure, propane, 90% full, TC 128 steel) with thermal protection (13 mm insulation with steel jacket) with an average conductance of 22.7

$W/m^2K$ will survive a 816°C engulfing pool fire for 100 minutes with a factor of safety (FOS = burst pressure/pressure) at 100 minutes of about 3.0. The same tank with all of the insulation lost under the steel jacket will fail the same simulation test because tank failure is predicted at 94 minutes into the test.

With AFFTAC, the pool fire appears to be the determining test for insulation defects. The torch fire simulation resulted in the tank being far away from failure at the 30 minute time limit.

The AFFTAC simulation results should be interpreted in the following way.

- Tanks with large areas of insulation missing under the steel jacket could come dangerously close to failure within 100 minutes when exposed to a pool fire.
- The torch fire test does not appear to be the determining factor with insulation deficiencies under a steel jacket, provided we accept the following AFFTAC modelling assumptions:

    - torch exposure is minor (<1% of tank surface)
    - torch heat flux is about 116 $kW/m^2$ (large propane torches could be 200 $kW/m^2$ or more)
    - tank pressure is well below the PRV std pressure
    - tank fill level is high (90%)

### *4.2 TMID Simulations*

The AFFTAC model suggested that a thermally protected 112 type tank with propane with 100% insulation defect (just steel jacket) would "just" fail the pool fire simulation test. Therefore by AFFTAC a tank that has slightly less than 100% defect (say 90%) could pass. However, accepting 90% defect would not be prudent as it includes no factor of safety to account for simulation uncertainties.

The questions to be answered now include:

- What happens if we apply some reasonable factor of safety?
- What amount of defect is acceptable?

The analysis about to be presented uses the concept of FOS to identify acceptable levels of insulation defect.

### 4.2.1 Torch Fire Simulation

It was shown earlier using AFFTAC that the torch test is not the determining factor for a tank with steel jacket. In this section TMID simulation results are shown for a torch fire simulation of a tank with insulation defects. The test case is a 112 car with propane as follows:

112J340W type car with thermal protection
propane
initial fill = 0.90
initial T = 15.6$^o$C
wall thickness = 17.5 mm
13 mm of insulation
PRV pop pressure P = 2.0 MPa (1.03 x 280.5 psig)
TC 128 steel
tank emissivity = 0.9
insulation defect from tank top to 90 degrees around side

In this case the insulation defect runs the entire length of the tank on one side. All protected areas have conductance 22.7 W/m$^2$ K (4 Btu/hr ft$^{2o}$F). The simulated torch is applied to a ring that goes all around the tank so that the torch heats all possible surfaces (vapour space protected and unprotected, liquid space protected and unprotected).

The TMID model was run with the following torch conditions:

temperature 1204$^o$C
heat flux factor 0.45 (torch emissivity)
exposure area 1% of tank

The following was predicted for the above tank after 30 minutes of torch exposure.

P = 0.7 MPa
P burst = 2.1 MPa
FOS = 3.0
T wall = 652$^o$C
Fill = 90%

This result is somewhat different than that from AFFTAC, but with the same conclusion. The predicted wall temperature is higher by about 35$^o$C and this results in a reduced burst pressure by about 18%. This discrepancy between AFFTAC and TMID for the torch case is due to the fact that AFFTAC assumes a conductance of 40 BTU/hr ft$^2$ $^o$R) when all the insulation is gone under the jacket. In fact, if there is no insulation the effective conductance is higher than this due to the effects of thermal radiation.

In any event this confirms that the torch fire does not appear to be the determining case (assuming 1% exposure and other AFFTAC torch assumptions). If we assume 30% exposure the following result is obtained at 30 minutes:

P = 1.07 MPa
P burst = 2.2 MPa
FOS = 2.1
T wall = 642$^{o}$C
Fill = 99%

This suggests the tank will be near liquid full at 30 minutes and therefore the rising liquid is cooling the vapour space wall. Once again the result suggests that the torch is not the limiting case. However, we should note that the predicted tank pressure of 1.07 MPa is probably incorrect because of the effects of liquid temperature stratification. If the tank did pressurize because of liquid temperature stratification then the FOS = 2.2/1.8 = 1.2

Let us repeat this run with a low liquid fill level of 80%. At 30 minutes, the outcome is:

P = 1.05 MPa
P burst = 2.1 MPa
T wall = 649$^{o}$C
Fill = 83%

Once again the tank passes the standard torch test even with 30% exposure, suggesting that this torch test will not be the determining test for insulation defects. Note that this conclusion only applies if we use the torch heat flux as defined earlier. A large torch from a burning PRV flare will be more intense than the torch used in the steel plate tests. This is because of the scale of the torch. Larger torches will behave more like an engulfing fire but with a higher flame temperature.

Let us simulate this case with the following assumptions (all other data as before):

torch temperature 1204$^{o}$C
torch emissivity 1.0 (rather than the 0.45 used earlier)
30% exposure
tank fill 90%

Note that this torch will heat the steel plate up much faster than required by the standard (i.e. 427$^{o}$C in 4 minutes). The result is that the tank fails after about 10 minutes, with the following details:

P = 0.88 Mpa
P burst = 0.88 MPa
T wall = 749 $^{o}$C

Fill = 96%

As can be seen from this example, the exact torch details determine the result. If we use the DOT standard torch then the tank with defective insulation passes the test. If we use a torch that more closely simulates a credible real-world torch event (i.e. burning PRV flare from an overturned tank) then the tank with defective insulation will probably fail the test.

**Table 4-3** summarizes the torch simulation results.

Table 4-3: Summary of TMID torch results

| | **1% Exposure 1204°C torch emm = 0.45** | **30% Exposure 1204°C torch emm = .45** | **30% Exposure 1204°C torch emm = 1.0** |
|---|---|---|---|
| Outcome | pass at 30 minutes | pass at 30 minutes | fail at 10 minutes |
| P MPa | 0.7 | 1.07 | 0.88 at fail |
| P burst MPa | 2.1 | 2.2 | 0.88 at fail |
| T wall °C | 652 | 642 | 749 at fail |
| FOS | 3.0 | 2.1 | 1.0 at fail |
| Fill | 90% | 99% | 96% at fail |

### 4.2.2 Pool Fire Simulation

It was shown earlier that the torch fire simulation test may not be the determining factor for a tank with steel jacket if we use the DOT standard torch conditions. In this section simulation results are shown for pool fire simulations of tanks with insulation defects. The test case is a 112 car with propane as follows:

112J340W type car with thermal protection
propane
diameter 3.0 m
length 17.3 m
initial fill = 0.90 (to give 99% full at 43°C)
initial T = 15.6°C
wall thickness = 17.5 mm
13 mm of insulation
PRV limited tank pressure P = 2.0 MPa (1.03 x 280.5 psig)
TC 128 steel
tank emissivity = 0.9
pool fire T = 816°C
fire emissivity 1.0
tank fire exposure 100%

In these cases the insulation defect runs the entire length of tank on one side. All protected areas have conductance 22.7 $W/m^2$ K (4 Btu/hr $ft^{2o}F$)

The first case we will consider is that of a 112 tank with a negligible defect that runs the length of the tank at the top (defect spans from 0 to 1 degree around the tank side, i.e. defect area < 0.3% of tank surface). The results of this simulation are shown in **Figure 4-1** to **Figure 4-5**.

The results show that TMID is in good agreement with AFFTAC for the case of a well insulated tank. The well insulated tank empties at about 170 minutes and by AFFTAC would fail sometime after that. In other words the well insulated tank far exceeds the pool fire simulation test. The wall temperature in the well insulated areas stays below 427 $^{o}$C for over 100 minutes and therefore passes the old plate test standard. At 427$^{o}$C the wall is still very strong and the tank is far from failure (i.e. FOS = 5). However, the small area where the insulation is defective reaches a temperature of 580$^{o}$C at 100 minutes. At this temperature the FOS is very close to the suggested limit of 1.6 and therefore the tank is very near failure based on this criterion.

Once again, the FOS plot deserves comment. As can be seen in the plot the FOS drops rapidly when the fire starts and drops to FOS = 2.0 at about 70 minutes. The FOS then drops very slowly from 2.0 to about 1.5 at 150 minutes. This shows that a small error in the prediction of FOS can lead to a very large error in time to failure prediction. This again supports the idea of applying a significant FOS.

Now let us vary the area of defective insulation to see how the tank condition varies. Recall that the defective insulation is assumed to be areas where there is no insulation material under the steel jacket. The following table summarizes the results from simulations where the defect size is varied. Once again, we use an FOS = 1.6 (recall that FOS = (burst pressure)/(tank pressure) ).

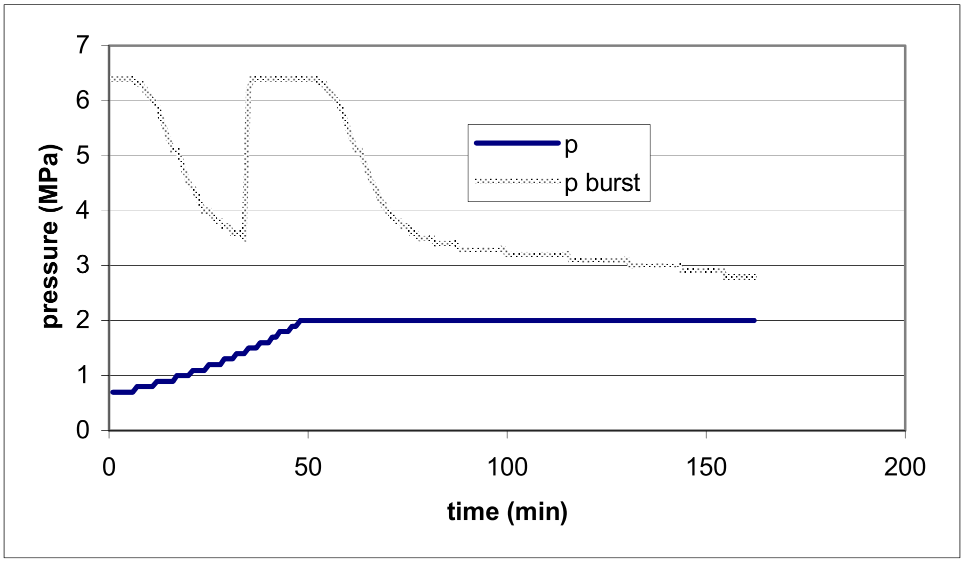



Figure 4-1: Pressure and Burst Pressure vs Time (TMID, insulated tank with small defect at tank top)

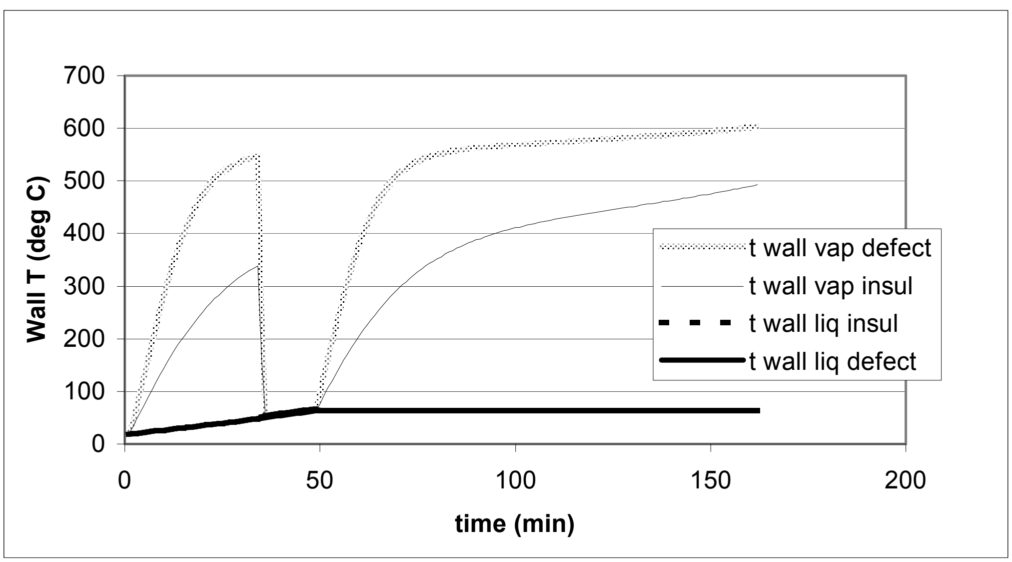



Figure 4-2: Wall Temperatures vs Time (TMID, insulated tank with small defect at tank top)

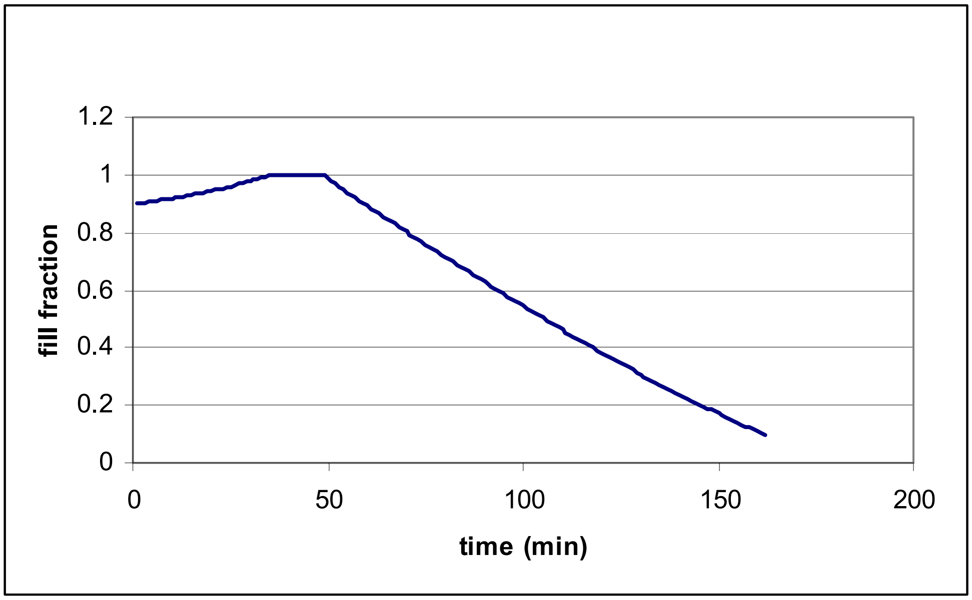



Figure 4-3: Fill vs Time (TMID, insulated tank with small defect at tank top)

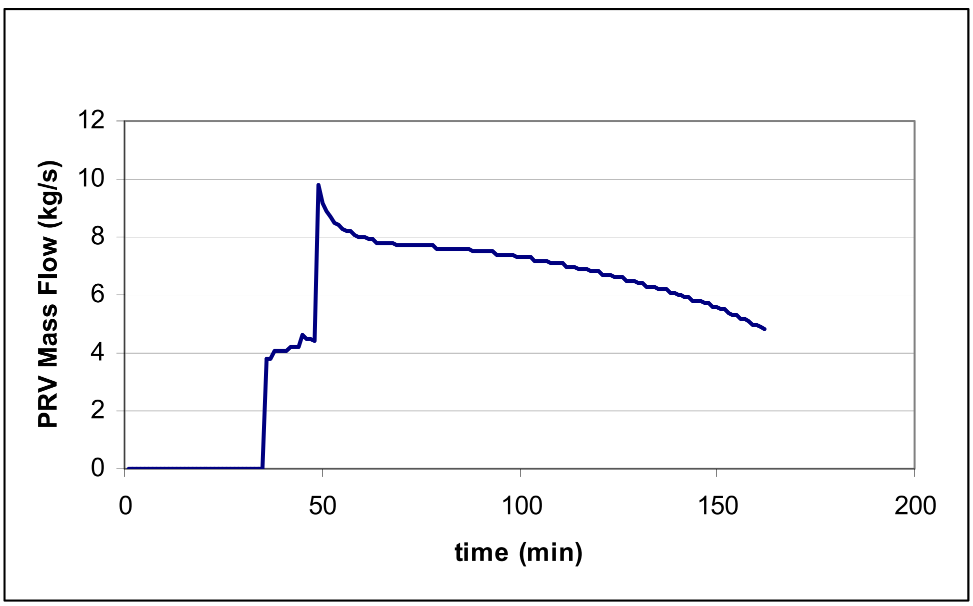



Figure 4-4: PRV Mass Flow vs Time (TMID, insulated tank with small defect at tank top)

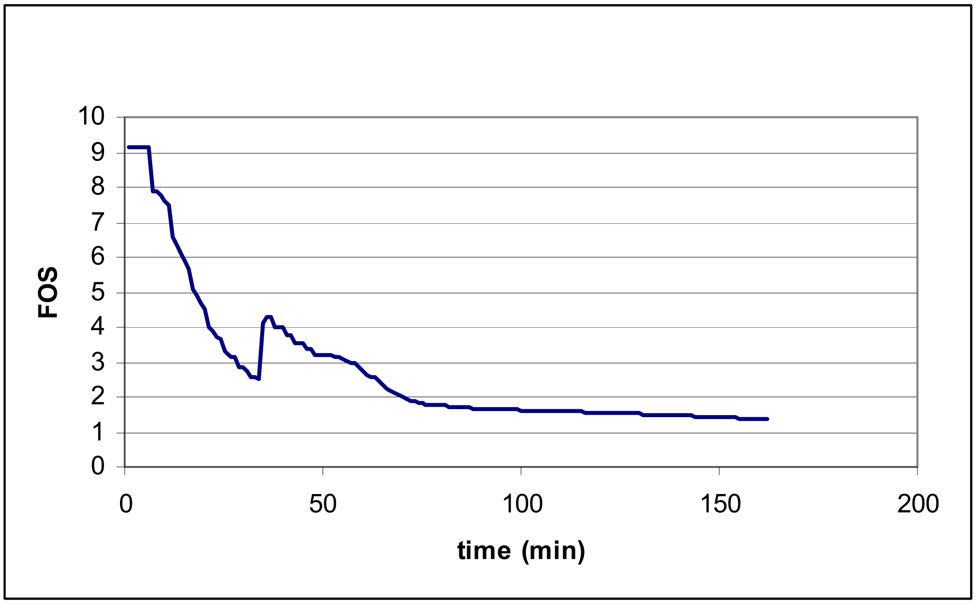



Figure 4-5: FOS based on Defect Area Temperature vs Time (TMID, insulated tank with small defect near tank top)

Table 4-4: Summary of Pool Fire Simulation Results (FOS = 1.6, insulation conductance 22.7 $W/m^2$ K)

| **Defect area** | **Fail time with FOS = 1.6** | **Wall T at fail** | **Liquid fill at fail** |
|---|---|---|---|
| from top to 10 deg<br>3% of area | 101.3 minutes | 570 | 52% |
| top to 20 deg<br>6% | 97.5 | 570 | 54% |
| top to 30 deg<br>8% | 93.6 | 570 | 56% |
| top to 90 deg<br>25% | 75 | 570 | 62% |
| top to 180 deg<br>50% | 66 | 570 | 55% |

Therefore, if we use a factor of safety of 1.6, and we assume insulation with constant conductance of 22.7 $W/m^2K$ everywhere else then we can only accept defects that cover up to about 4% of the tank surface. Recall that this conductance represents insulation that would just pass the plate test standard.

Now let us consider the case of good quality ceramic blanket insulation such as Kaowool (see Appendix F).

Table 4-5: Summary of Pool Fire Simulation Results using an FOS = 1.6 with Kaowool Properties (density 72.2 $kg/m^3$).

| **Defect area** | **Fail time with FOS = 1.6** | **Wall T at fail** | **Liquid fill at fail** |
|---|---|---|---|
| from top to 10 deg<br>3% of area | 184 minutes | 570 | 48% |
| top to 60 deg<br>17% | 115 | 570 | 54% |
| top to 80 deg<br>22% | 102 | 570 | 68% |
| top to 90 deg<br>25% | 98.4 | 570 | 67% |
| top to 100 deg<br>28% | 95 | 570 | 66% |
| top to 180 deg<br>50% | 77 | 570 | 57% |

If we use a factor of safety of 1.6, and we assume 13 mm of Kaowool ceramic blanket insulation everywhere else then we can accept defects that cover up to about 23% of the tank surface.

### 4.2.3 Sensitivity Study Results

Let us now study how the previous results are affected by changes in important variables. We want to know how the results are affected by:

i) effect of FOS
ii) initial fill
iii) defect position
iv) PRV limited tank pressure
v) tank surface emissivity
vi) commodity

i) Effect of FOS

First let us look at the results if we use FOS = 1.0. **Table 4-6** gives a summary of results with the baseline case (propane, D = 3m, L = 17.4 m, 90% full, 2 MPa PRV, insulation conductance 22.7 W/m$^2$K, etc.).

Table 4-6: Summary for FOS = 1.0

| **Defect Top (deg)** | **Defect Bottom (deg)** | **Fail/empty Time (min) (FOS = 1)** | **FOS at Time** | **Defect Vapour Wall T ($^o$C)** | **Liquid T ($^o$C)** | **Tank Fill** | **Lading Mass (kg)** |
|---|---|---|---|---|---|---|---|
| 0 | 0 | 173 empty | - | - | 60 | .02 | 6900 |
| 0 | 30 | 160 empty | 1.19 | 631 | 60 | .02 | 6900 |
| 0 | 60 | 150 empty | 1.13 | 641 | 60 | .02 | 6900 |
| 0 | 90 | 140 empty | 1.06 | 651 | 60 | .02 | 6900 |
| 0 | 120 | 129 fail | 1 | 659 | 60 | .033 | 7154 |
| 0 | 180 | 109 fail | 1 | 659 | 60 | .029 | 7360 |
| | | | | | | | |

The simulation suggests that even with 50% defect the tank will survive the 100 minute simulation if FOS = 1.0 is applied. With smaller defects the tanks go empty before failure is indicated. Note that TMID stops the simulation when the tank has 2% liquid by volume (i.e. liquid empty). As can be seen from the table, when the tank is empty it is dangerously near failure.

Now let us vary the FOS from 1.2 to 1.7 and find the defect area needed for failure at 100 minutes. The results are shown in **Table 4-7**. Note that these results apply to the case of insulation with constant conductance of 22.7 W/m$^2$K.

Table 4-7: Summary of Pool Fire Simulation Results using Varying FOS, (112 tank, propane, 2 MPa PRV, 17.5 mm wall, 90% full, 16 C initial T)

| **FOS** | **Allowable Defect Area in % of Tank Surface** | **Fail Time with FOS** | **Wall T at Fail** | **Fill at Fail** |
|---|---|---|---|---|
| 1.7 | 0% | 93 minutes | 542 | 62% |
| 1.6 | 4% | 100 | 570 | 53% |
| 1.5 | 17% | 100 | 585 | 43% |
| 1.4 | 28% | 100 | 600 | 36% |
| 1.3 | 42% | 100 | 615 | 22% |
| 1.2 | 53% | 100 | 630 | 12% |

The selected FOS has a dramatic effect on the allowable defect area. As noted earlier, it is suggested that FOS = 1.6 based on fire, PRV and material strength considerations. If, for example, the fire variability is ignored then an FOS of 1.4 is reasonable and this would allow up to 28% defective insulation.

ii) initial fill

Let us fix the defect at 0-90 deg (25% defect) and vary the tank initial fill from 90% to 20%. All other factors remain the same including FOS = 1.6.

Table 4-8: Summary of Pool Fire Simulation Results Using an FOS = 1.6, with Tank Initial Fill Varied from 20 to 90%. (112J340W tank, propane, 816$^{o}$C pool fire).

| **Initial Fill** | **Fail Time with FOS = 1.6** | **Wall T at Fail** | **Liquid Fill at Fail** |
|---|---|---|---|
| 90% | 75 min | 572$^{o}$C | 62% |
| 80% | 56 | 572 | 76 |
| 50% | 38 | 578 | 56 |
| 20% | 34 | 579 | 17 |

Fill plays an important roll in the time to failure. This is because of the heat transfer assumptions used in the model. AFFTAC assumes the liquid surface acts as a heat sink to cool the vapour space wall. Low fill levels result in less cooling of the vapour space wall and therefore higher wall temperatures are predicted.

The following table shows the effect of varying defect area when the tank is initially filled to 80%.

Table 4-9: Summary of Pool Fire Simulation Results using an Initial Fill of 80% with Constant Insulation Conductance of 22.7 W/m$^2$K.

| **Defect Area** | **Fail Time with FOS = 1.6** | **Wall T at Fail** | **Liquid Fill at Fail** |
|---|---|---|---|
| from top to 10 deg<br>3% of area | 86 minutes | 570 | 60% |
| top to 60 deg<br>17% | 58 | 570 | 80% |
| top to 90 deg<br>25% | 54 | 570 | 79% |

As can be seen, no defect is allowed with this case.

iii) defect position

Let us now hold the size of the defect the same but move it around the side of the tank. The FOS = 1.6 and all other factors are held constant (fill = 90%).

Table 4-10: Summary of Pool Fire Simulation Results using an FOS = 1.6, with Position of Defect Varied Around Side of Tank (112J340W tank, propane, 816°C pool fire)

| **Defect Area** | **Fail Time with FOS = 1.6** | **Wall T at Fail** | **Liquid Fill at Fail** |
|---|---|---|---|
| from top to 90 deg | 75 min | 572°C | 61% |
| from 20 to 110 deg | 77 | 572 | 58 |
| from 40 to 130 deg | 79 | 572 | 54 |
| from 60 to 150 deg | 85 | 572 | 46 |
| from 80 to 170 deg | 104 | 572 | 27 |

From above we see that the worst defect position is near the tank top. This is what we would expect. If the defect is near the tank bottom we can increase the allowable defect to over 25% according to this simulation.

The problem with applying this conclusion is that a tank may roll over in an accident. Therefore if a defect is near the bottom of a tank it could be as dangerous as one near the tank top.

iv) PRV limited tank pressure

In TMID the tank pressure is assumed to be held at 103% of the PRV start to discharge pressure by the PRV. So far this pressure has been set to 2.0 MPa. The following results apply for the case where the PRV controls the tank pressure to be 1.8 MPa (263 psig). As before the other tank conditions are:

112J340W type car with thermal protection
propane
diameter 3.0 m
length 17.3 m
initial fill = 0.90 (to give 99% full at 43°C)
initial T = 15.6°C
wall thickness = 17.5 mm
13 mm of insulation
TC 128 steel
tank emissivity = 0.9
pool fire T = 816°C
fire emissivity 1.0
tank fire exposure 100%

Table 4-11: Summary of Pool Fire Simulation Results using an FOS = 1.6 with Reduced PRV Limited Tank Pressure (P = 1.8 MPa).

| **Defect Area** | **Fail Time with FOS = 1.6** | **Wall T at Fail** | **Liquid Fill at Fail** |
|---|---|---|---|
| from top to 10 deg<br>3% of area | 146 minutes | 594 | 19% |
| top to 20 deg<br>6% | 141 | 594 | 21% |
| top to 60 deg<br>17% | 115 | 594 | 31% |
| top to 80 deg<br>22% | 101 | 594 | 38% |
| top to 90 deg<br>25% | 95 | 594 | 41% |

As can be seen a reduced tank pressure has a very significant effect on time to failure. A lower pressure means a lower stress and this means a higher wall temperature is needed to fail the tank. A higher wall temperature requires a lower tank fill condition, which takes more time.

This small change in tank pressure increased the allowable defect from 4% to about 23% for the assumed conditions.

v) tank surface absorptivity

So far, all computer runs have assumed a tank surface emissivity/absorptivity of 0.9. As noted earlier in this report this value of emissivity is considered appropriate for the materials considered here. This is in contrast to AFFTAC, that recommends the use of 0.8. The following results show what happens when emissivity is changed from 0.9 to 0.8. All other factors are as before.

As can be seen the reduced emissivity causes a lower heating rate by the fire and this increases the time to tank predicted failure. With this change the allowable defect changes from 4% to about 15%.

Table 4-12: Summary of Pool Fire Simulation Results Using an FOS = 1.6 with Reduced Tank Surface Emissivity of 0.8.

| **Defect Area** | **Fail Time with FOS = 1.6** | **Wall T at Fail** | **Liquid Fill at Fail** |
|---|---|---|---|
| from top to 10 deg<br>3% of area | 117 minutes | 570 | 41% |
| top to 20 deg<br>6% | 114 | 570 | 43% |
| top to 50 deg<br>14% | 101 | 570 | 50% |
| top to 90 deg<br>25% | 87 | 570 | 55% |

It should be noted that surface emissivity plays a major role because of its effect on the radiation between the tank wall and the steel jacket when the insulation is missing. The heat flux between the jacket and the wall is calculated from the following:

$$q = \frac{\varepsilon}{2-\varepsilon}\sigma\left(T_j^4 - T_w^4\right)$$

where,

q = heat flux W/m$^2$
$T_w$= wall temperature
$T_j$ = jacket temperature
= surface emissivity
= Stefan-Boltzman constant

With the above equation the surface emissivity has a strong effect on q.

vi) commodity

So far all computer runs have been conducted for thermally protected 112J340W tank-cars with pure propane as the commodity. Other commodities of interest include:

i) anhydrous ammonia for 105 and 112 type cars
ii) chlorine for 105 type cars

The different commodities will affect the following:

i) pressurization rate
ii) thermal expansion of liquid
iii) PRV flow and time to empty

The specific heat of the lading determines how quickly it will heat up. The latent heat of vaporization determines how quickly the tank will empty through the PRV. The worst case is where the lading empties quickly from the tank thereby exposing more wall to the vapour space.

**Table 4-13** shows some examples of commodities and their properties. In this study we were asked specifically to consider propane, chlorine and anhydrous ammonia. As can be seen from the table ammonia has a much higher heat of vaporization than propane and therefore will boil off much more slowly than propane. Chlorine is much more dense than propane and this too increases the time for the chlorine to leave the tank through the PRV.

Table 4-13: Some Data on Different Commodities

| **Commodity** | **Tank Car Type** | **Density (kg/m$^3$)** | **Specific Heat of Liquid $C_p$ (kJ/kg K)** | **Heat of Vapor-ization $h_{fg}$ (kJ/kg)** | **critical pressure (bar)** | **critical temp-erature (K)** |
|---|---|---|---|---|---|---|
| | | | | | | |
| propane | 105, 112 | 506 | 3.1 at 50$^o$C | 428 | 42.7 | 370 |
| ammonia | 105, 112 | 681 | 4.8 at 25$^o$C | 1357 | 112.8 | 406 |
| chlorine | 105 | 1400 | | 288 | 77 | 417 |
| propylene | 105, 112 | 520 | | 437 | 46.2 | 365 |
| n-butane | 105, 112 | 582 | 2.3 at 0$^o$C | 385 | 38 | 425 |
| ethylene | 113 | 567 | | 484 | 51.2 | 283 |
| ethylene oxide | 105, 111 | 870 | 2.0 at 20$^o$C | 579 | 71.9 | 469 |

Table 4-14 shows AFFTAC simulation results for propane and anhydrous ammonia in 112J340W type tanks (AFFTAC does not consider chlorine). The ammonia is released from the tank much more slowly due to its higher density, specific heat and heat of vaporization. Therefore this shows that propane is the design worst case for thermally protected 112 type tanks.

The 105 cars that carry chlorine and anhydrous ammonia are insulated tanks that must also meet thermal protection requirements. The conductance required to meet the thermal insulation standard is much smaller then that required for thermally protected tanks. Therefore, the thermally insulated tank should pass the torch and pool fire simulation with more allowable defect than the thermally protected tank. This is true only if the thermal insulation used in the tank can withstand the high fire temperatures for the required length of time. It also assumes that the tank PRV can handle the increased flow caused by the insulation defects. It should be noted that tanks with thermal insulation may have relatively small PRVs if they are sized using the AAR sizing formula. If a thermally insulated tank with a small PRV has large insulation defects then this could lead to excessive pressure buildup in a fire accident. This has not been investigated in detail in this report.

Table 4-14: Comparison of Different Commodities with AFFTAC (112 tank, 17.5 mm wall, 90% initial fill, initial temperature 16$^{o}$C, no insulation under jacket – i.e. 100% defect)

| **Commodity** | **Liquid Empty Time** | **Fail Time** | **Burst P at 100 min** | **P at 100 min** | **FOS at 100 min** | **Wall T at 100 min** | **Fill at 100 min** |
|---|---|---|---|---|---|---|---|
| | | | | | | | |
| propane | 90 | 97 | 210 | 210 | < 1 | 708$^{o}$C | liquid empty |
| anhydrous ammonia | not empty after 200 min | no fail | 541 psi | 267 psi | 2.03 | 534$^{o}$C | 0.74 |

### 4.2.4 Special Considerations

It should be noted that the wall temperature and its chemical effects on the commodities have not been studied in detail.

For example, it is known that if chlorine comes in contact with steel at above 215$^{o}$C, intense local heating will take place and the steel can ignite (see TIPS manual for chlorine, Environment Canada, 1984). This fact alone suggests insulation defects should not be allowed on 105 cars that carry chlorine.

Insulated 105 cars include 102 mm of insulation (51 mm fibre glass and 51 mm of ceramic blanket) covered by a steel jacket. This insulation system provides excellent protection from fires and by TMID estimates it will keep the wall temperature below 215$^{o}$C in the vapour space for over 100 minutes. However, if there is a significant insulation defect in the vapour space the wall temperature there could exceed 215$^{o}$C in under 10 minutes (see Appendix E for simulation results). This suggests that chlorine cars may not tolerate any insulation defects.

### 4.2.5 Summary of Sensitivity Study

The following table summarizes the results of the sensitivity study.

Table 4-15: Summary of Sensitivity Study

| **Variable Changed** | **Change** | **Effect on Allowable Defect** | **Comment** |
|---|---|---|---|
| insulation properties | change from constant conductance of 22.7 $W/m^2K$ to actual Kaowool properties | increase from 4% to 23% | real-world ceramic blanket insulation probably has properties in this range |
| FOS | change FOS from 1.6 to 1.4 | increase from 4% to 28% | |
| initial fill | change initial fill from 90% to 80% | decrease from 4% to 0% | real world fill levels are in the range of 80-90% |
| defect position | move defect from tank top to tank bottom | increase from 4% to 25% | not relevant because tank can roll over in an accident |
| PRV limited tank pressure | change from 2.0 MPa to 1.8 Mpa | increase from 4% to 22% | |
| surface emissivity | 0.9 to 0.8 | increase from 4% to 15% | real-world tank surfaces are probably around 0.9 |
| commodity | propane to ammonia | increase from 4% to 50% | propane appears to be worst case due to its rapid release rate |

# 5 Analysis of Thermal Insulation Systems with Defects

Thermal insulation systems are typically installed to insulate the commodity from normal ambient conditions. These systems may also be needed for thermal protection from fires.

## 5.1 Thermal Insulation Systems

In general if a pressure tank-car requires thermal it must meet the following requirement.

*If insulation is a specification requirement, it shall be of sufficient thickness so that the thermal conductance at 15.5 C (60 F) is not more than 1.533 kJ/h m$^2$ C (0.075 Btu/h ft$^2$ F) temperature differential. If exterior heaters are attached to the tank, the thickness of the insulation over each heater element may be reduced to one-half that required for the shell.*

For tanks that need both thermal insulation and thermal protection, it can be shown that this requirement of conductance of 0.075 Btu /hr ft$^{2o}$F far exceeds (by about a factor of 50 if we use 4 Btu/hr ft$^{2o}$F as the limit to pass the thermal protection requirement) that needed for the thermal protection requirement. However, for the insulation to pass the thermal protection requirement it must be a high temperature insulation. Common insulation systems include 5 cm of fibreglass insulation (low temperature) and 5 cm of ceramic blanket (high temperature insulation).

If an insulated tank must also be thermally protected, then insulation defects on such tanks will be governed by the limitations specified in the chapter on thermal protection systems.

## 5.2 Insulation Conductance

For the case of thermally insulated tanks a model was developed to calculate the overall average conductance for the tank with and without insulation defects. The model considered the various layers of insulation and the effects of internal and external convective heat transfer. It was assumed that if there was an insulation defect, there would be no insulation present and that stagnant air will fill the insulation defect space. The model included the following details:

- convection and thermal radiation from ambient temperature to outer surface of steel jacket
- conduction through jacket
- conduction through insulation (for no defect) or, conduction and thermal radiation through air space (for defect)

- conduction through tank wall
- convection to liquid

Insulation discontinuities have been discussed earlier in this report.

For pressure cars with thermal insulation the requirement is that the overall conductance cannot exceed 1.533 kJ/hr $m^{2o}C$ (or 0.426 $W/m^2K$). If we ignore the external convective heat transfer coefficient and assume 51 mm of rockwool and 51 mm fibre glass insulation then this would result in a conductance of about 0.39 $W/m^2$ K which betters the requirement by about 8%.

If we assume the tank is 90% full then the inside convective coefficient will be high and can be ignored in the conductance calculation. It is important to note that if there is an insulation defect, there will be air in the defect space and this too will act as a good insulator provided the space is closed. However with the thermal insulation gone, thermal radiation from the jacket to the tank wall is possible and this increases the overall conductance. The model was used to estimate the overall conductance for a tank with an insulation defect under the steel jacket. Table 5-1 shows the results of this analysis.

Table 5-1: Estimates of Insulation and Defect Conductance for Various Heat Transfer and Temperature Conditions.

| | **Normal System Conductance $W/m^2$ K** | **Defective System Conductance $W/m^2$ K** | **Required** |
|---|---|---|---|
| h = 1 $W/m^2K$<br>T diff = 1 K | 0.37 | 2.7 | 0.426 |
| h = 5<br>T diff = 1 | 0.38 | 3.3 | 0.426 |
| h = 50<br>T diff = 1 | 0.39 | 4.5 | 0.426 |
| h = 5000<br>T diff = 1 | 0.39 | 4.9 | 0.426 |
| h = 5<br>T diff = 800 | 0.39 | 43.4 | 0.426 |

As can be seen from the table, the external convection does not change the insulation conductance very much but it does change the defect conductance significantly. The assumed temperature difference is also important because of the thermal radiation link when the insulation is defective.

**Table 5-2** shows that very little defect (< 2%) is acceptable to maintain the 0.426 $W/m^2K$ overall conductance requirement.

Table 5-2: Estimates of Overall Conductance vs Defect Size and External Convective Coefficient (assuming 1 K temperature difference between surroundings and liquid).

| **defect as % of total tank area** | **overall conductance $h = 5\ W/m^2\ K$** | **overall conductance $h = 50\ W/m^2\ K$** | **required** |
|---|---|---|---|
| 0 | 0.38 | 0.39 | 0.426 |
| 1 | 0.41 | 0.43 | 0.426 |
| 2 | 0.44 | 0.47 | 0.426 |
| 5 | 0.52 | 0.60 | 0.426 |
| 10 | 0.67 | 0.80 | 0.426 |
| 50 | 1.9 | 1.2 | 0.426 |

Note: h value shown is for tank outer surface, thermal radiation is accounted for. No discontinuities (i.e. insulation spacers, structures, etc.) accounted for.

# 6 Detailed Analysis of Insulation Defects

The previous sections showed how insulation defects affect the overall response of the tank-car system to fire impingement or to ambient conditions. This section will now look at the defects in a more detailed way. In this chapter we will consider how defect size affects stress and temperature when a tank is exposed to engulfing fire.

## 6.1 Local Heat Transfer

**The local heat transfer analysis was done to see how the size of the defect affects the temperature distribution in the defect.**

**Figure 6-1** shows the system being analysed. We would expect that as the defect gets smaller its temperature would be reduced by heat conduction to the surrounding protected material. Details of the local heat transfer model can be found in Appendix H.

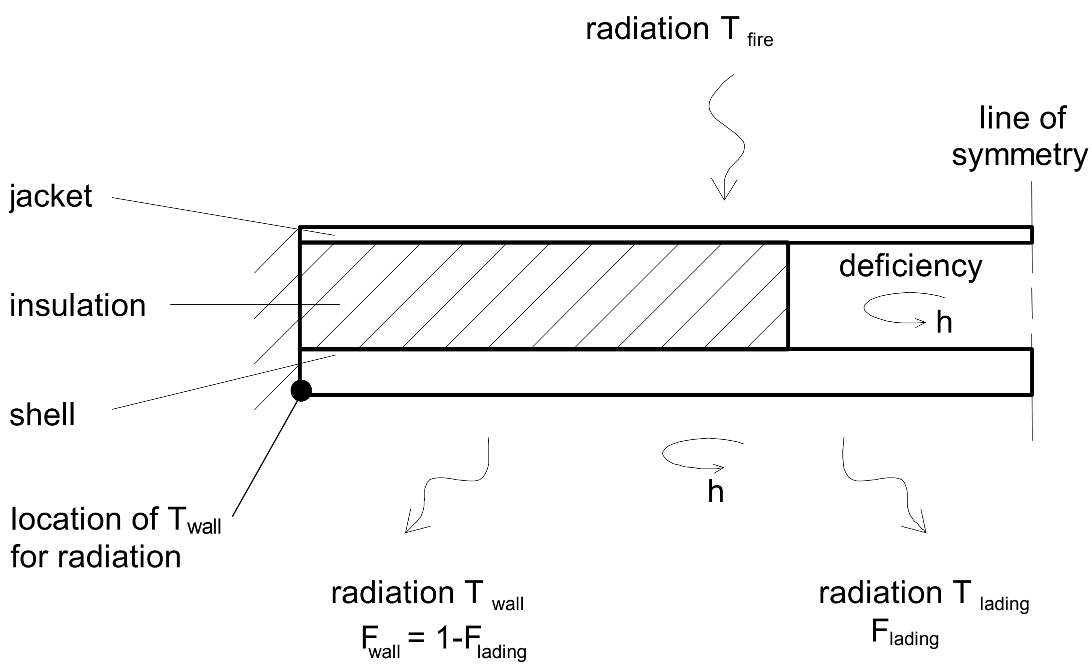



Figure 6-1: Sketch of System in Local heat Transfer Analysis

### 6.1.1 Effect of Defect Size on Defect Temperature Distribution

**Figure 6-3** shows how defect size affects the defect maximum temperature. As can be seen from the Figure, the defect size is not important for defects larger than about 0.5 m across for a long strip defect or about a 1 m across if it is an isolated block. In other words, any defect of this size or larger is considered to be “large”. As the defect gets smaller than this the temperature decreases because of the cooling effect of the surrounding protected wall.

These results suggest that as the defect gets smaller than about 0.2 m across for a long strip defect, or about 0.4 m across for a isolated block defect, the safety margin increases significantly. Therefore defects smaller than this would be considered “small” and therefore a tank covered with a few “small” defects separated by protected material is safer than a tank with one “large” defect.

The next question is: how close together do defects have to be for them to effectively act as a single large defect? This is summarized in **Figure 6-2**.

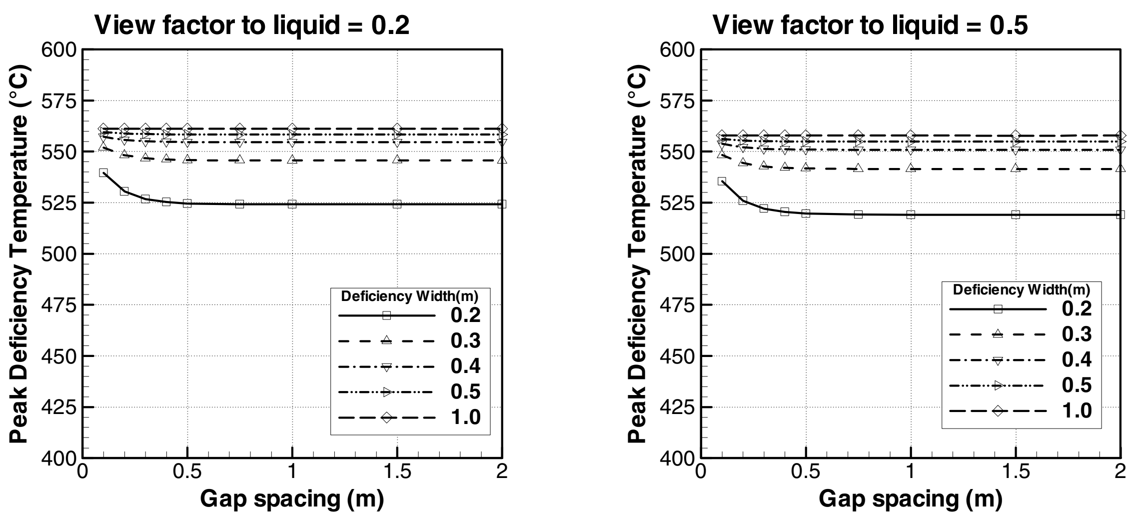



Figure 6-2: Effect of Deficiency Spacing (13 mm insulation)

As can be seen in the Figure, small defects (< 0.2 m for strip defect) should be at least 0.5 m apart to be viewed as separate defects.

## 6.2 Local Stress

As with the heat transfer, the stress in the defect will be affected by the size of the defect fore certain defect shapes. If the defect is very small then as the wall area temperature increases it will try to expand. The surrounding cool material will not allow it to expand and as a result compressive stresses will grow to reduce or offset the tensile stresses in the defect wall area. In other words the small defect is offloaded by heating. As the defect gets larger this effect will diminish. As the defect area is offloaded the surrounding protected material takes up the load. It should be noted that the hoop stress will not be offloaded if the defect runs the full length of the tank or the longitudinal stress will not be reduced if the defect extends all the way around the tank circumference.

### 6.2.1 Effect of Defect Size on Defect Stress

A preliminary finite element analysis was conducted to estimate the effects of defect size on the stress distribution in a tank wall. A finite element model was constructed to estimate the stresses in defects (i.e. hot patches in the vapour space wall) surrounded by cooler protected material. Further details of this model can be found in Appendix H.

It is expected that small defects will have reduced stress because the surrounding material will take over the load. This is only true for small block defects. Defects that run the entire length of the tank or the entire circumference will not offload their stress to surrounding material.

**Figure 6-4** shows normalized stress (stress at defect centre/ hoop stress in remainder of tank) vs defect temperature and size for isolated block defects. As can be seen the stress is reduced for defect temperatures above about 500$^{o}$C. The size of defect does not appear to be important for the size of defects considered (0.2 to 1 m). This stress reduction is not expected to take place in large defects.

This stress applies for a small "block" defect that does not run the length of the tank. For a defect that runs the entire tank length the stress is not reduced because the defect is not surrounded by protected material.

This result suggests that small defects may not be critical. However, this is based on a preliminary analysis and should be used with caution.

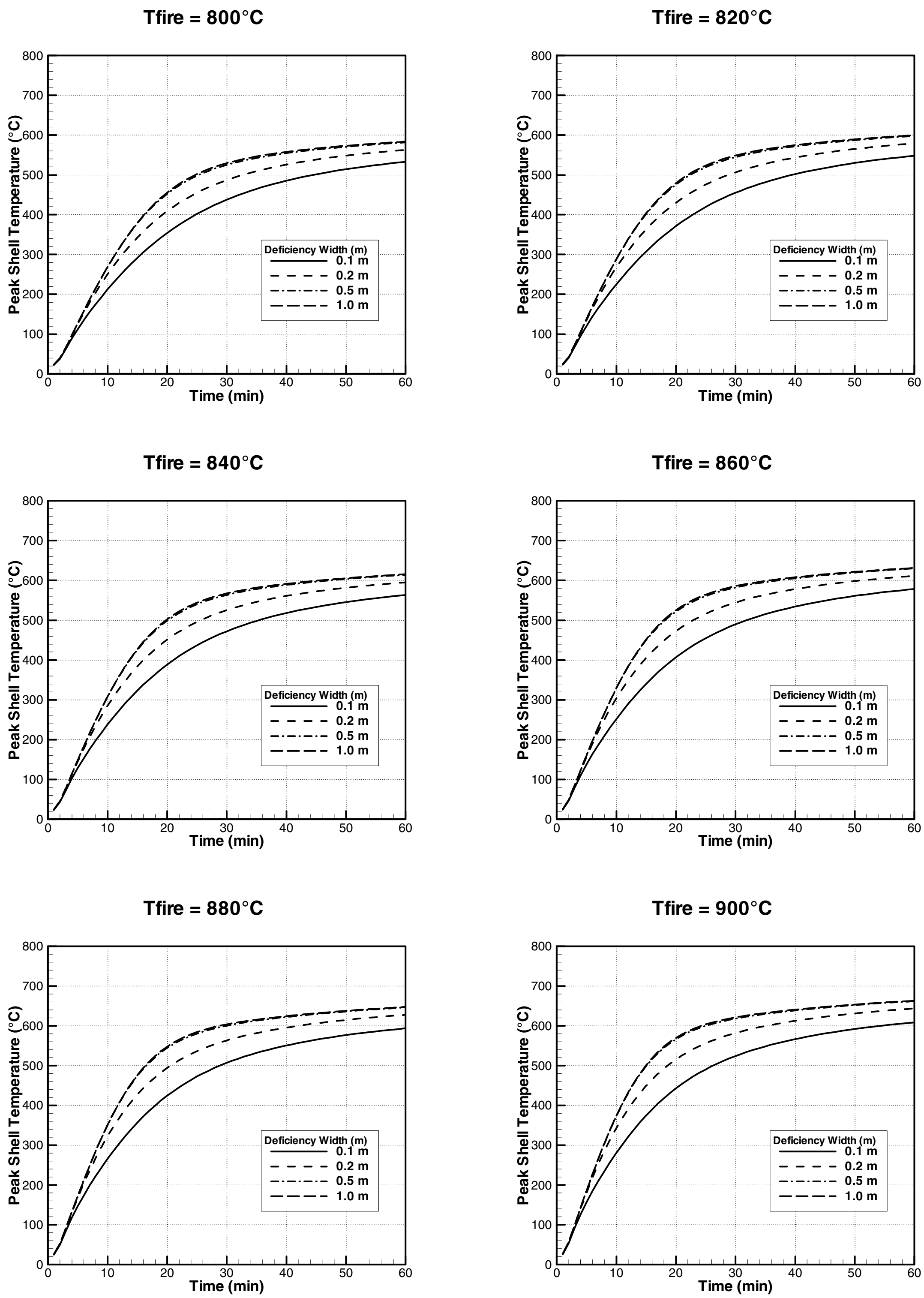



Figure 6-3: Tank Wall Temperature under Defect (13 mm insulation, view factor to liquid F = 0.2)

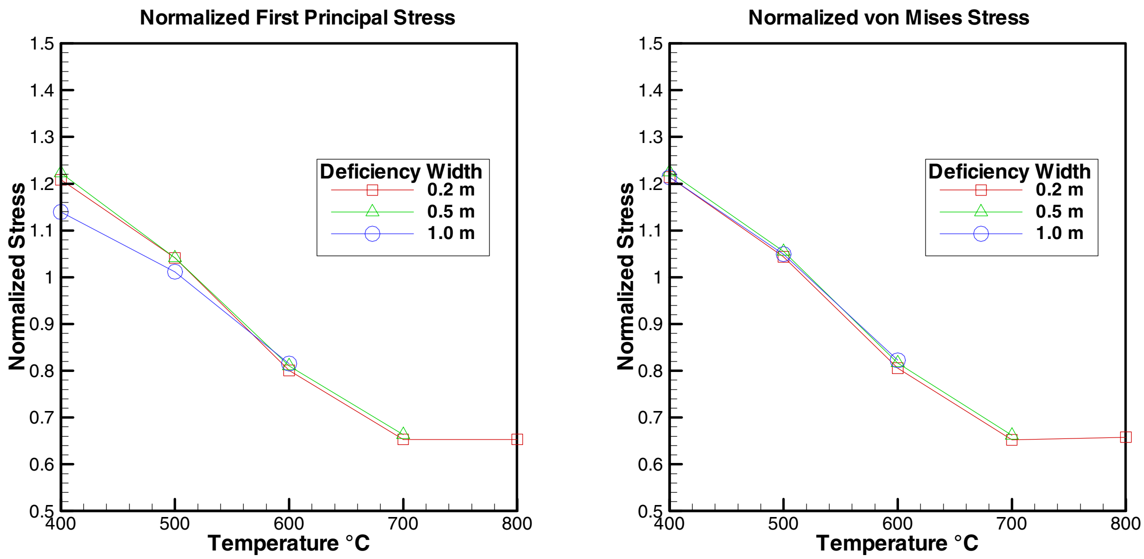



Figure 6-4: Normalized Stress in Centre of Defect Hot Spot for Various Hot Spot Temperatures and Sizes.

# 7 Conclusions and Recommendations

Based on the analysis the following conclusions are made.

## *7.1 Thermally Protected Tanks*

A computer model was developed to model a tank-car with insulation defects exposed to engulfing and torch type fires. This model, called TMID (Tank Model with Insulation Defects), is based for the most part on assumptions similar to the FRA AFFTAC 3.0 computer code. It should be noted that this report suggests that the methods used by AFFTAC may not be conservative and therefore there is some uncertainty in applying this type of model to this study.

The TMID model is two-dimensional in that it models the tank as a long cylinder. Insulation defects are modelled as regions where there is no insulation between the steel jacket and the tank wall. Defects are assumed to run the entire tank length on one side of the tank. Defect position is specified by giving the angle from the tank top to the defect top and bottom.

The baseline case considered the following type of tank:

112J340W tank
propane
tank diameter = 3 m, length = 17.3 m
wall thickness = 17.5 mm
insulation conductance 22.7 $W/m^2K$
insulation thickness 13 mm
initial fill 90%
initial temperature 15.6$^oC$
PRV assumed to limit tank pressure to a maximum of 1.03 x start to discharge pressure
PRV start to discharge pressure 1.93 MPa (280.5 psig)
PRV capacity assumed large enough to maintain assumed pressure
tank material TC 128 steel

Based on the above tank:

- By AFFTAC, current thermal protection systems are more than adequate to protect a tank for 100 minutes from an engulfing fire with an effective radiating temperature of 816$^oC$.

- By AFFTAC, current thermal protection systems are more than adequate to protect a tank for 30 minutes from an torching fire with an effective radiating temperature of 1204$^o$C, and an effective torch emissivity of 0.536.

- AFFTAC simulations suggest that the pool fire test is the determining test. However, this conclusion changes if the torch properties are adjusted to more accurately represent a credible torch fire accident.

- By AFFTAC and TMID large defects can bring a tank to failure under engulfing fire conditions within 100 minutes.

- Because of uncertainties in the analysis it was concluded that a factor of safety (FOS) should be applied in this analysis. For the case of thermal protection, the FOS is defined as (tank burst pressure at 100 min in fire)/(tank pressure at 100 min in fire). The current thermal protection systems with no defects have FOS > 3.0 at the 100 minute mark in the pool fire simulation. Based on various uncertainties (fire temperature, tank pressure, tank material properties) it was suggested that an FOS = 1.6 would be reasonable.

- If a factor of safety of FOS = 1.6 is used then a 112 tank with propane exposed to fire for 100 minutes can have between 4-24% defective insulation provided a steel jacket is present and is not in direct contact with the tank primary wall. The 24% end of the range assumes the remainder of the tank is protected with perfect insulation with properties as quoted for Kaowool ceramic blanket with 72 kg/m$^3$ density. The 4% end of the range assumes the remainder of the tank is protected with insulation with a conductance of 22.7 W/m$^2$K which represents the maximum conductance that would pass the FRA plate test standard (i.e. keep steel plate sample temperature below 427$^o$C for 100 minutes when exposed to a pool fire).

- The allowable defect area was strongly affected by input variables including:

  - assumed FOS
  - wall thickness
  - PRV setting
  - tank fill
  - fire temperature
  - commodity
  - tank surface emissivity

- Detailed analysis of defects suggested that:

  - as defect gets small the defect temperature may be reduced
  - for small defects, the normal stress may be reduced at high temperatures
  - several small, well separated defects are safer than one large defect

- Defects in the vapour space are more important than defects in the liquid space. However, the possibility of tank rollover makes this difficult to apply in the field.

The above conclusions have come from computer simulation results and therefore are not fully validated. Because of the extreme nature of this problem it is recommended that some testing be conducted to confirm these results.

### *7.2 Thermally Insulated Tanks*

For thermally insulated tanks the analysis is based only on the overall tank thermal conductance.

- Current non-defective thermal insulation systems just meet the required conductance as specified in CGSB standard. Therefore even very small defects may result in the tank not meeting the standards. This does not account for insulation discontinuities.

- If a defect is present then the external convective film coefficient affects the defect thermal conductance. Based on nominal convective coefficients and accounting for thermal radiation in the insulation annulus the allowable defect area is around 2%. This does not account for insulation discontinuities.

- some commodities such as chlorine cannot tolerate high temperatures (i.e. chlorine will cause steel to ignite at temperatures above 215$^{o}$C). Therefore some commodities cannot tolerate any insulation defects.

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# Appendix A: Standards for Thermal Protection and Thermal Insulation of Tank Cars

## § 79.18 Thermal Protection Systems

(a) Performance Standard. When this standard requires thermal protection on a tank car, it shall have sufficient thermal resistance so that there will be no release of any lading from within the tank car, except release through the pressure relief device, when subjected to:

(1) A pool fire for 100 min, and

(2) A torch fire for 30 min.

(b) Thermal Analysis

(1) Compliance with the requirements of par. (a) of this section shall be verified by analysing the fire effects on the entire surface of the tank. The analysis must consider the fire effects on and the heat flux through tank discontinuities, protective housings, underframes, metal jackets, insulation, and thermal protection. A complete record of each analysis shall be made, retained and, upon request, made available for inspection and copying by an authorized representative of Transport Canada. The procedures outlined in *Temperatures, Pressures and Liquid Levels of Tank Cars Engulfed in Fires*, DOT/FRA/OR&D-84/08.11, (1984) shall be deemed acceptable for analysing the fire effects on the entire surface of the tank car.

(2) When the analysis shows that the thermal resistance of the tank car does not conform to par. (a) of this section, the thermal resistance of the tank car shall be increased by using a system listed by Transport Canada under par. (c) of this section or by testing a new and untried system and verifying it in accordance with Appendix D of this standard.

(c) Systems that No Longer Require Test Verification. Transport Canada maintains a list of thermal protection systems that comply with the requirements of Appendix D of this standard and that no longer require test verification. Information necessary to equip tank cars with one of these systems is available from the Director.

(d) Jacketed thermal protection systems shall be flashed around all openings so as to be weathertight. The exterior surface of a carbon steel tank and the inside surface of a carbon steel jacket shall be given a protective coating.

## § 79.100-4 Insulation

(a) If insulation is applied, the tank shell and manway nozzle must be insulated with an approved material. The entire insulation must be covered with a metal jacket of a thickness not less than 11 gauge Manufacturers Standard Gauge (3.04 mm) (0.1196 in) and flashed around all openings so as to be weathertight. The exterior surface of a carbon steel tank and the inside surface of a carbon steel jacket must be given a protective coating.

(b) If insulation is a specification requirement, it shall be of sufficient thickness so that the thermal conductance at 15.5 C (60 F) is not more than 1.533 kJ/h $m^2$ C (0.075 Btu/h $ft^2$ F) temperature differential. If exterior heaters are attached to the tank, the thickness of the insulation over each heater element may be reduced to one-half that required for the shell.

### § 79.101 Individual Specification Requirements Applicable to Pressure Tank Car Tanks

In addition to § 79.100, the individual specification requirements are as follows:

| TC Specification | Individual Specification Requirements | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Insulation | Bursting pressure, kPa (psi) | Minimum plate thickness, mm (in) | Test pressure, kPa (psi) | Minimum manway cover thickness, mm (in) | Bottom outlet | Bottom washout | Reference |
| 105A100ALW | Yes | 3448 (500) | 15.9 (5/8) | 690 (100) | 63.5 (2½)[2] | No | No | — |
| 105A200ALW | Yes | 3448 (500) | 15.9 (5/8) | 1379 (200) | 63.5 (2½)[2] | No | No | — |
| 105A300ALW | Yes | 5171 (750) | 15.9 (5/8) | 2069 (300) | 66.7 (2_)[2] | No | No | — |
| 105A100W | Yes | 3448 (500) | 14.3 (9/16)[3] | 690 (100) | 57.2 (2¼) | No | No | — |
| 105A200W | Yes | 3448 (500) | 14.3 (9/16)[3] | 1379 (200) | 57.2 (2¼) | No | No | — |
| 105A300W | Yes | 5171 (750) | 17.5 (11/16)[1] | 2069 (300) | 57.2 (2¼)[7] | No | No | — |
| 00W | Yes | 6895 (1000) | 17.5 (11/16)[1] | 2758 (400) | 57.2 (2¼)[7] | No | No | — |
| 105A500W | Yes | 8619 (1250) | 17.5 (11/16)[1] | 3448 (500) | 57.2 (2¼) | No | No | 73.314(o) |
| 105A600W | Yes | 10342 (1500) | 17.5 (11/16)[1] | 4137 (600) | 57.2 (2¼) | No | No | 79.314(p), |

| TC Specification | Individual Specification Requirements | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Insulation | Bursting pressure, kPa (psi) | Minimum plate thickness, mm (in) | Test pressure, kPa (psi) | Minimum manway cover thickness, mm (in) | Bottom outlet | Bottom washout | Reference |
| | | | | | | | | 73.314(g) |
| 109A100ALW | Optional | 3448 (500) | 15.9 (5/8) | 690 (100) | 63.5 (2½)[2] | No | Optional | — |
| 109A200ALW | Optional | 3448 (500) | 15.9 (5/8) | 1379 (200) | 63.5 (2½)[2] | No | Optional | — |
| 109A300ALW | Optional | 5171 (750) | 15.9 (5/8) | 2069 (300) | 66.7 (2_)[2] | No | Optional | — |
| 109A300W | Optional | 3448 (500) | 17.5 (11/16)[1] | 2069 (300) | 57.2 (2¼) | No | Optional | — |
| 112A200W | Optional[4] | 3448 (500) | 14.3 (9/16)[3, 5] | 1379 (200) | 57.2 (2¼) | No | No | — |
| 112A340W | Optional[4] | 5861 (850) | 17.5 (11/16)[1] | 2344 (340) | 57.2 (2¼) | No | No | — |
| 112A400W | Optional[4] | 6895 (1000) | 17.5 (11/16)[1] | 2758 (400) | 57.2 (2¼) | No | No | — |
| 112A500W | Optional[4] | 8619 (1250) | 17.5 (11/16)[1] | 3448 (500) | 57.2 (2¼) | No | No | — |
| 114A340W | Optional[4] | 5861 (850) | 17.5 (11/16)[1] | 2344 (340) | Note 6 | Optional | Optional | 103 |
| 114A400W | Optional[4] | 6895 (1000) | 17.5 (11/16)[1] | 2758 (400) | Note 6 | Optional | Optional | 103 |
| 120A200ALW | Yes | 3448 (500) | 15.9 5/8 | 1379 (200) | 63.5 (2½)[2] | Optional | Optional | 103 |
| 120A100W | Yes | 3448 (500) | 14.3 (9/16)[3] | 690 (100) | 57.2 (2¼) | Optional | Optional | 103 |

| TC Specification | Individual Specification Requirements | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Insulation | Bursting pressure, kPa (psi) | Minimum plate thickness, mm (in) | Test pressure, kPa (psi) | Minimum manway cover thickness, mm (in) | Bottom outlet | Bottom washout | Reference |
| 120A200W | Yes | 3448 (500) | 14.3 (9/16)[3] | 1379 (200) | 57.2 (2¼) | Optional | Optional | 103 |
| 120A300W | Yes | 5171 (750) | 17.5 (11/16)[1] | 2069 (300) | 57.2 (2¼) | Optional | Optional | 103 |
| 120A400W | Yes | 6895 (1000) | 17.5 (11/16)[1] | 2758 (400) | 57.2 (2¼) | Optional | Optional | 103 |
| 120A500W | Yes | 8619 (1250) | 17.5 (11/16)[1] | 3448 (500) | 57.2 (2¼) | Optional | Optional | 103 |

***Notes to Pressure Tank Car Specifications***

1. *When steel of* 448 175 *to* 558 495 kPa *(65 000 to 81 000 psi) minimum tensile strength is used, the thickness of the plates shall not be less than* 16 mm *(5/8 in) and when steel of* 558 495 kPa *(81 000 psi) minimum tensile strength is used, the minimum thickness of the plates shall not be less than* 14 mm *(9/16 in).*

2. *When approved material other than aluminum alloys are used, the thickness shall not be less than* 57 mm *(2¼ in).*

3. *When steel of* 448 175 kPa *(65 000 psi) minimum tensile strength is used, the minimum thicknesses of the plates shall not be less than* 13 mm *(½ in).*

4. *Tank cars not equipped with a thermal protection or an insulation system used for the transportation of a Class 2 (compressed gas) material shall have at least the upper two-thirds of the exterior of the tank, including manway nozzle and all appurtenances in contact with this area, finished with a coat of white paint.*

5. *For inside diameter of* 2210 mm *(87 in) or less, the thickness of the plates shall not be less than* 13 mm *(½ in).*

6. *See AAR Specifications for Tank Cars, Appendix E, E4.01 and § 79.103-2.*

7. *When the use of nickel is required by the lading, the thickness shall not be less than* 50 mm *(2 in).*

# Appendix B: Procedures for Simulated Pool and Torch Fire Testing

## PROCEDURES FOR SIMULATED POOL- AND TORCH-FIRE TESTING

**D1.** This test procedure is designed to measure the thermal effects of new or untried thermal protection systems and to test for system survivability when exposed to a 100 min pool fire and a 30 min torch fire.

(a) Simulated Pool-fire Test

(1) A pool-fire environment shall be simulated in the following manner:

(i) The source of the simulated pool fire shall be hydrocarbon fuel with a flame temperature of 871 ± 37.8 C (1600 ± 100 F), throughout the duration of the test.

(ii) A square bare plate with thermal properties equivalent to the material of construction of the tank car shall be used. The plate dimensions shall be not less than 30.48   30.48 cm (1   1 ft) by nominal 1.6 cm (0.625 in) thick. The bare plate shall be instrumented with not less than nine thermocouples to record the thermal response of the bare plate. The thermocouples shall be attached to the surface not exposed to the simulated pool fire and shall be divided into nine equal squares with a thermocouple placed in the centre of each square.

(iii) The pool-fire simulator shall be constructed in a manner that results in total flame engulfment of the front surface of the bare plate. The apex of the flame shall be directed at the centre of the plate.

(iv) The bare plate holder shall be constructed so that the only heat transfer to the back side of the bare plate is by heat conduction through the plate and not by other heat paths.

(v) Before the bare plate is exposed to the simulated pool fire, none of the temperature recording devices may indicate a plate temperature in excess of 37.8 C (100 F) nor less than 0 C (32 F).

(vi) A minimum of two thermocouple devices shall indicate 427 C (800 F) after 13 ± 1 min of simulated pool-fire exposure.

(2) A thermal protection system shall be tested in the simulated pool-fire environment described in par. (a)(1) of this appendix in the following manner:

(i) The thermal protection system shall cover one side of a bare plate as described in par. (a)(1)(ii) of this appendix.

(ii) The non-protected side of the bare plate shall be instrumented with not less than nine thermocouples placed as described in par. (a)(1)(ii) of this appendix to record the thermal response of the plate.

(iii) Before exposure to the pool-fire simulation, none of the thermocouples on the thermal protection system configuration may indicate a plate temperature in excess of 37.8 C (100 F) nor less than 0 C (32 F).

(iv) The entire surface of the thermal protection system shall be exposed to the simulated pool fire.

(v) A pool-fire simulation test shall run for a minimum of 100 min. The thermal protection system shall retard the heat flow to the plate so that none of the thermocouples on the non-protected side of the plate indicate a plate temperature in excess of 427 C (800 F).

(vi) A minimum of three consecutive successful simulation fire tests shall be performed for each thermal protection system.

(b) Simulated Torch-fire Test

(1) A torch-fire environment shall be simulated in the following manner:

(i) The source of the simulated torch shall be a hydrocarbon fuel with a flame temperature of 1204 ± 37.8 C (2200 ± 100 F), throughout the duration of the test. Furthermore, torch velocities shall be 64.4 ± 16 km/h (40 ± 10 mph) throughout the duration of the test.

(ii) A square bare plate with thermal properties equivalent to the material of construction of the tank car shall be used. The plate dimensions shall be at least 121.92 121.92 cm (4 4 ft) by nominal 1.6 cm (0.625 in) thick. The bare plate shall be instrumented with not less than nine thermocouples to record the thermal response of the plate. The thermocouples shall be attached to the surface not exposed to the simulated torch and shall be divided into nine equal squares with a thermocouple placed in the centre of each square.

(iii) The bare plate holder shall be constructed so that the only heat transfer to the back side of the plate is by heat conduction through the plate and not by other heat paths. The apex of the flame shall be directed at the centre of the plate.

(iv) Before exposure to the simulated torch, none of the temperature recording devices may indicate a plate temperature in excess of 37.8 C (100 F) or less than 0 C (32 F).

(v) A minimum of two thermocouples shall indicate 427 C (800 F) in 4 min ± 30 s of torch-simulation exposure.

(2) A thermal protection system shall be tested in the simulated torch-fire environment described in par. (b)(1) of this appendix in the following manner:

(i) The thermal protection system shall cover one side of the bare plate identical to that used to simulate a torch fire under par. (b)(1)(ii) of this appendix.

(ii) The back of the bare plate shall be instrumented with not less than nine thermocouples placed as described in par. (b)(1)(ii) of this appendix to record the thermal response of the material.

(iii) Before exposure to the simulated torch, none of the thermocouples on the back side of the thermal protection system configuration may indicate a plate temperature in excess of 37.8 C (100 F) nor less than 0 C (32 F).

(iv) The entire outside surface of the thermal protection system shall be exposed to the simulated torch-fire environment.

(v) A torch-simulation test shall be run for a minimum of 30 min. The thermal protection system shall retard the heat flow to the plate so that none of the thermocouples on the backside of the bare plate indicate a plate temperature in excess of 427 C (800 F).

(vi) A minimum of two consecutive successful torch-simulation tests shall be performed for each thermal protection system.

**Appendix C: TMID Listing**

(Not available in electronic format/
Non disponible en format électronique)

## Appendix D: Comparison Runs for AFFTAC and TMID

(Not available in electronic format/
Non disponible en format électronique)

Appendix E: TMID and AFFTAC Simulation Output

(Not available in electronic format/
Non disponible en format électronique)

**Appendix F: Kaowool Insulation Properties**

(Not available in electronic format/
Non disponible en format électronique)

## Appendix G: Overall Thermal Conductance Model

(Not available in electronic format/
Non disponible en format électronique)

**Appendix H: Local Heat Transfer and Stress Analysis**

(Not available in electronic format/
Non disponible en format électronique)

# Appendix I: Surface Emissivity

ii) **surface emissivity of tank steel**

In this analysis it will be assumed that the steel surface has an average emissivity of 0.9. This value is not the same as that used in the AFFTAC code where the emissivity is assumed to be 0.8. It is the opinion of the authors of this report that this value is low and should be closer to 0.9. A low surface emissivity reduces the heat transfer from the fire to the tank. Therefore it should be noted that using 0.8 is not a conservative assumption.

The following data illustrates this.

from Hottel and Sarofim

| | |
|---|---|
| sheet steel, rough oxide layer | 0.8 |
| steel plate, rough | 0.94 – 0.97 |
| white enamel fused on iron | 0.90 |
| black or white lacquer | 0.8 – 0.95 |
| flat black lacquer | 0.96 – 0.98 |
| candle soot | 0.95 |

Based on the above list it appears a conservative assumption for emissivity would be 0.9-0.95.

# Appendix J: Radiation View Factor In Vapour Space

Discussion of Vapour Space Heat Transfer

At temperatures above 400$^{oC}$ the strength of steel decreases rapidly. A 112W340 tank with internal pressure at the PRV set pressure (around 2.0 MPa) will burst when the wall temperature reaches about 620-640$^{o}$C. Therefore the problem of predicting tank failure becomes one of determining if and how fast the vapour space wall temperature reaches 620$^{o}$C.

It will be shown that small changes in heat transfer assumptions can have a dramatic effect on the time it takes for the wall to achieve this critical temperature.

One of the most important factors is the cooling effect of the liquid and vapour lading on the inside of the tank shell. The cooling is by convection and radiation. At high temperatures the radiation will dominate and therefore we will only consider radiation here. Ignoring convection the vapour space is a conservative assumption.

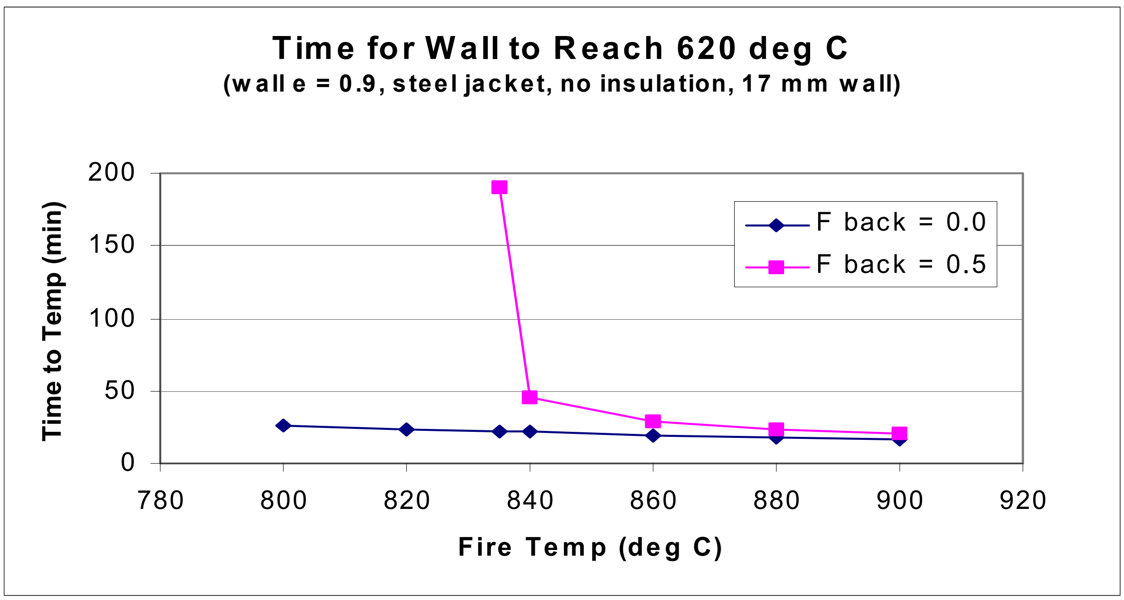


Figure 1: Effect of Vapour Space View Factor on Predicted Wall Temperature

First we will use the AFFTAC assumption that the propane vapour does not absorb thermal radiation. This means the vapour space wall only sees the cool liquid and itself. If the wall sees cool liquid then the wall temperature will be less than if the wall only sees hot wall. This is illustrated in Figure 1.

The figure shows the time to reach 620$^{o}$C as a function of fire temperature for two different inside wall cooling conditions. If the radiation view factor F from the inside wall to the cool liquid lading is 0.5 ( this simulates a tank fill level of greater than 50%) then the wall will be cooled by the liquid and vapour lading (assumed to be at 60$^{o}$C). If the view factor is F = 0.0 (simulates a liquid empty tank) then the wall is radiatively isolated from the lading and it only radiates to itself (assumed to be hot). As can be seen from the figure this single assumption dramatically alters the outcome. For the well cooled case (high fill condition) the wall will never achieve 620°C if the fire is at less than 830°C. On the other hand, the uncooled wall (low fill condition) will quickly reach 620°C no matter what the fire temperature is.

The AFFTAC model calculates the view factor from the wall to the liquid as follows:

$$F = \frac{A_{liquid}}{A_{vapourwall}}$$

where

$A_{liquid}$ = area liquid surface
$A_{vapourwall}$ = area of vapour space wall

This is an approximation but is considered to be a reasonable approach. It gives a good average view factor for the vapour space. With the above approach the AFFTAC model calculates F = 0.0 when the tank is empty. When F = 0.0 the wall temperature will continue to rise. Therefore, with the AFFTAC model (for a tank with only steel jacket) the tank will usually fail soon after it is empty if it does not fail earlier. Therefore a critical defect (as far as AFFTAC is concerned) is one that makes the tank empty in less than 100 min.

It should be noted that the above approach does not take into account that the lading vapour will also absorb some of the thermal radiation. It has been crudely estimated (based on data for Methane, see Hottel and Sarofim) that the vapour could absorb as much as 50% of the thermal radiation from the wall. This of course depends on the commodity being considered. This means the AFFTAC approach should be very conservative.

AFFTAC Approach for a Single Hot Spot

If the AFFTAC code were used to model a small defect with the remainder of the tank thermally protected the AFFTAC code would not predict failure within 100 minutes because it would take much longer than 100 minutes for the tank to empty.

The AFFTAC code can be used to model this problem by doing the analysis in two steps.

step 1: use AFFTAC to model an insulated and jacketed tank to determine
- time to fail
- time to empty
- variation in liquid level with time

Use the above data to determine heat transfer environment for hot spot in vapour space wall.

step 2: based on data from step 1, estimate hot spot T vs time

- assume failure when hot spot T = 620$^o$C
- assume that a defect is present in the vapour space wall
- assume defect does not change overall tank response

Assuming a 112 type car with mineral wool insulation and a steel jacket the time to empty will be approximately 170 minutes. At 100 minutes the tank is still 50% full suggesting considerable cooling effect to the vapour space wall. Assuming an 815$^o$C fire (1500$^o$F), and a surface emissivity of 0.8, then no small defect needs to be repaired.

As stated earlier AFFTAC model will show tank failure in 100 minutes or less if tank is empty in 100 minutes or less. This would require a very large defect (of the order of 50% of the tank surface.

# Appendix K: Pressure Relief Valve Requirements

From CGSB E43-147 Aug 1997

## § 79.15 Pressure Relief Devices

Except for Class 106, 107, 110, AAR 204W and 113 tank cars, tanks must have a pressure relief device made of material compatible with the lading, that conforms to the following requirements:

(a) Performance Standard. Each tank must have a pressure relief system having sufficient flow capacity to prevent pressure build-up in the tank to no more than the flow rating pressure of the pressure relief device in fire conditions as defined in Appendix A of the AAR Specification for Tank Cars.

(b) Settings for Pressure Relief Devices.

(1) A reclosing pressure relief valve must have a minimum start-to-discharge pressure equal to the sum of the static head and gas padding pressure and the lading vapour pressure at the following reference temperatures:

(i) 46 C (115 F) for non-insulated tanks;

(ii) 43 C (110 F) for tanks having a thermal protection system incorporating a metal jacket that provides an overall thermal conductance at 15.5 C (60 F) of no more than 10.22 kJ/h $m^2$ C (0.5 Btu/h $ft^2$ F) temperature differential; and

(iii) 41 C (105 F) for insulated tanks.

(2) The start-to-discharge pressure of a pressure relief device may not be lower than 5.17 bar (75 psig) or exceed 33% of the minimum tank burst pressure except that tanks built prior to October 1, 1997 having a minimum tank burst pressure of 34.47 bar (500 psig) or less may be equipped with a reclosing pressure relief valve having a start-to-discharge pressure of not less than 14.5% of the minimum tank burst pressure but no more than 33% of the minimum tank burst pressure.

(c) Flow Rating of Pressure Relief Devices.

(1) The total flow capacity of each reclosing and non-reclosing pressure relief device must conform to Appendix A of the AAR Specifications for Tank Cars.

(2) The manufacturer of any reclosing or non-reclosing pressure relief device must design and test the device in accordance with Appendix A of the AAR Specifications for Tank Cars.

(3) The flow rating pressure must be 110% of the start-to-discharge pressure for tanks having a minimum tank burst pressure greater than 34.47 bar (500 psig) and from 110 to 130% for tanks having a minimum tank burst pressure less than or equal to 34.47 bar (500 psig).

(d) Testing Reclosing Pressure Relief Valves.

(1) The tolerance for the start-to-discharge pressure for a reclosing pressure relief valve is ±3 psi for valves with a start-to-discharge pressure of 6.98 bar (100 psig) or less and ±3% for valves with a start-to-discharge pressure greater than 6.98 bar (100 psig).

(2) The vapour-tight pressure of a reclosing pressure relief valve must be at least 80% of the start-to-discharge pressure.

(e) Combination Pressure Relief Systems. A non-reclosing pressure relief device may be used in series with a reclosing pressure relief valve. The pressure relief valve must be located outboard of the non-reclosing pressure relief device.

(1) When a breaking pin device is used in combination with a reclosing pressure relief valve, the breaking pin must be designed to fail at the start-to-discharge pressure specified in par. (b) of this section, and the reclosing pressure relief valve must be designed to discharge at no greater than 95% of the start-to-discharge pressure.

(2) When a rupture disc is used in combination with a reclosing pressure relief valve, the rupture disc must be designed to burst at the start-to-discharge pressure specified in par. (b) of this section, and the reclosing pressure relief valve must be designed to discharge at no greater than 95% of the start-to-discharge pressure. A device must be installed to detect any accumulation of pressure between the rupture disc and the reclosing pressure relief valve. The detection device must be a needle valve, trycock, or tell-tale indicator. The detection device must be closed during transportation.

(3) The vapour-tight pressure and the start-to-discharge tolerance is based on the discharge setting of the reclosing pressure relief device.

(f) Non-reclosing Pressure Relief Device. In addition to par. (a) and (c) of this section, a non-reclosing pressure relief device must conform to the following requirements:

(1) Until October 1, 1998, a non-reclosing pressure relief device must incorporate a rupture disc designed to burst at a pressure no less than 100% of the tank test pressure but no more than 33% of the tank burst pressure. After that date, a non-reclosing pressure relief device must incorporate a rupture disc designed to burst at 33% of the tank burst pressure.

(2) The approach channel and the discharge channel may not reduce the minimum flow capacity of the pressure relief device.

(3) The non-reclosing pressure relief device must be designed to prevent interchange with other fittings installed on the tank car, must have a structure that encloses and clamps the rupture disc in position (preventing any distortion or damage to the rupture disc when properly applied), and must have a cover, with suitable means of preventing misplacement, designed to direct any discharge of the lading downward.

(4) The non-reclosing pressure relief device must be closed with a rupture disc that is compatible with the lading and is manufactured in accordance with Appendix A of the AAR Specifications for Tank Cars. The tolerance for a rupture disc is +0 to -15% of the burst pressure marked on the disc.

(g) Location of Relief Devices. Each pressure relief device must communicate with the vapour space above the lading as near as practicable on the longitudinal centre line and centre of the tank.

(h) Marking of Pressure Relief Devices. Each pressure relief device and rupture disc must be permanently marked in accordance with the Appendix A of the AAR Specifications for Tank Cars.

# Appendix L: Outage and Filling Limits

4.3 Outage and Filling Limits

4.3.1 When filling tanks with liquids, sufficient ullage (outage) must be left to ensure that neither leakage nor permanent distortion of the tank will occur as a result of an expansion of the liquid caused by temperatures likely to be encountered during transportation.

4.3.2 Dangerous goods may not be loaded into the dome of a tank car. If the dome of the tank does not provide sufficient outage, vacant space must be left in the shell to provide the required outage.

4.3.3 Liquids and liquefied gases for which there is no specific outage requirement in § 73.314 through § 73.319 must be so loaded that the minimum outage will be as follows:

(1) 1% of the total capacity of a tank, or compartment thereof, or at least 1% of the total capacity of the tank and dome for tank cars and multi-unit tank car tanks at one of the following reference temperatures:

(a) 46 C (115 F) for uninsulated tanks;

(b) 43 C (110 F) for thermally-protected tanks that have a thermal system that meets the requirement of § 79.18 and provides overall thermal conductance at 16 C (60 F) not exceeding 10.22 kJ/h $m^2$ C (0.5 Btu/h $ft^2$ F) temperature differential;

(c) 41 C (105 F) for insulated tanks.

(2) for a material which meets the definition of poisonous by inhalation, the outage must be at least 5% of the total capacity of the tank or compartment at one of the following reference temperatures:

(a) 46 C (115 F) for uninsulated tanks;

(b) 43 C (110 F) for thermally-protected tanks that have a thermal system that meets the requirement of § 79.18 and provides overall thermal conductance at 16 C (60 F) not exceeding 10.22 kJ/h $m^2$ C (0.5 Btu/h $ft^2$ F) temperature differential;

(c) 41 C (105 F) for insulated tanks.