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TOWARDS A SIMPLIFIED MEANS OF PREDICTING TIMES TO FAILURE OF HYDROCARBON-BEARING VESSELS AFFECTED BY HEAT

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ABSTRACT

Times to failure of a steel vessel containing natural gas condensate are predicted by means of a novel approach in which, using equations for non-steady heat conduction, times for the vessel to attain failure temperature under the influence of nearby burning are calculated. One aim of the paper is to set out the method of calculation for the benefit of developers of offshore risk assessment methods, who will incorporate it so that it is applied to whatever conditions prevail.

1. INTRODUCTION

In undertaking an assignment to investigate the safety of a cylindrical vessel used for storage of condensate at an offshore installation, in particular its capability to withstand heat in the event that a gas pipe in the same module failed and the leaked inventory burnt as a jet fire, we utilised analytical solutions for non-steady conductive heat transfer. This provided estimates of times for the cylinder to reach temperatures sufficiently high to cause failure. Such calculations were performed for a number of scenarios including envelopment of the cylinder in post-combustion gas having cooled to about 500 ^{o}C. In this paper we have set up the calculations in such a way as to be of general interest whilst retaining some of the details of the original problem, e.g., the cylinder dimensions and material. It is particularly hoped that the method and results will be noted by developers of software for offshore risk assessment for possible incorporation into such software.

In considering the effect of a jet fire on a nearby hydrocarbon storage vessel, it has to be remembered that the length of the jet fire depends on the size of the leak, that is, the dimension of the orifice through which gas escapes. For a relatively small leak, therefore, direct impingement is not inevitable. The need to use the available space at an offshore platform advantageously results in very compact configurations of pipes and vessels. Nevertheless, there might well be enough flexibility for positioning a condensate vessel such as the one under consideration far enough from any pipe for there to be no direct impingement unless the leak size is uncommonly large. In this event, the vessel will become bathed in post-combustion gas which has partially cooled, and this will transfer heat by convection to the vessel surface A supply of gas to the vessel to sustain the convection will continue for as long as the jet fire does. Heat will be transferred from the outside surface of the vessel to its interior by conduction. The first scenario which we analyse, therefore, is that where a cylinder of condensate is heated in this way, that is, uniformly and without direct flame impingement. Condensate, which is released from natural gas during its transfer from the wellhead, contains alkanes up to about C_5.

2. SCENARIO 1: IMPINGEMENT BY GASES HAVING COOLED TO ≈ 500 ^{o}C

The hottest part of the jet fire itself will approach adiabatic flame temperatures, that is, about 2000 ^{o}C. We consider here a situation such that at the stage where the post-combustion gases encounter the cylindrical vessel the temperature is about 500 ^{o}C. The next paragraph therefore summarises the problem and gives values for the quantities required.

Consider a cylinder 1550 mm i.d., 1600 mm o.d., of axial length much larger than the diameter, containing condensate. The cylinder and contents are initially uniformly at 20 ^{o}C (= T_i). Because of a jet fire some distance away the cylindrical vessel begins to experience convection from the post-combustion gases which are at 500 ^{o}C (= T_g) and transfer heat to the cylinder with convection coefficient (= h) of 25 $Wm^{-2}K^{-1}$. This represents fairly mild convection due to natural drift of the gases towards the cylinder. Gas directed at the cylinder by a pump or fan would give rise to h values of 100 $Wm^{-2}K^{-1}$ or greater.

The cylinder is made of carbon steel, thermal conductivity (k) 45 $Wm^{-1}K^{-1}$, thermal diffusivity (α) 1×10^{-5} m^2s^{-1}, density (σ) 7800 kgm^{-3}, specific heat (c) 470 $Jkg^{-1}K^{-1}$. The steel fails, with resulting rupture, when it uniformly reaches 400^{o}C (= T_f).

Below we obtain an estimate of the time taken for rupture of the vessel.

Since the wall thickness (= L) of the cylinder is much smaller than the diameter, to a satisfactory approximation the wall can be taken to be a plane wall experiencing convection with coefficient h [1]. Fig. 1 shows this arrangement.

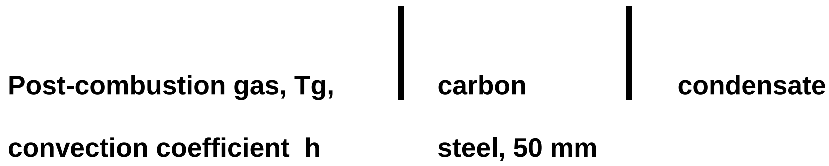

Fig. 1: Schematic of the vessel wall

Note that once heat is transferred into the condensate it remains there, first as sensible heat in the liquid, then as latent heat of vaporisation and finally as sensible heat in the gas. In effect therefore the inside surface of the cylinder is insulated. In applying the analytical solution for a plane wall therefore, we shall need to use the boundary conditions of convection at one side and insulation at the other. The solution to the non-steady conduction equation for this system is [1]:

$$\theta = C_1 \exp(-\xi_1^2 Fo) \cos(\xi_1 x/L) \qquad (1)$$

where

$$\theta = \frac{T - T_g}{T_i - T_g}$$

and x = the co-ordinate in the direction of heat transfer such that x = 0 at the insulated surface, x = L at the surface with convection.

Fo, the Fourier number, is given by:

$$Fo = \alpha t/L^2$$

ξ_1 is the root of:

$$\xi_1 \tan\xi_1 = Bi$$

where Bi, the Biot number, is:

$$Bi = hL/k$$

and C_1 is calculable from:

$$C_1 = \frac{4 \sin \xi_1}{2\xi_1 + \sin(2\xi_1)}$$

Now equation (1) is actually the first term in a power series which is the total solution to the conduction equation, and is a good approximation to the full solution (i.e. the higher terms are negligible) provided that:

$$Fo > 0.2$$

Substituting our values of α and L, this condition is equivalent to:

$$t > 50 \text{ s}$$

so the single-term solution is valid at all times after about one minute. Inserting our values of h, L and k, the Biot number is 0.03, and for this value of Bi the solution of the equation for the eigenvalue ξ_1 is obtainable from tables [1] as:

$$\xi_1 = 0.1732 \text{ radians}$$

and C_1 by substitution into the equation defining it is:

$$C_1 = 1.0049$$

Equation (1) therefore becomes:

$$\theta = 1.0049 \exp(-0.030Fo) \cos(0.1732x/L) \qquad (2)$$

We shall use equation (2) to estimate how long it will take for the outside surface of the cylinder to reach 400°C ($\theta = 0.208$) and the temperature of the inside surface at that time. At the outside surface, x/L = 1:

$$\theta = 1.0049 \exp(-0.030Fo) \cos(0.1732) = 0.618$$

Solving for Fo:

$Fo = 52.0 \Rightarrow t =$ **13000 s (3.6 hour)**

In order to determine the inside surface temperature at this time, we substitute this value of Fo into equation 2 together with a value of zero for x/L to give:

$\theta = 0.211 \Rightarrow T =$ **399°C**

Hence because of its thinness the wall has an almost uniform temperature, and the time of 3.6 hours calculated corresponds very closely to the time at which the cylinder will be uniformly at 400°C, the failure time.

The failure temperature of 400 ^oC is imposed by metal weakening and exacerbation of this by the pressure of the contents which, at such a temperature, will be considerable and will be dependent on the quantity of hydrocarbon inventory which the vessel is holding. The metal in the example here is fairly thin and has a good thermal conductivity therefore, as we have seen, temperature gradients across the conducting dimension are minute. Sometimes vessels with thicker walls - say 1 inch - are used in hydrocarbon storage and are made of an alloy steel, with considerable amounts of nickel, instead of plain carbon steel as in the example herein. These alloy steels have thermal conductivities only a third to a half those of carbon steels, and the combination of increased thickness and lower thermal conductivity means that a significant temperature gradient develops during heating. The internal stress brought about by this is a contributory factor in failure.

We now pass on to other scenarios, and at the conclusion of the work compare results from all of them.

3. SCENARIO 2: VESSEL UNIFORMLY RECEIVING HEAT FLUX FROM ITS SURROUNDINGS DURING ESCALATION

Suppose that the vessel under consideration is some distance from the jet fire but that because of escalation there has been heating of its surroundings in such a way that the vessel uniformly receives thermal flux equivalent to that from an emissive medium at about 700 K, that is, at about 13 kWm^{-2}. Hence whereas the previous scenario had a boundary condition of constant convection *coefficient*, with an actual flux dependent on the temperature difference between the impinging gases and the vessel surface, this second scenario has a boundary condition of constant flux. Physically, this second boundary condition is more vigorous. With the vessel surface at the initial temperature of 20 ^oC, the convective flux q_c is:

$$q_c = (25 \times 480)\ \mathrm{Wm^{-2}} = 12\ \mathrm{kWm^{-2}}$$

declining to 2.5 kWm^{-2} once the surface has approached the fail temperature of 400 ^oC. By contrast, this second scenario involves a constant flux of 13 kWm^{-2}, hence we expect a much shorter failure time.

The principle that the cylindrical surface can be treated as a plane wall with insulation at one side carries through, of course, to this second scenario. The conceptual situation then is a flat plate with constant heat flux at one side and insulated at the other, and this has been solved exactly [2,3]. At larger Fourier numbers (longer times) the solution is of the form of a linear increase in the temperature of the surface exposed to the flux:

$$\frac{dT}{dt} = Q / c\sigma L \qquad \mathrm{Ks^{-1}} \qquad (3)$$

where Q is the flux (W m^{-2}). Substituting the values of Q, c, σ and L:

$$\frac{dT}{dt} = 0.071 \qquad \mathrm{Ks^{-1}}$$

Therefore the time required for a rise in the temperature from 20 to 400 ^oC is **5352 s (1.5 hour)**, under half the time required when the convective heat flux considered previously is responsible for heating the vessel. Our prediction earlier concerning the relative times required in the two cases is therefore upheld. We take the time for surface temperature attainment of T_f to be that for this temperature to be reached across the conducting dimension in view of the very low thermal resistance, as demonstrated fully for scenario 1.

Before passing on to the third and last scenario that we shall consider, we examine the effects of the heating on the contents of the vessel. We have seen that heat transferred through the vessel wall becomes initially sensible heat in the liquid, then latent heat, finally sensible heat in the gas. The fluid contents are a perfect *insulator* in the sense that heat received them cannot be transferred on; this is a consequence of the configuration of the system and is central to the calculations above. However, once an increment of heat passes into the contents of the vessel its effect on their temperature is very quickly evened out because of motion within the fluid. The vessel contents, therefore, whilst rising in temperature with time will at all times display a flat temperature profile and, paradoxically, this is a definition of a perfect *conductor*! It is implicit in the treatment of the vessel contents as being an insulator that loss of heat through discharge of inventory through any relief valve fitted to the vessel is very small.

4. SCENARIO 3: ENGULFMENT OF THE VESSEL BY THE JET FIRE FLAME

Previously we have considered either relatively cool post-combustion gases or radiation from nearby structures as being the source of heat supply to the vessel wall, to the exclusion of direct flame contact. This we now consider.

As already mentioned, there is the possibility of discharge of hydrocarbon through the pressure relief valve of the vessel and this can ignite. However, this has been viewed as fairly improbable on the following grounds:

- Such discharge is gradual, and any gaseous inventory passing out of the relief valve is likely to be dispersed before building up to sufficient concentration for ignition. In any case, in scenario 1 the gas close to the vessel is deficient in oxygen, being composed largely of post-combustion gas. In scenario 2 the air between the radiating surfaces and the vessel, being transparent to the radiation, will remain much cooler than either those surfaces or the receiving vessel surface.

- Auto ignition temperatures of the constituent compounds of condensate are very high [4], e.g. about 500^{o}C for propane, 430^{o}C for n-butane.

Accordingly, if flame engulfment does occur it is more likely to by a jet fire from a pipe having leaked. Consider a jet fire resulting from the release of 25 kgs^{-1} of gas. It is readily shown that this will release heat at 1.4×10^9 W. From the correlation [5]:

$L = 18.5\ Q^{0.41}$

where

L = jet fire length (m)
Q = rate of gas release (kgs^{-1})

The length of the jet fire if not obstructed will be 69 m. We now consider the case where the vessel *does* obstruct the flame and become engulfed by it. This contrasts with the previous two scenarios in two ways. First, whereas each of those considered a uniform condition over the vessel - either a uniform convection coefficient or a uniform heat flux - jet fire impingement will supply heat to a localised area. Secondly, the heat supply is enormously larger than in either of the two cases considered previously. Depending on the area impacted by the flame and the proportion of heat transferred to the vessel rather than to the surroundings, a local flux of the order of MW m^{-2} or higher is expected.

Incorporating jet fire impingement into boundary conditions for application of the conduction equations used previously is therefore not feasible, but direct evidence [6] can be drawn on in order to obtain a rough idea of the time to failure as a result of direct flame impingement. A 1/8 inch (3.2 mm) sheet of a particular steel exposed to a gasoline fire at one side requires approximately half a minute to reach 400^{o}C, and a 1/2 inch plate of the same material requires about 2 minutes. To attempt to fine-tune any comparison between this and the situation under consideration is unnecessary, since in offshore risk assessment the view is usually taken that if vessel failure takes less than a few tens of seconds it is treated as being immediate, there being nil time for emergency responses to take effect. Clearly, direct jet fire impact on the vessel of condensate would be expected to result in immediate failure as defined thus.

5. SUMMARY AND CONCLUDING REMARKS

The results of these calculations and deliberations are summarised in Table 1 below.

The originality of this work has been chiefly in application of the plane wall non-steady conduction equation to the matter in hand, full justification for such application having been given. The reader will be able to modify the calculations for specific cases, e.g., other vessel materials, different failure temperatures. The collective message of the predictions summarised in Table 1 is that the time for failure with convective heat transfer from partially cooled post-flame gas or with radiative flux from affected plant and structures will be long in comparison with those for particular emergency procedures, e.g., evacuation of personnel to the temporary refuge, engagement of sprinklers and deluge sets. Importantly, such information has the potential to contribute to the development of software packages for predicting escalation paths in offshore fires.

Table 1: Summary of the predictions made

Scenario 1: Heating of the vessel uniformly by post-combustion gases at 500^{o}C.	Failure of the vessel after 3-4 hours predicted by application of the conduction equation with a convective boundary condition.
Scenario 2: Uniform flux of 13 kWm^{-2} on the vessel outside surface.	Failure of the vessel after 1-2 hours predicted by application of the conduction equation with a surface flux boundary condition.
Scenario 3: Direct jet fire impingement on to the vessel.	Effectively immediate failure predicted.

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