Fire Research Note

NO. 609

FIRE PROTECTION OF LIQUID FUEL STORAGE TANKS

by

P. H. THOMAS and MARGARET LAW

FIRE
RESEARCH
STATION

FIRE PROTECTION OF LIQUID FUEL STORAGE TANKS

by

P. H. Thomas and Margaret Law

SUMMARY

The amount of venting to relieve the pressure and of water to cool tanks of flammable liquids exposed to fires are examined. The rise in pressure and temperature have been assessed from estimates of the heat such a tank would receive from a fire in its own or another bund. From the values to which the pressure and temperature rise should be limited the amount of pressure relief and water cooling have then been calculated. The pressure relief is close to the values recommended by the National Fire Protection Association though the provision of water cooling gives some grounds for reducing the sizes of vent area. Theoretically about $^1/6$th of the rate recommended by the National Fire Protection Association is capable of extracting all the heat to which the tank is exposed but this requires all the water to be vaporized and none lost by splashing. This would require impractically small nozzles distributed closely over the whole of the circumferential area of the tank and in view of this it is doubtful that any worthwhile reduction in the recommended rates could be effected. without further experimental study.

November, 1965.

MINISTRY OF TECHNOLOGY AND FIRE OFFICES' COMMITTEE
JOINT FIRE RESEARCH ORGANIZATION

FIRE PROTECTION OF LIQUID FUEL STORAGE TANKS

by

P. H. Thomas and Margaret Law

1. Introduction

If a liquid fuel storage tank is liable to be heated by a fire it must be protected to prevent excessive rises in the vapour pressure and in the temperatures of the tank and the fuel. Excess pressures can be relieved by vents and excess temperatures reduced by cooling with water. Since cooling also reduces the pressure rise, the vent area needed in combination with cooling may be smaller than if no water were applied but vents are still necessary. With water sprays keeping the tank temperature down to 100°C the pressure rise with no vent at all would be about 11 p.s.i. for octane initially at 10°C, which is sufficient to burst many tanks. For hexane and heptane the pressure rise would be even greater.

So that the necessary protection may be given in the most economic way it has been decided to examine the problem from first principles. Vent areas and rates of water application are therefore discussed in this note and then compared with the values recommended in the past by the National Fire Protection Association (N.F.P.A.)[1].

2. Heat supply to the tank

A tank may be heated by a fire in its own bund or by radiation from the flames of a nearby tank. Heating due to the former has been measured[2] and is approximately 1.5 cal $cm^{-2}s^{-1}$ (20,000 Btu $ft^{-2}hr^{-1}$). Existing data on the emissivity and thickness of flames from such a bund fire are inadequate to make an accurate independent estimate: the figure of 1.5 cal $cm^{-2}s^{-1}$ is, however, quite consistent with reasonable assumptions as to these quantities over a wide range of tank sizes. The assumption that the flames around a tank are 'thick' leads to heating rates at least twice this value but the assumption is almost certainly too extreme.

Heating from a fire in a separate tank can be estimated from the radiating intensity and size of the flames and from the spacing between the tanks. The radiating intensity depends on the thickness and temperature of the flames and in this situation it is more reasonable to assume the flames are 'thick'. The heating rate would then have an upper limit[3] of 4 cal $cm^{-2}s^{-1}$. The height of the flame from a large liquid fuel fire is between one and two times its base diameter[4] and at a diameter distance, which is the tank separation recommended by the Institute of Petroleum Code[5], the radiation intensity on a tank wall would be reduced to a fraction between 0.24 and 0.36 (Appendix 1), so that an exposed tank would receive an intensity of radiation between 0.96 and 1.44 cal $cm^{-2}s^{-1}$ (13,000 - 19,000 Btu $ft^{-2}hr^{-1}$) according to the flame height. A surface directly facing the flames receives the maximum level of radiation so that in the worst situation the wall of an exposed tank receives more heat than its roof. The upper figure of 1.44 is close to the figure of 1.5 cal $cm^{-2}s^{-1}$ for heating by flames in the bund but it must be borne in mind that with flames in the bund heat is supplied to the tank over its whole perimeter, while with radiation from a nearby tank heat is supplied on one side only. Hence if we assume that in a bund fire the flames are as high as the tank, the total heat input to the tank surrounded by the fire is $1.5 \times 10^4 \pi HD$ cal/s, where H and D are height and diameter of the tank in metres, (6.3×10^4 H'D' Btu/hr, H' and D' in ft) and the input to nearby exposed tanks 0.96 to 1.44×10^4 HD cal/s ($1.3 - 1.9 \times 10^4$ H'D' Btu/hr). (see Table 1).

Table 1

Heating rates of exposed tanks

	Total heat input	Maximum heat input per unit area
Fire surrounding tank	4.7×10^4 HD cal/s (6.3×10^4 H'D' Btu/hr)	1.5 cal $cm^{-2}s^{-1}$ (2.0×10^4 Btu $ft^{-2}hr^{-1}$)
Fire in separate tank	$0.96 - 1.44 \times 10^4$ HD cal/s ($1.3 - 1.9 \times 10^4$ H'D' Btu/hr)	1.5 cal $cm^{-2}s^{-1}$ (2.0×10^4 Btu $ft^{-2}hr^{-1}$)

H and D are height and diameter in metres.
H' and D' are height and diameter in feet.

The upper limit for the total heat input from a nearby tank will therefore be taken as 1.4×10^4 HD cal/s. The upper limit for the rate of heating on unit area of tank wall will be taken as 1.5 cal $cm^{-2}s^{-1}$. The value of total heat input is used to calculate the pressure rise in the tank. The maximum heat input on unit area of tank wall is used to calculate the maximum temperature rise of the wall.

Wind will deflect the flames, tending to decrease the radiative heating and increase the convective heating to a neighbouring tank. The overall effect on the heating rate however is small and will be neglected (see Appendix 2).

3. Pressure rise and vent area

Heating the tank will vaporize some of the liquid and raise the temperature of the vapour and both these effects raise the pressure in the tank until it is high enough for the loss through the vent to equal the rate of generation of vapour. The relative proportions of liquid and vapour will affect the pressure rise and we need to choose the worst condition, that is, the one which gives the maximum pressure rise; it is shown in Appendix 2 that for some fuels it is more onerous to assume that the tank contents are all liquid and for others that the contents are all vapour and the choice for any one fuel depends on the value of the latent heat of evaporation (L) and on the specific heat (Cp) and temperature of the vapour.

If the tank is designed for a maximum allowable pressure rise of Δp max, which is assumed to be small compared with atmospheric pressure, then the vent area A for a rate of heat input Q is given in Appendix 2 for a discharge coefficient of 0.7

as $$A = \frac{\dot{Q}}{0.7\,L}\sqrt{\frac{1}{2\rho g \Delta p_{max}}} \qquad (1)$$

where T is absolute temperature of the vapour

ρ is vapour density

and $L < Cp\,T$

If $L > Cp\,T$ then $Cp\,T$ should be substituted for L in equation (1)

The time constant for pressure rise is very short i.e. there is no "build up" time and the pressure may be regarded as instantaneously at the value appropriate to the instantaneous heating rate.

The equation used by the N.F.P.A. to calculate vent diameters is not given but it is based on an intensity of 6,000 Btu $ft^{-2}hr^{-1}$ over the perimeter of the tank, that is it is based on a value of $\dot{Q}$ of 1.4×10^4 HD cal/s (1.9×10^4 H'D' Btu/hr). We can use this value of $\dot{Q}$ in equation (1) and compare the value of vent diameter so derived with the diameter given by the N.F.P.A. Taking octane as a typical fuel* the vent diameters are compared in Table 2 assuming typical tank dimensions.

Table 2

Comparisons of vent diameters recommended by the N.F.P.A. with diameters calculated from equation (1)

Tank capacity U.S. Gallons	Tank height ft H'	Tank diameter ft D'	Vent diameter - inch			
			Pressure rise $-\Delta p_{max}$			
			3 in water		5 p.s.i.	
			N.F.P.A.	Eqn (1)	N.F.P.A.	Eqn (1)
4,000	10	8	$6\frac{3}{4}$	7	$2\frac{1}{2}$	$2\frac{1}{2}$
56,000	20	22	$12\frac{3}{4}$	$16\frac{1}{4}$	5	$6\frac{1}{4}$
475,000	30	52	20	31	$7\frac{3}{4}$	$11\frac{3}{4}$

There is good agreement for the small tank but equation (1) gives larger diameters for the large tanks. Both sets of diameters vary as the fourth root of the pressure rise, Δp max, as would be expected from equation (1) but the N.F.P.A. diameters vary as the fourth root of the tank volume, whereas with heat input related to the surface area, they might be expected to vary as the cube root**.

*It can be shown that for heavier fuels the calculated diameters are not much greater than for octane, e.g. for decane (kerosene) they are 10 per cent greater.

**In reference (2) it is claimed that it is allowable to take proportionately smaller vent areas for large tanks than would be calculated. It may have been assumed that the larger tanks have a smaller proportion of their surface area enveloped in flame.

It is interesting to note that the value of $\dot{Q}$ used in Table 2 is the same as the value we have estimated as the upper limit for the rate of heat input from a nearby tank fire and bearing in mind the broad assumptions it would seem reasonable to accept the N.F.P.A. vent diameters as adequate for this hazard, if experience has justified the reduction employed for large tanks.

4. Temperature rise of tank

For an intensity of 1.5 cal $cm^{-2}s^{-1}$ the equilibrium temperature rise of the tank wall above the liquid level would be over 700°C, with a time constant of the order of a minute, so that whichever way the tank is heated, whether by a nearby burning tank or by its own bund fire, provision must be made for cooling. Keeping the tank walls at 100°C using water would be sufficient to maintain the integrity of the tank and prevent auto-ignition of the fuel. Somewhat higher temperatures would still be safe but it would not then be practicable to use water.

In estimating the cooling needed we should consider the worst condition, when the contents of the tank are all vapour and there is no large sink for heat within the tank. Of the heat falling on the tank wall part will raise the temperature of the wall, part will raise the temperature of the cooling water and part will raise the temperature of the vapour in the tank. If the temperature of the water rises to 100°C, part or all of it will be converted to steam and the latent heat of vaporization will play a large part in cooling. A lower limit to the amount of water needed can thus be estimated by assuming all the water is vaporized. An upper limit can be estimated assuming there is no vaporization at all and that the tank wall temperature is kept down to a value less than 100°C. Since these estimates will be based on the assumption that the tank wall is uniformly wetted, the attainment of unbroken wetting must also be considered.

To simplify the problem of calculating temperature rises we neglect the radial thermal resistance and capacity of the tank wall (which are small) so that as shown in Appendix 3 the water and tank wall may be assumed to be at the same temperature. If the intensity on the wall is I, and the water is applied at a rate $\dot{m}''$ (weight of water per unit area per unit time) and there is no vaporization, then the temperature rise of the water if it is not at 100°C is as given in Appendix 2.

$$\theta_w = \frac{I}{\dot{m}''c\left(1 + \frac{\alpha}{\dot{m}''c}\right)} \qquad (2)$$

where c is the specific heat of water

and I is the heat absorbed by the water per unit area of tank surface. α is the heat transfer coefficient from the wall to the vapour; it is shown in Appendix 2 that this quantity is independent of the height of the tank for all practical sizes. Values of $\dot{m}''$ are shown in Table 3.

Table 3

Amounts of water $\dot{m}''$ required on the assumption that the water is heated to the temperature of the tank wall and none is lost by vaporization

I cal $cm^{-2}s^{-1}$	Θ_w - temperature rise of water and tank from 20°C degC							
	20		40		60		80	
3.0	0.15	1.85	0.075	0.92	0.05	0.62	0.0375	0.46
2.0	0.10	1.24	0.05	0.63	0.033	0.42	0.025	0.31
1.5	0.075	0.93	0.0375	0.46	0.025	0.31	0.0187	0.23
1.0	0.05	0.62	0.025	0.31	0.017	0.21	0.0125	0.15
0.5	0.025	0.31	0.0125	0.15	0.008	0.10	0.006	0.075

Units of left hand columns gm $cm^{-2}s^{-1}$
Units of right hand columns Imp. gal $ft^{-2}min^{-1}$

If no water is vaporized the rate recommended by the N.F.P.A. of 0.17 Imp. gal $ft^{-2}min^{-1}$ is seen from Table 3 to be based on heating to 100°C with an intensity of just below 1.5 cal $cm^{-2}s^{-1}$. If all the water is vaporized the lower limit to the amount necessary for an intensity of 1.5 cal $cm^{-2}s^{-1}$ is 0.0024 g $cm^{-2}s^{-1}$ or 0.03 gal $ft^{-2}min^{-1}$. If the tank wall were completely smooth even the lower of these rates of flow would several times exceed the minimum rate necessary to wet the surface at 100°C (see Appendix 2) but the likely disruptive effect of the wind and of surface irregularities means that a freely falling film cannot be relied on to produce unbroken wetting. The water would need to be applied in such a way that no area greater than about 40 cm^2 (say 6 in^2) remained dry (see Appendix 2) and to obtain wetting of such uniformity even with the negligible water run off implied by vaporizing all the water, means that the water would have to be applied from nozzles distributed over the whole of the area to be protected. If the spray nozzles were, say, 2 ft from the surface it is unlikely that the distance apart for relatively uniform coverage could exceed 4 ft so that for an overall rate of 0.03 gal $ft^{-2}min^{-1}$ each nozzle would have to provide about 0.4 gal $ft^{-2}min^{-1}$. This raises difficulties because the nozzles would have to be very small for use in the open. Even at a pressure as low as 4 p.s.i. the nozzle diameter could not exceed about 1/12 inch. Such an installation, even neglecting the loss of water by splash might be impracticable. Fewer, but larger nozzles would have to be further from the surface of the tank and this could lead to the sprays being blown about by winds. It is therefore more practical to have a larger flow rate forming a uniform film which will carry water at least for some distance down the tank surface. Nozzles do not then have to be provided over the whole surface area of the tank and each one can therefore be larger.

Summing up the above remarks; the maximum economy of water will lead to difficulties in ensuring the degree of uniformity required and will probably raise the costs of the distribution pipe work and nozzle fittings. Certainly further consideration would have to be given to the practical design of the water system if any considerable saving were to be required in the water flows necessary.

If the rate is that recommended by the N.F.P.A.[6] of 0.2 U.S. gal $ft^{-2}min^{-1}$ or 0.17 Imp. gal $ft^{-2}min^{-1}$ there will be uniform wetting and the value of θ_w calculated from equation (2) is 109 degC ($\alpha/\dot{m}''c$ is negligible compared with unity). For an ambient temperature of 10 - 20°C it would follow that a little of the water must be vaporized.

It is shown in Appendix 2 that for such flows the thickness of the film of water falling down the tank is between $\frac{1}{2}$ and 1 mm and therefore large enough to absorb the major part of the radiated heat before it reaches the tank wall. This means the water is probably hotter than the tank and that vaporization occurs at the free water surface rather than as a result of "boiling" from the hot tank wall which might tend to disrupt the water layer.

5. Vent area with water cooling

If the tank is cooled by water then we can allow for this reduction in heat input when calculating the vent area needed and

$$\dot{Q} = \dot{Q}_c < \alpha\,\theta_w \times \text{(heated tank surface area)}$$

i.e. $\dot{Q}_c < \alpha\,\theta_w \times 10^4 \pi HD$ for a bund fire

$\dot{Q}_c < \alpha\,\theta_w \times 10^4 HD$ for a nearby tank fire

where θ_w is given by equation (2).

For a water application rate of 0.17 Imp. gal $ft^{-2}min^{-1}$ the tank would be at about 100°C and for octane vapour $\alpha \simeq 3 \times 10^{-4}$ cal $cm^{-2}s^{-1}degC^{-1}$ so that $\dot{Q}_c$ is 0.095×10^4 HD cal/s (0.13×10^4 H'D' Btu/hr) for a bund fire and 0.030×10^4 HD cal/s (0.04×10^4 H'D' Btu/hr) for a fire in a nearby tank. These values of $\dot{Q}_c$ can be compared with the value 1.4×10^4 HD cal/s for $\dot{Q}$ used in calculating the vent diameter in Table 2. If water is applied then the diameter in this Table can be reduced to a quarter for a bund fire and this will provide more than enough protection for the less severe condition of a nearby fire.

Conclusions

Whether a tank is exposed to fire in its own bund or to radiation from a nearby burning tank, its temperature above the liquid level will be well over 100°C within a short time unless it is cooled. With a water application rate of 0.17 Imp. gal $ft^{-2}min^{-1}$, the value recommended by the N.F.P.A., and a fire in the bund the vent diameters needed for various degrees of pressure relief are given in Table 4. For protection against fire in a nearby tank the vents can be smaller and these are also given in Table 4.

The amounts of water calculated in this note are found to be similar to the recommendations of the N.F.P.A., if no allowance is made for vaporization. In view of the uncertainties of some of the quantities involved there do not appear to be any grounds for recommending any changes. The design of the water protection must be such as to ensure good coverage of the tank and this requirement becomes more onerous if less water is to be used.

Table 4

Vent diameters* for tanks cooled by water applied at 0.17 Imp. gallon ft^{-2}min^{-1}

Tank capacity 1,000 Imperial gallons	Vent diameter - inch							
	Bund Fire				Fire in nearby tank			
	Pressure rise				Pressure rise			
	3 in water	1 p.s.i.	$2\frac{1}{2}$ p.s.i.	5 p.s.i.	3 in water	1 p.s.i.	$2\frac{1}{2}$ p.s.i.	5 p.s.i.
1	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
4	2	$1\frac{1}{4}$	1	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{2}$
10	$2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
40	$4\frac{1}{4}$	$2\frac{1}{2}$	2	$1\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1
100	$5\frac{1}{4}$	3	$2\frac{1}{4}$	2	3	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$
400	8	$4\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{1}{4}$	$4\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{3}{4}$
1,000	11	$6\frac{1}{4}$	5	$4\frac{1}{4}$	$6\frac{1}{4}$	$3\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{1}{2}$

*These diameters are for free openings assuming a discharge coefficient of 0.7

References

(1) CROSBY - FISKE - FORSTER. "N.F.P.A. Handbook of Fire Protection" 12th Edition 1962. N.F.P.A. Boston, U.S.A.

(2) DUGGAN, J. J., GILMOUR, C. H. and FISHER, P. F. "Venting of Tanks Exposed to Fire". N.F.P.A. Quarterly Oct. 1943. p. 132 - 153.

(3) RASBASH, D. J., ROGOWSKI, Z. W. and STARK, G. W. V. "Properties of Fires of Liquids". Fuel XXXV No. 1. Jan. 1956.

(4) BLINOV, V. I. and KHUDIAKOV, G. N. Dokl. Akad. Nauk. S.S.S.R. 1957, 113, 1094.

(5) Institute of Petroleum Refining Safety Code. Institute of Petroleum, London, 1956.

(6) "Water Spray Systems for Fire Protection". N.F.P.A. No. 15-1962 N.F.P.A. U.S.A.

(7) JAKOB, M. "Heat Transfer" Vol. II. John Wiley and Sons, New York, 1957. p. 356 - 368.

(8) HAMILTON, D. C. and MORGAN, W. R. "Radiant Interchange Configuration Factors". U.S. National Advisory Committee for Aeronautics Technical Note 2836. December 1952.

(9) BRESSER, R. "Experiments in the evaporation of thin liquid films". 2. Ver. Dtsch. Ing. 1958 100 630-638.

(10) HARTLEY, D. E. and MURGATROYD, W. "Criteria for the break-up of thin liquid layers flowing isothermally over solid surfaces". Int. J. Heat Mass Transfer 1964. 7 (9) 1003-15.

(11) THOMAS, P. H. "The effect of wind on plumes from a line source of heat" Fire Research Note 572. (Nov. 1964).

(12) RANKINE, A. O. Experimental Studies in Thermal Convection. Proc. Phys. Soc. London (1950) 63 Part 5 (365A) 417.

APPENDIX 1

Flame radiation from a burning tank

Flames may be coming from fuel burning in the bund, see Fig. 1(a), or from the open roof of the tank, see Fig. 1(b). Their height H is between one and two times their base diameter D.

The minimum distance from other tanks is D. Tanks of the same size or smaller will be at this distance and they will receive the maximum level of radiation either on the wall or on the roof depending on their size and whether the situation of the flames is as in Fig. 1(a) or 1(b). The maximum level on a wall will be at a point opposite the centre of the flames, Fig. 2(a), and on a roof at a point opposite the centre of the base of the flames, Fig. 2(b).

The reduction in intensity with distance from various shape radiators has been computed[8], this reduction factor generally being known as the configuration factor. For liquid fuel fires the flames are narrower at the top than at the base but for simplicity it can be assumed that they are uniform in width, i.e. that the flame shape is rectangular; the error introduced by this assumption is on the safe side.

Table A 1 gives the configuration factors for flame heights of one and two diameters assuming a rectangular radiator.

Table A 1

Maximum configuration factor at distance D

Flame height	Tank wall	Tank roof
D	0.24	0.07
2D	0.36	0.12

The table shows that the worst condition is heating of the tank wall.

APPENDIX 2

1. The pressure rise in a vented tank

We shall calculate the flow of vapour through the vent for what we shall show below are the two extreme cases:

(1) all the contents are vapour
(2) all the contents are liquid

We write $\dot{Q}$ as the rate at which thermal energy enters the mass M of vapour at temperature $T^{o}K$ in a tank containing a volume V of vapour.
C_p and C_v are the specific heats of the vapour at constant pressure and volume respectively and assumed constant.

$\gamma = C_p/C_v$
p is the pressure in the tank
ρ is the density of vapour
t is the time

The energy balance for the gaseous phase is obtained as follows:
We neglect the small rate of change of liquid volume and consider a fixed volume V in the tank of air and vapour, containing a mass m of internal energy E.
If $\dot{Q}$ is the rate at which heat enters the gaseous phase and no heat enters the liquid we can write

$$\dot{Q} = \frac{d}{dt}(mE) - H\frac{dm}{dt} \tag{1}$$

where H is the enthalpy of the vapour-air mixture.

For small pressure differences a first approximation to the equation for the discharge through the vent is the same as the Bernoulli equation viz

$$\frac{dm}{dt} = -A_v\sqrt{2g\rho(p-p_o)} \tag{2}$$

where p_o is the atmospheric pressure and A_v the aerodynamic free area of the vent.

It does not matter whether we take the boundary of the gaseous system just within or just outside the tank since H is constant in the discharge. Equation (1) can equally well be obtained by considering a fixed mass m expanding to fill the tank by expelling a mass δm.

viz $$\dot{Q} = m\frac{dE}{dt} + p\delta V \tag{3i}$$

where $$\frac{m}{V} = \frac{\delta m}{\delta V} = \rho \tag{3ii}$$

where V is the constant volume of the gases in the tank.

Equations (1), (3i) and (3ii) are consistent since

$$H = E + P/\rho$$

The thermodynamic equations give

$$\frac{dE}{dt} = C_V \frac{dT}{dt} + \left\{ T\left(\frac{dP}{dt}\right)_V - P \right\} \frac{dV}{dt} \qquad (4i)$$

We now consider the pressure and temperature to be low enough to treat the vapour air mixture as a perfect gas for which

$$\frac{P}{\rho} = RT = (C_p - C_v)T \qquad (4ii)$$

$$\text{and} \quad T\left(\frac{dP}{dt}\right)_V = P \qquad (4iii)$$

From equations (1), (2), (3) and (4)

$$\frac{(\gamma-1)}{V} \dot{Q} = \frac{dp}{dt} + A_V\sqrt{2g\rho(p-p_0)} \qquad (5)$$

which describes how the pressure rises towards a maximum value. Since our design criterion allows only a small pressure rise we put

$$p\sqrt{p-p_0} \simeq p_0\sqrt{p-p_0}$$

We then have

$$\frac{dp}{\dot{Q}\frac{(\gamma-1)}{V} - \gamma A_V p_0 \sqrt{\frac{2(p-p_0)}{MV}}} = dt \qquad (6)$$

An upper limit to the maximum pressure p_{max} is obtained from equation (5) with $\frac{dp}{dt}$ equal to zero and M equal to its initial i.e. largest value.

$$\Delta p_{max} = p_{max} - p_0 = \frac{\dot{Q}^2(\gamma-1)^2\rho_0}{2A_V^2\gamma^2 p_0^2} = \frac{\dot{Q}^2}{2\rho A_V^2 C_p^2 T^2} \qquad (7)$$

the suffix o denoting initial values.

We now show this pressure is reached quickly. The integral of the left hand side lies between the values taking M as its initial value M_o and a lower value M_L, this lower value being obtained from equation (2) with $p - p_o$ set at its upper value. Thus with M equal to M_1 and $M_L < M_1 < M_o$ equation (6) is integrated by introducing

$$\omega = \frac{\sqrt{p-p_0}\; V\gamma A_V p_0}{\dot{Q}(\gamma-1)}\sqrt{\frac{2}{VM_1}} \qquad (8)$$

from which

$$\int_0^{\omega_t} \frac{\omega\, d\omega}{1-\omega} = \frac{\gamma^2 A_v^2 p_0^2 t}{m_1 \dot{Q} (\gamma - 1)} \qquad (9)$$

A time constant for the pressure change can be defined by

$$\tau_p = \frac{m_1 \dot{Q} (\gamma - 1)}{\gamma^2 A_v^2 p_0^2} \qquad (10)$$

A time constant for the change in M is obtained from equation (2)

$$\tau_m = \frac{M_0}{A_v \sqrt{2 \rho_0 (p_{max} - p_0)}} \qquad (11)$$

$\therefore$ from equations (7), (10) and (11)

$$\frac{\tau_p}{\tau_m} = \frac{2\, \Delta p_{max}}{p_0}$$

Thus, if the vent area is designed to limit Δp_{max} to a small value the pressure changes quickly to a quasi-steady value while M changes more slowly.
In equation (9) we put W_t equal to the value of $1/\sqrt{2}$ which from equations (7) and (8) corresponds to a rise in pressure of half the maximum rise. The integral in equation (9) can be evaluated

$$\int_0^{1/\sqrt{2}} \frac{\omega\, d\omega}{1-\omega} = \left[\log \frac{1}{1-\omega} - \omega \right]_0^{1/\sqrt{2}} = 0{\cdot}52$$

and the corresponding value of t from equations (9) and (7) is

$$t_{\frac{1}{2}} = \frac{0{\cdot}52 \times 2 \times \Delta p_{max} \times V}{\dot{Q} (\gamma - 1)}$$

$$\simeq \frac{\Delta p_{max}\, D}{4 I (\gamma - 1)} \qquad (12)$$

For a tank 30 ft (900 cm) in diameter, Δp_{max} less than 5 lb in^{-2} and I equal to 1.5 cal $cm^{-2}s^{-1}$ (i.e. 4,300 ft lb $ft^{-2}s^{-1}$) we have

$$t_{\frac{1}{2}} \simeq \frac{1}{\gamma - 1} \quad s$$

so that for $\gamma - 1 > 0.1$ say, $t_{\frac{1}{2}}$ is of order 10 s or less and in terms of an exposure of several minutes at least it is possible to regard the pressure rise in the tank as equal to its quasi steady value corresponding to the instantaneous mass of vapour in the tank i.e. the term dp/dt can be omitted from equation (5).

If as an opposite extreme we consider all the heat to enter the liquid and that no air remains in the tank we have a steady state solution that

$$\frac{\dot{Q}}{L} = A_V \sqrt{2\rho(p - p_0)} \qquad (13)$$

with T and m constant.
where L is the latent heat of the vapour.

The pressure rise is given by

$$\Delta p_{max} = \frac{1}{2\rho} \left(\frac{\dot{Q}}{L A_V} \right)^2 \qquad (14)$$

This is exactly the same as equation (7) except that we have in effect replaced C_p T by L so that the vent area should be calculated from the smaller of these two quantities.

2. Protection by a water film

2.1. The absorption of heat by a water film

We shall show here that any practical thickness of water film running down tanks can be regarded as being heated to the wall temperature either by conduction or by radiation absorption.

We consider the wetting of a tank wall by a film of water flowing under gravity. The minimum thickness which wets an unwetted area is discussed below at the end of this section. We shall assume the tank surface is at Θ_w above ambient, the surface vertical and the flow $\dot{m}'$ per second per unit perimeter.

The height of the film is taken as H - the tank height - and μ is the viscosity of water at the temperature of the water film.

The rate at which heat is taken up by the water from unit perimeter of a hot wall may be written as

$$\dot{q}' = \dot{m}' c \, \theta_w \phi \tag{15}$$

where c is the specific heat of water

ϕ is the ratio of the final mean water temperature rise to θ_w

By definition $\dot{q}' = I H$ (16)

where I is the heat absorbed by water per unit area of tank surface. There are two likely mechanisms for the heating of the water. For thicker films radiation is absorbed directly by the water and the water is then likely to be hotter than the wall. For thin films radiation is transmitted through the water and heat is absorbed by the water by conduction from the tank walls i.e. the water is cooler than the wall ($\phi < 1$). We consider the latter first. Jakob[7] describes the theory of heat transfer between a freely falling film and a vertical surface at uniform temperature. This is based on the work of Nusselt and Dukler and Bergelin and the following formulae can be used

$$\frac{\dot{m}'}{\mu} = 3s^* + 2.5 s^* \log_e s^* - 64 \quad (\text{for } s^* > 30) \tag{17}$$

$$s^* = g^{\frac{1}{2}} s^{3/2} \tag{18}$$

and ϕ is given by Jakob as function of ξ and $\xi = \frac{\nu k H}{g s}$

where ρ_w is the density

g is the acceleration due to gravity

k is the thermal diffusivity of water (= 0.0015 cgs units)

ν is the kinematic viscosity $= \mu / \rho_w$ (= 0.005 cgs units)

s is the thickness of the film

Fig (3) shows the calculated value of $1/\phi$, the ratio of the actual flow required to the flow needed if all the water were heated to the wall temperature, for

H = 900 cm ($\sim$30 ft), 300 cm (10 ft) and 150 cm (5 ft). It is seen that there is a critical value of $\dot{m}'/\mu$ which for the largest H is about 1400 when $S^* \simeq 140$ corresponding to a value of ϕ of about 0.60 - 0.70. Above this value of $\dot{m}'/\mu$ the loss of efficiency more than outweighs the extra cooling capacity of the greater flow, i.e. the thickness of the film is so large that it falls too quickly to reach the temperature of the tank wall.

Calculations show it is sufficient for the purposes of this paper to take the criterion for the critical value of $\frac{IH}{\Theta_W \mu}$ in this range of H as 1400, when

$$\phi \simeq 0.7$$

i.e. from Jakob $\quad \xi \simeq 0.2$

and $\quad S^* = (500H)^{\frac{3}{8}}$

or

$$\frac{0.29}{\nu k} = \frac{H}{S^4} = 2.6 \times 10^7 \text{cm}^{-3} \qquad (19)$$

For H equal to 900 cm we have S = 0.076 cm and this corresponds to $S^* = 133$ at the critical condition, which is close to the value obtained from Fig. (3). For H = 150 cm the film thickness at the critical value is reduced by $1/6^{\frac{1}{4}}$ i.e. to 0.048 cm. Since the effective attenuation coefficients for thermal radiation in films of water are of order 100 cm^{-1}, the above values of the thickness are quite large enough for most of the radiated heat to be absorbed in the water and this means that any larger thickness of water film can be regarded as absorbing all the incident radiation entering through the film and any smaller thickness absorbing the heat by conduction from the wall if the radiation is not absorbed.

2.2. Uniformity of Cover

The following calculation, albeit only an approximate representation of the effect of a break in the water coverage, shows that only a very small break can be tolerated.

We assume that the heat transfer coefficients α for the tank surface are the same on the fuel vapour and the air sides. We shall neglect the finite conductance across the tank wall.

Consider a circular area of radius 'a' with the tank temperature Θ_W above ambient at the edge of this area. The equation of radial heat conduction is therefore

$$I - 2\alpha\theta = \omega K \frac{1}{r}\frac{d}{dr}\left(r\frac{d\theta}{dr}\right) \qquad (20)$$

where w is the wall thickness of the tank

K is its thermal conductivity

and θ is the local temperature rise at radius r in the wall. θ is θ_w at $r = a$ and by symmetry $\frac{d\theta}{dr}$ is zero at $r = 0$. The solution to equation (20) with these conditions is

$$\theta = \left(\theta_\omega - \frac{I}{2\alpha}\right)\left[\frac{I_0\left(r\sqrt{\frac{2\alpha}{K\omega}}\right)}{I_0\left(a\sqrt{\frac{2\alpha}{K\omega}}\right)}\right] + \frac{I}{2\alpha} \tag{21}$$

where I_0 is the modified Bessel function of the first kind of order zero. The difference between the centre and the edge temperature is therefore

$$\Delta\theta = \left(\frac{I}{2\alpha} - \theta_\omega\right)\left[1 - \frac{1}{I_0\left(a\sqrt{\frac{2\alpha}{K\omega}}\right)}\right] \tag{22}$$

with $I \simeq 1.5$ cal cm^{-2}s^{-1} and $\omega \simeq 1$ cm, $\theta_w \simeq 100^\circ$C, $\alpha \simeq 5 \times 10^{-4}$ cal cm^{-2}s^{-1}degC^{-1}, and $K = 0.1$ cal cm^{-1}degC^{-1}d^{-1} we obtain

$$\Delta\theta = 1400\left[1 - \frac{1}{I_0\left(\frac{a}{10}\right)}\right]$$

If $\Delta\theta$ must not exceed, say 50 degC

$$I_0\left(\frac{a}{10}\right) = 1 + \frac{1}{27}$$

For I_0 close to unity we can write

$$I_0\left(\frac{a}{10}\right) \simeq 1 + \left(\frac{a}{20}\right)^2$$

so that the unwetted area must not exceed

$$\pi a^2 = \pi \times \frac{400}{27} \simeq 45 \text{ cm}^2$$

This shows that unwetted areas larger than a few square cm cannot be tolerated.

2.3. Minimum flow to wet surface

The minimum rate of wetting for a film at 100°C reported by Bresser[9] was 103 lb $ft^{-1} hr^{-1}$ and this agrees with theoretical calculations by Hartley and Murgatroyd[10]. This is a flow of 0.4 g $cm^{-1} s^{-1}$ (or of $\frac{\dot{m}}{\mu}$ of 80). For a 30 ft high tank this flow is about $^1/_{30}$th of a flow of 0.17 Imp. gal $ft^{-2} min^{-1}$. Although this might suggest that the flow necessary to give protection would produce a stable film, irregularities in the surface or a wind may well destroy it and the water must be applied in a way which does not rely on a gravity film fed from the top.

2.4. Disruption of film by wind and flames

A rough criterion which should be good enough for our purpose for disruption of the water film by flames is that the shear due to the flames on a smooth surface is small compared with the wall shear.

The shear of the flames gases which are assumed to be moving quickly relative to the liquid film, is

$$\tau_g = \frac{C_f \rho v^2}{2}$$

where ρ is the density of flame gases

and C_f is a friction factor which is a function of Reynolds No. but which may be taken here as of order 0.005

The wall shear is from equation (18)

$$\tau_{wall} \simeq \rho_w s g = \rho_w (g s^* \nu)^{2/3}$$

and

$$\frac{\tau_g}{\tau_{wall}} \simeq \frac{C_f}{2}\left(\frac{\rho}{\rho_w}\right)\frac{v^2}{(g s^* \nu)^{2/3}} \qquad (23)$$

For flames 30 ft high v is unlikely to exceed 25 ft/s $\simeq$ 750 cm/s. With ρ as 0.3×10^{-3} g/cm^3, ν for water as 0.005 cm^2/s and s^* at its lower value of 50

$$\frac{\tau_g}{\tau_{wall}} = \frac{0.005}{2} \times \frac{0.3}{1000} \times \frac{0.56 \times 10^6}{(981 \times 50 \times 0.005)^{2/3}}$$

$$\simeq \frac{1}{100}$$

which suggests that the upward shear of flames will not disrupt the flowing film. However, winds produce shear forces of similar order of magnitude as flames but in the horizontal direction and these are likely to disrupt the film, at least on the windward side, which will inevitably be the side exposed to a nearby fire.

3. The effect of wind on heating

Wind will deflect the flames and will affect both the convective and the radiative heat transfer.

The burning rate of the exposed liquid could be increased by the wind but this effect can probably be neglected here because the liquid surface is partly sheltered from the wind either by the tank walls or the bund.

The temperature downwind of a long line fire will exceed that of a shorter line fire because the dilution will be less. Thus an estimate of the effect of wind on the convective heating assuming the fire to be infinitely long will give an overestimate. A correlation of the data[11] obtained by Rankine[12] gave the maximum downwind temperature rise at a distance x from a line heat source of strength $\dot{Q}'$ heat units per unit length of line per unit time as

$$\theta = 2.25\,\frac{v^2 T_0}{g x}\left(\frac{U}{v}\right)^{0.19} \tag{24}$$

where v is the characteristic thermal velocity = $\left(\frac{g\dot{Q}'}{\rho c T_0}\right)^{\frac{1}{3}}$

T is the absolute temperature

ρ is the density of air

c is the specific heat of air

U is the wind speed

This equation is derived from data in the cool region of the downwind plume where the effect of density differences other than on the buoyancy are negligible and is valid in the region $1 < \frac{U}{v} < 3$

For the purpose of this paper it is sufficient to neglect the weak effect of the term $\left(\frac{U}{v}\right)^{0.19}$ which can be taken as unity. The value of $\dot{Q}'$ can be taken as the total rate of heat release by the fire divided by the diameter. If a calorific value of 10,000 cal/g and a liquid with a density of 0.89 g/cm^3 and a burning rate of 5 mm/min are taken as representative figures, the heat release per unit area is

$$\dot{Q}'' = \frac{10{,}000 \times 0.8 \times 0.5}{60} = 67 \text{ cal cm}^{-2}\text{s}^{-1}$$

and for a 30 ft diameter tank we have

$$\dot{Q}' = \frac{\pi}{4} \times 67 \times 930 = 49 \text{ K cal cm}^{-1}\text{s}^{-1}$$

$$\text{and} \quad \mathcal{V} \simeq 800 \text{ cm/s} \quad \simeq \quad 18 \text{ m.p.h.}$$

A line plume is unlikely to be bent over in winds which do not exceed the horizontal entrainment velocity in still air and a real plume from a finite source will be even less affected. This entrainment velocity is $0.29\mathcal{V}$ and is equivalent to $7\frac{1}{2}$ ft/s (or 5 m.p.h.). In strong winds of over 800 cm/s (or 18 m.p.h.) the temperature rise one diameter away is about

$$\Theta = \frac{2.25 \times 800 \times 300}{981 \times 930} = 470^{\circ}\text{C}$$

neglecting the weak effect of $(\frac{U}{\mathcal{V}})^{0.19}$ once $\frac{U}{\mathcal{V}}$ exceeds 1. This suggests that the flames from one tank might then impinge on another tank.

The heat transfer by convection across a turbulent boundary base to a solid surface is given by

$$Nu = 0.0356\ (Pr\ Re)^{0.8} \tag{25}$$

where Nu is the Nusselt number

Pr is the Prandtl number

and Re is the Reynolds number

With $U \simeq 800$ cm/s and $\Theta \simeq 470^{\circ}$C this gives heat transfer rates of about 0.2 cal cm^{-2}s^{-1} which is about $^{1}/_{7}$ of the value estimated for the radiation.

The deflection of the flames will tend to increase slightly the radiation exchange factor between the flames and the sides of the neighbouring tank and reduce the flame emissivity. The combined effect is not necessarily to increase the radiation to the exposed tank. In view of the smallness of the increase in the heating by convection no allowance is made for any change in the heating rate of the tank walls due to a wind. The heating of the roof, however, may well be increased substantially in these strong winds.

4. The temperature of the tank wall

If the thermal resistance of the tank wall is neglected the equation governing the average temperature rise Θ_w above ambient of the tank wall is

$$I = \dot{m}''c\,\Theta_w + w\rho_s c_s \frac{d\Theta_w}{dt} + \alpha(\Theta_w - \Theta_g) \tag{26}$$

where Θ_g is the temperature rise of the vapour in the tank

ρ_s is the tank wall density

c_s is the tank wall specific heat

It is assumed that initially the water and the tank wall are at the ambient temperature 20°C.

The solution to equation (26) in the absence of any additional relation governing the value of Θ_g lies between the solution omitting the last term i.e. putting

$$\Theta_s = \Theta_w$$

and the solution obtained by putting

$$\Theta_g = 0$$

for which the solution is readily shown to be

$$\theta_w = \frac{I}{\dot{m}''c + \alpha}\left[1 - e^{-\left(\frac{\dot{m}''c+\alpha}{\omega\rho_s c_s}\right)t}\right] \qquad (27)$$

for a flow of 0.17 Imp. gal $ft^{-2} min^{-1}$ and $\alpha \simeq 5 \times 10^4$ c g s units $\dot{m}''c$ is more than an order of magnitude larger than α which can therefore be neglected for all practical flow rates.

The value of the time constant $\frac{\omega\rho_s c_s}{\dot{m}''c+\alpha}$ is less than 1 minute so for a long exposure with water cooling the tank wall may be regarded as effectively at its quasi-steady temperature

$$\theta_w = \frac{I}{\dot{m}''c + \alpha} \qquad (28)$$

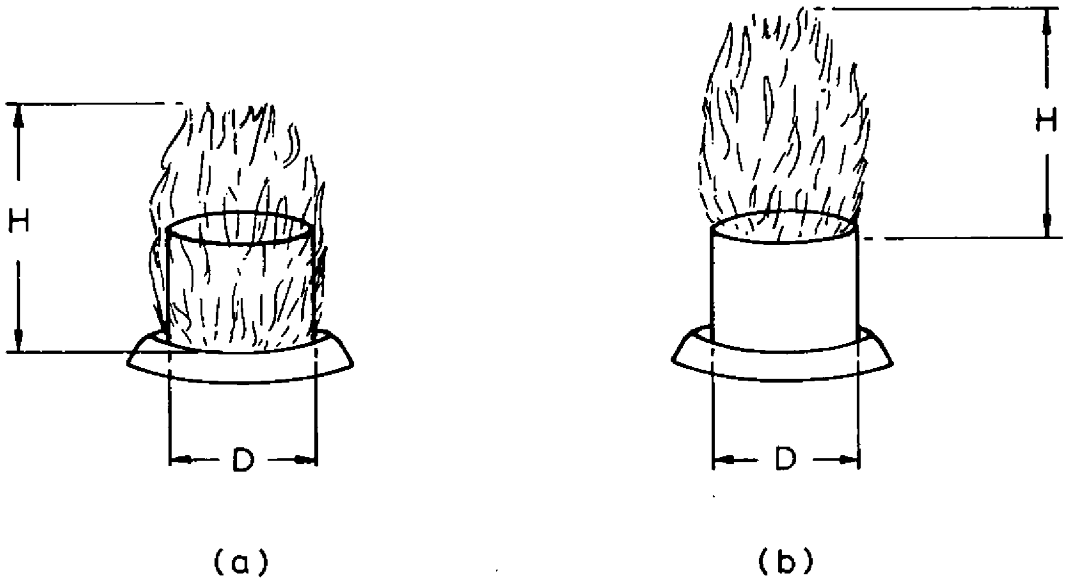

FIG. 1. FLAMES IN BUND AND TANK

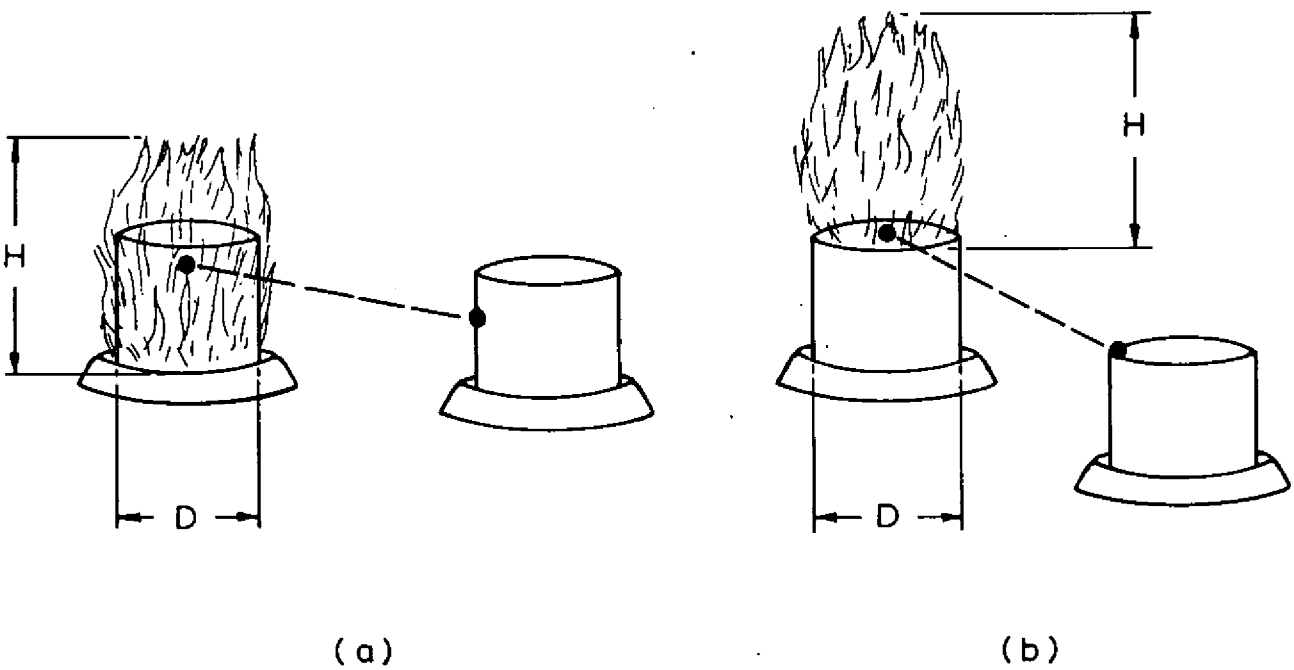

FIG. 2. RADIATION ON WALL AND ROOF

1/6444 F.R.609

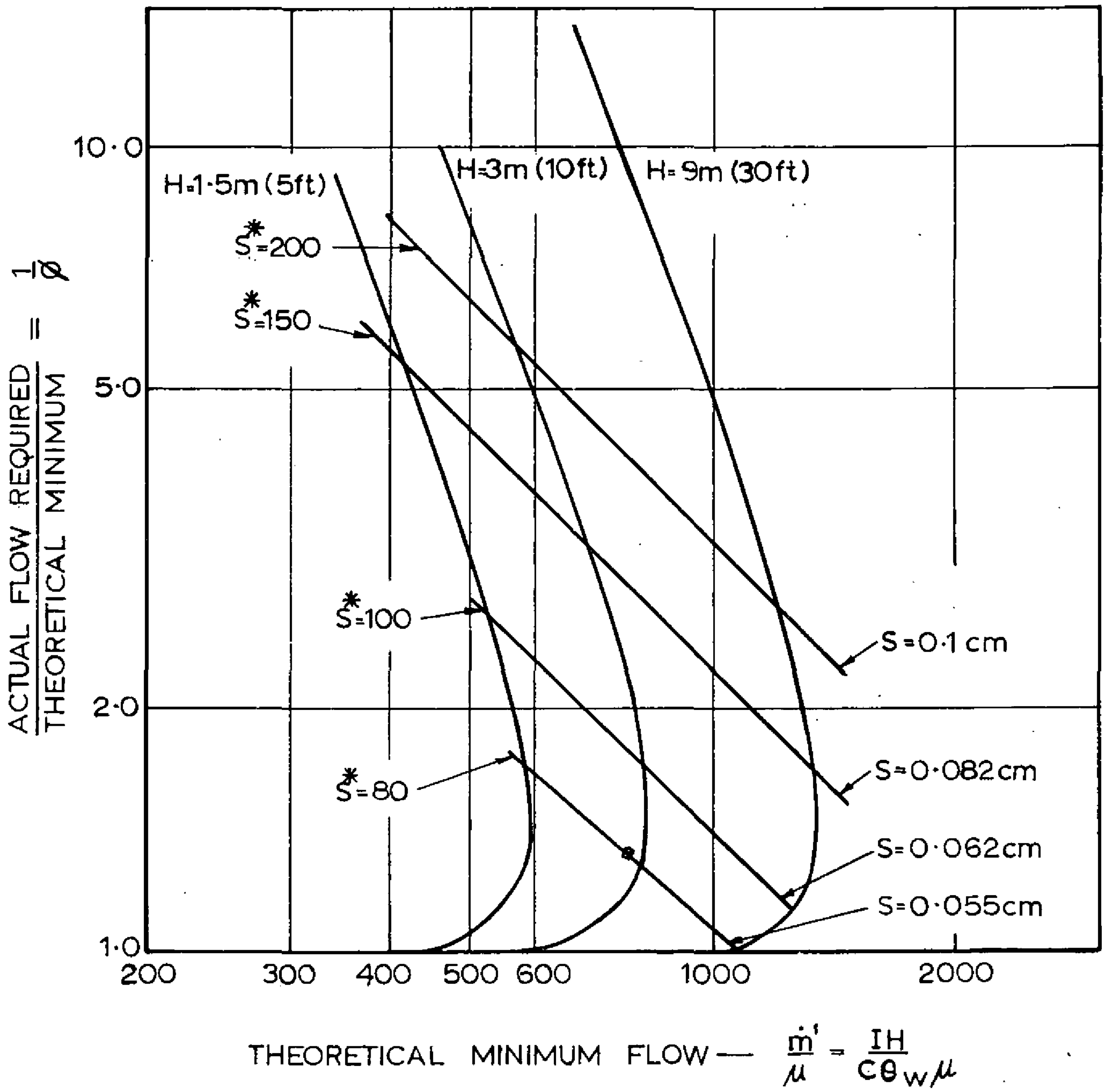

FIG.3. REQUIRED FLOW RATE FOR THIN FILM

1/6445 F.R.609