FSS 000180 R

UNCLASSIFIED

AEEW - M 2321

1991

A CONSIDERATION OF SOME SIMPLE MODELS FOR ASSESSING HEAT TRANSFER TO OBJECTS ENGULFED IN POOL FIRES

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MARCH 1985

SC21

SUMMARY

Two simple models for assessing the heat flux into objects engulfed in pool fires are considered. The two models differ in that the first models radiation from the fire as coming from a solid black surface surrounding the object while the second represents the flames as a uniform medium of finite thickness. The readings from thermocouples measuring flame temperature is considered using the two models and it is shown that the measured temperature may differ significantly from the true flame temperature. A simple design of instrument for measuring the effective black body radiation temperature is discussed.

CONTENTS

FIGURES

1 INTRODUCTION

Many structures and objects have to be designed so that they can withstand a hydrocarbon pool fire. Building structures, for example, must retain their strength, fire walls must provide adequate insulation, and fuel tanks must have relief valves of adequate size to prevent pressurisation. Nuclear transport flasks, in particular, are required by the IAEA regulations (Reference 1) to be capable of withstanding a pool fire of half an hour duration without releasing any radioactive material. It is important therefore that relatively simple methods exist for assessing the likely heat flux into any given object of arbitrary geometry in a pool fire.

This paper describes two fairly simple models, suggests typical values for the various parameters, and shows how these models can be used to interpret experimental measurements and design improved instrumentation.

2 THE 'TWO TEMPERATURE' MODEL

In the first fire model that will be considered it is asssumed that, for radiation heat transfer, the fire can be represented as a solid surface enclosing the test object, similar to a furnace. Convection heat transfer is assumed to occur over all external surfaces using a uniform convection coefficient. This model has been used widely in the assessment of temperatures in transport flasks during fire tests. The emissivity of the flames is often assumed to be less than unity. The IAEA regulations, for example, stipulate a flame emissivity of 0.9. In this simple model if the surface surrounding the object is given an emissivity which is less than unity, then direct radiation incident upon the object will be a fraction ϵ of that from a black body. Depending upon the assumed geometry of the surrounding surface, however, radiation emitted from the 'fire' may also be incident upon itself and some of this will be reflected onto the test object. Similarly some of the radiation emitted from the object will be reflected from the fire surface back onto itself. The magnitude of the radiation incident upon the test object, and its distribution, will therefore depend upon the geometry of the 'fire' surface. The heat flux into the object is thus a function of the assumed geometry of the 'fire' surface, an undesirable situation. In practice virtually none of the radiation emitted from the object will be reflected by the fire back onto itself.

These problems can be avoided by modelling the 'fire' surface as a black body (i.e. $\epsilon = 1.0$) since then no radiation will be reflected. The emissivity of the fire can still be represented, however, by fixing the 'fire' surface at a temperature T_r given by:

$$T_r^4 = \epsilon_a T_a^4 + \epsilon_f T_f^4 \qquad \ldots 1$$

The term involving T_a, which generally is insignificant, corresponds to radiation from the environment outside the fire which is incident upon the test object. If the radiative properties of the flames are assumed to be independent of wavelength (i.e. to be 'grey') then the effective emissivity of the ambient, ϵ_a, will be given by:

$$\epsilon_a = 1-\epsilon_f \qquad \ldots 2$$

The effective radiation temperature, T_r, will be less than the flame temperature, T_f. An emissivity of 0.9, for example, reduces a flame temperature of 800°C to an effective radiation temperature of 772°C. For convection heat transfer the real flame temperature is still appropriate and hence two different source temperatures are used to model heat transfer to surfaces inside the fire. The heat flux into any surface, ignoring any radiation from other surfaces of the test object, will be given by:

$$q = F\epsilon_s\sigma\,(T_r^4 - T_s^4) + h\,(T_f - T_s) \qquad \ldots 3$$

Where F is the view factor of the 'fire' surface from the surface being considered.

It is interesting to note that if an object reaches thermal equlibrium with the fire (i.e. q=o) then the surface temperature will be given by

$$T_s = \left(T_r^4 + \frac{h}{F\epsilon_s\sigma}\,(T_f - T_s)\right)^{\frac{1}{4}} \qquad \ldots 4$$

Since the T_r term is to the fourth power it dominates the RHS of this expression and the surface temperature, T_s, will be almost equal to T_r, the effective radiation temperature and not the true flame temperature T_f.

3 THE 'UNIFORM PROPERTY' MODEL

In the previous model the radiation from the fire was modelled as being emitted only from a surface outside the test body. Emission of radiation from flames within features such as fin cavities was therefore not represented and neither was the effect of variations in flame thickness. The second model to be considered overcomes both these inadequacies.

In this model the flames are represented as an absorbing and emitting medium of uniform temperature. In order to make the model as simple as possible the flames are assumed to be non-scattering and the radiation properties are assumed to be independent of wavelength (i.e. the flames are 'grey'). The absorption coefficient and emission coefficient are therefore equal and the radiation from the flames is a function only of this coefficient, K, the flame temperature, T_f, and the assumed geometry of the fire. Heat transfer by convection is again modelled by a uniform convection coefficient. The flame temperature used to model convection, T_f, in this model is the same temperature as is used to model radiation.

The radiation incident upon a surface is calculated by integrating over the field of view of the surface. Consider for example the simple plane geometry shown in Figure 1 in which the fire is assumed to be of uniform thickness and infinite in extent. The heat flux into the surface of the test object will be given by:

$$q = \sigma\epsilon_s\ (\epsilon_f T_f^4 - \epsilon_a T_a^4 - T_s^4) + h\ (T_f - T_s) \qquad \ldots 5$$

where ϵ_f and ϵ_a are effective emissivities and

$$\epsilon_f + \epsilon_a = 1 \qquad \ldots 6$$

If there were no absorption of radiation the view factor of the ambient from the surface would be given by:

$$F = \int_A \frac{\cos^2\theta}{\pi r^2}\, dA \qquad \ldots 7$$

$$= \int_{\theta=o}^{\theta=\frac{\pi}{2}} \frac{\cos^2\theta}{\pi r^2}\ 2\pi r\sin\theta \cdot \frac{r d\theta}{\cos\theta} \qquad \ldots 8$$

$$= \int_{\theta=o}^{\theta=\frac{\pi}{2}} \sin 2\theta\ d\theta = 1 \qquad \ldots 9$$

With absorption this will be reduced to an effective view factor F' given by

$$F' = \int_{\theta=o}^{\theta=\frac{\pi}{2}} \sin 2\theta e^{-Kr} d\theta \qquad \ldots 10$$

$$= \int_{\theta=o}^{\theta=\frac{\pi}{2}} \sin 2\theta e^{-Kd\sec\theta} d\theta \qquad \ldots 11$$

Thus the effective emissivity ϵ_a is given by:

$$\epsilon_a = \frac{F'}{F} = \int_{\theta=o}^{\theta=\frac{\Pi}{2}} \sin 2\theta e^{-Kd\sec\theta} d\theta \qquad \ldots 12$$

The value of ϵ_f, derived from equations 6 and 12, is shown as a function of the product Kd in Figure 2. It should be noted that the value of the effective fire emissivity, ϵ_f, will not in general remain constant over the surface of the test object, even when a uniform flame thickness is assumed, but will vary with position depending upon the geometry of the object and the assumed geometry of the fire.

4 TYPICAL PARAMETER VALUES

The question of what values for the various parameters to use in the models is a topic of considerable debate. This is partly due to the variable nature of pool fires. The heat flux to the test object depends not only upon the geometry of the object, pool and any wind shielding but also upon the fuel being used and the weather conditions (in particular the wind). Even under identical conditons heat fluxes may not be reproducible since the fire may have several stable 'modes' due to chimney effects and may switch from one mode to another at random times. The parameters required to model particular fires may therefore vary widely. Measured flame temperatures typically vary from around 800°C to 1200°C for example (Reference 2) which implies a variation in radiation heat fluxes greater than a factor of 3.

For design purposes it would seem appropriate to assume a fairly severe fire and a flame temperature of 1100°C is typical of many such fires (Reference 2). Measured heat fluxes into plane surfaces are typically of the order of $150kW/m^2$ which corresponds an effective radiation temperature (for the 'two temperature' model) of around 1000°C. Very few measurements of the absorption/emission coefficient of flames in pool fires have been reported so considerable uncertainty exists. The value of K is generally considered to be of the order of $1m^{-1}$ however.

Many experimentally measured values of the convection coefficient exist (Reference 2) but it is considered that most of these measurements are subject to considerable error. The problem in measuring the convection coefficient is that heat transfer in pool fires is generally dominated by radiation and separating this, which is itself difficult to measure, from the effect due to convection can lead to significant errors. It can be argued, however, that since convection heat transfer is not dominant, errors in the convection coefficient will not be significant.

From films of pool fires the gas velocity in the flames has been measured to be between 5 and 10m/s. If these velocities are applied to standard heat transfer correlations (References 3 and 4) the resulting convective heat transfer coefficients are predicted to lie between 10 and 15 $W/m^2°C$.

5 INSTRUMENTATION READINGS

A wide variety of instrumentation has been used to measure various properties and parameters of pool fires, often with only limited success. The most common instrument is a plain thermocouple used to measure the flame temperature. In this section an analysis is presented of the expected readings from a thermocouple in a pool fire, based upon the two models described previously. The expected readings from other instrumentation, such as heat flux meters, can be readily determined using a similar analysis.

Consider a thermocouple, parallel to the plane surface of the test object, immersed in the flames surrounding the object (Figure 1). The thermocouple may be considered as an infinitely long cylinder although the tip of the thermocouple may be better represented as a sphere.

If the 'two temperature' model is used then the view factor of both the assumed fire surface and the surface of the test object will be 0.5. This is true irrespective of whether the thermocouple is modelled as a cylinder or sphere and irrespective of the distance of the thermocouple from the test object. The temperature measured by the thermocouple (i.e. its equilibrium temperature) will be given by:

$$\epsilon_c \sigma T_c^4 = \epsilon_c \left[\frac{1}{2}\left[\epsilon_s \sigma T_s^4 + (1-\epsilon_s)\sigma T_r^4\right] + \frac{1}{2}\sigma T_r^4\right] + h(T_f - T_c) \quad \ldots 13$$

$$T_c = \left[\frac{1}{2}\left(\epsilon_s T_s^4 + (2-\epsilon_s)\, T_r^4\right) + \frac{h}{\epsilon_c \sigma}(T_f - T_c)\right]^{1/4} \quad \ldots 14$$

The convection term on the right hand side is dominated by the radiation terms, which are independent of ϵ_c, and the measured temperature is therefore insensitive to the emissivity of thermocouple.

If, for example, a fire is represented by the following parameters:

T_f = 1373K (1100°C)
T_r = 1273K (1000°C)
T_s = 373K (100°C)
ϵ_s = 0.9
ϵ_c = 0.8
h = 10 W/m²°C

then, from equation 14, the temperature measured by the thermocouple will be 836°C which is significantly less than both the true flame temperature and the effective radiation temperature.

A more accurate representation can probably be obtained by using the 'uniform property' model. The equilibrium temperature of the thermocouple will then be given by:

$$\epsilon_c \sigma T_c^4 = G_s \epsilon_c \sigma T_s^4 + G_a \epsilon_c \sigma T_a^4 + G_f \epsilon_c \sigma T_f^4 + h(T_f - T_c) \quad \ldots 15$$

where G_s, G_a and G_f are factors which take into account the integrated view factors and effective emissivities and are related by the equation:

$$G_f + G_a + G_s = 1 \quad \ldots 16$$

If the flame has a thickness d and the thermocouple is a distance x from the surface of the test object then, if the thermocouple is assumed to be a sphere, G_a, G_f, and G_s will be given by:

$$G_s = \epsilon_s \int_0^{\frac{\pi}{2}} \frac{\sin\theta}{2} e^{-Kd\sec\theta} d\theta \quad \ldots 17$$

$$G_a = \int_0^{\frac{\pi}{2}} \frac{\sin\theta}{2} e^{-K(d-x)\sec\theta} d\theta + \epsilon_a \frac{(1-\epsilon_s)}{\epsilon_s} G_s \quad \ldots 18$$

where ϵ_a is given by equation (12)

$$G_f = 1 - G_a - G_s \quad \ldots 19$$

Thus, for example, if a fire and test object surface are modelled by the following parameters:

T_f = 1373K (1100°C)
T_s = 373K (100°C)
T_a = 293K (20°C)
ϵ_s = 0.9
ϵ_c = 0.8
K = $0.9m^{-1}$.
d = 1.0m
h = 10 $W/m^2 \cdot C$

Then the temperature measured by the thermocouple (i.e. its equilibrium temperature) will vary as a function of x, its distance from the test object, as shown in Figure 3. It can be seen that the maximum temperature (964°C), which is obtained in this case when the thermocouple is about mid-way between the test object and the outside of the flame, although greater than that predicted using the 'two temperature' model is still significantly less than the true flame temperature. If the thermocouple is modelled as a cylinder then G_s, G_a and G_f are given by:

$$G_s = \frac{\epsilon_s}{\pi} \int_0^{\frac{\Pi}{2}} \int_0^{\frac{\Pi}{2}} \cos\Phi e^{-Kx\sec\theta\sec\Phi} \, d\theta d\Phi \qquad ...20$$

$$G_a = \frac{1}{\pi} \int_0^{\frac{\Pi}{2}} \int_0^{\frac{\Pi}{2}} \cos\Phi e^{-K(d-x)\sec\theta\sec\Phi} d\theta d\Phi + \epsilon_a \frac{(1-\epsilon_s)}{\epsilon_s} G_s \qquad ...21$$

$$G_f = 1 - G_s - G_a \qquad ...22$$

It can be shown that these equations are equal to equations 17 to 19 and the resulting equilibrium temperature is hence the same as for a sphere.

6 INSTRUMENTATION DEVELOPMENT

It has been shown that the temperature measured by a bare thermocouple is a function of the position of the thermocouple and the temperature of the test object and that the measured temperature may correspond to neither the true flame temperature nor the effective radiation temperature. Radiation shielding can be used to increase the temperature measured by a thermocouple but the shielding needs to be very efficient to eliminate the normally dominating effect of radiation. The measured temperature will probably again correspond to neither the true flame temperature nor the effective radiation temperature.

A simple instrument which may give more meaningful measurements than a bare thermocouple consists of a small plate, insulated on all but the front face, with a thermocouple measuring its equilibrium temperature. If this instrument is placed at the surface of the test object, facing outwards, then the radiation incident upon it will be the same as is incident upon the test object. Using the 'two temperature' model the equilibrium temperature of the instrument will be given by:

$$\epsilon_c \sigma T_c^4 = \epsilon_c \sigma T_r^4 + h(T_f - T_c) \qquad ...23$$

which, using the parameters from the previous section, gives a temperature of 1007°C. This corresponds very closely to the effective radiation temperature, an even better estimate of which can be obtained by correcting the measurement for the effect of convection.

The same instrument can be used to determine the true flame temperature if it is positioned so that it is looking through the greatest possible depth of flame. In practice this position is probably on the edge of the fire facing inwards. If, for example, the 'uniform property' model is used with the parameters from the previous section then by looking through a 3m depth of flame the instrument would measure a temperature of 1091°C which is within 1% of the true flame temperature.

7 CONCLUSIONS

Two simple models have been presented for predicting heat fluxes to objects engulfed in pool fires. For objects with plain surfaces the simple 'two temperature' model will probably be adequate in most cases. For more complex geometries, especially those having re-entrant cavities, the 'uniform property' model is probably more appropriate.

These simple models have been used to study the expected readings from thermocouples used to measure flame temperature. This showed measured temperature to be significantly less than the true flame temperature and to be a function not only of the flame properties but also the position of the thermocouple and the temperature of the test object. A simple instrument for measuring effective radiation temperatures may provide more meaningful measurements which can be used in the calculation of the heat flux to the test object.

NOMENCLATURE

d flame thickness
F view factor
F' effective view factor
G radiation factor
h convection coefficient
K absorption/emission coefficient
q heat flux
ℓ distance through flame (see Figure 1)
T temperature
x distance of thermocouple from object surface
ϵ emissivity
σ Stefan's constant
θ angle from normal to surface (see Figure 1)
Φ angle from normal to surface in plane of cylinder axis

Subscripts

a ambient
c thermocouple
f flame
r effective radiation
s surface of test object

REFERENCES

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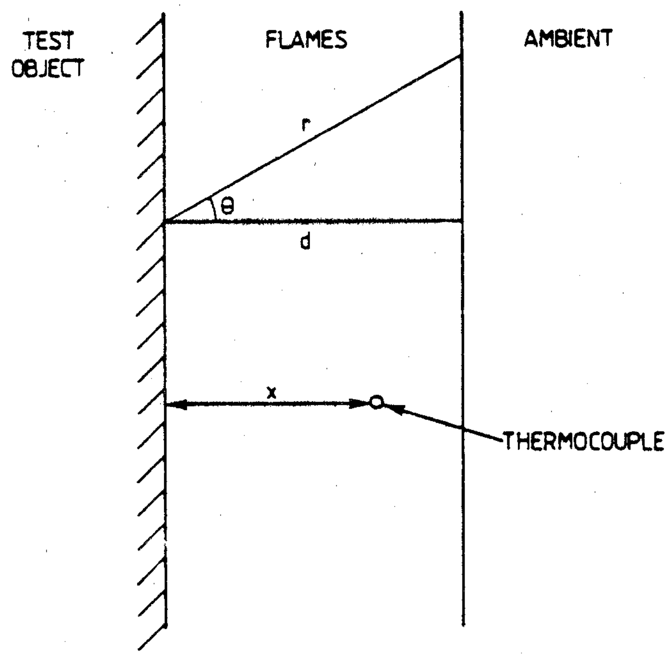

FIG.1 THE ASSUMED GEOMETRY OF THE FIRE

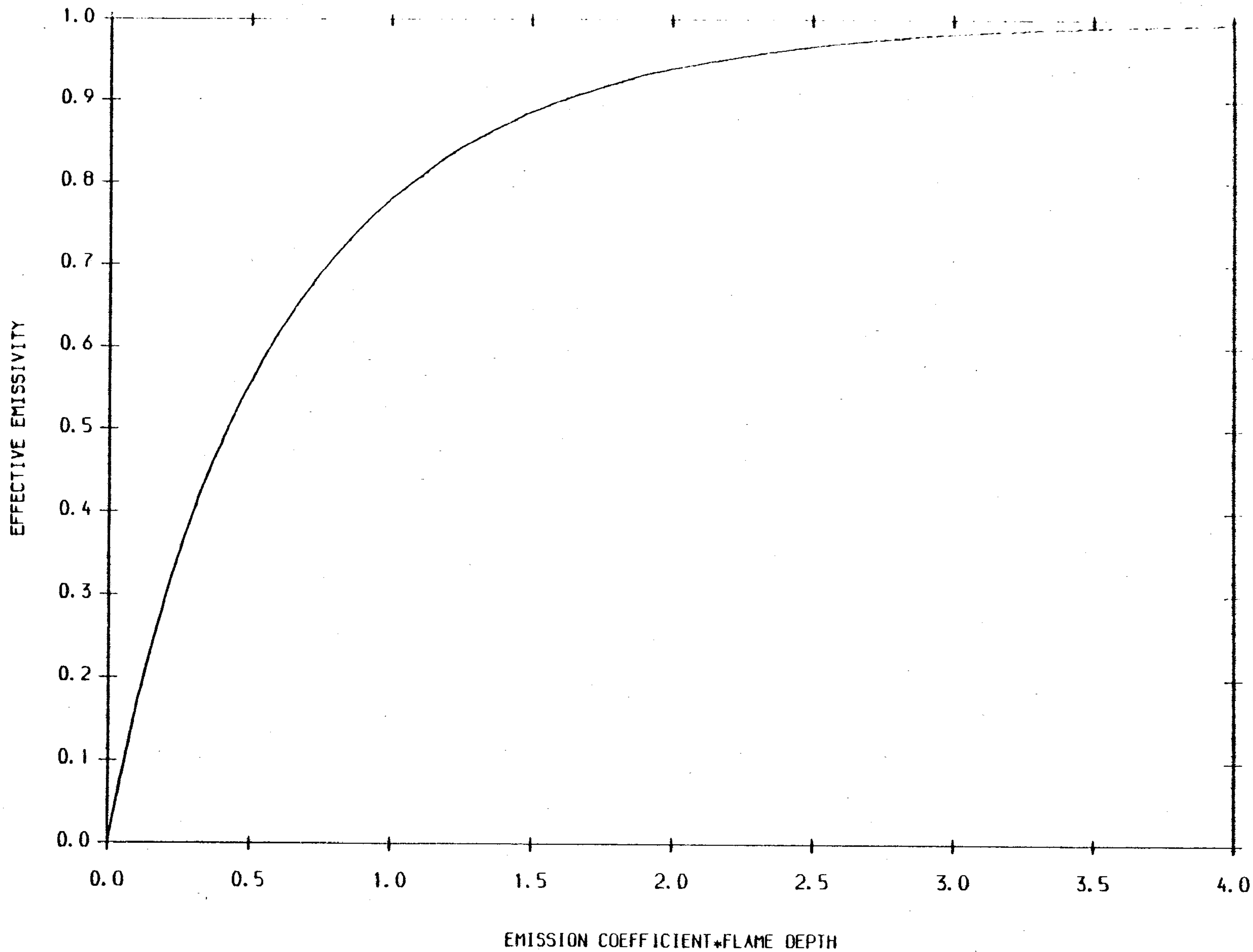

FIG. 2 VARIATION OF EFFECTIVE EMISSIVITY WITH FLAME DEPTH

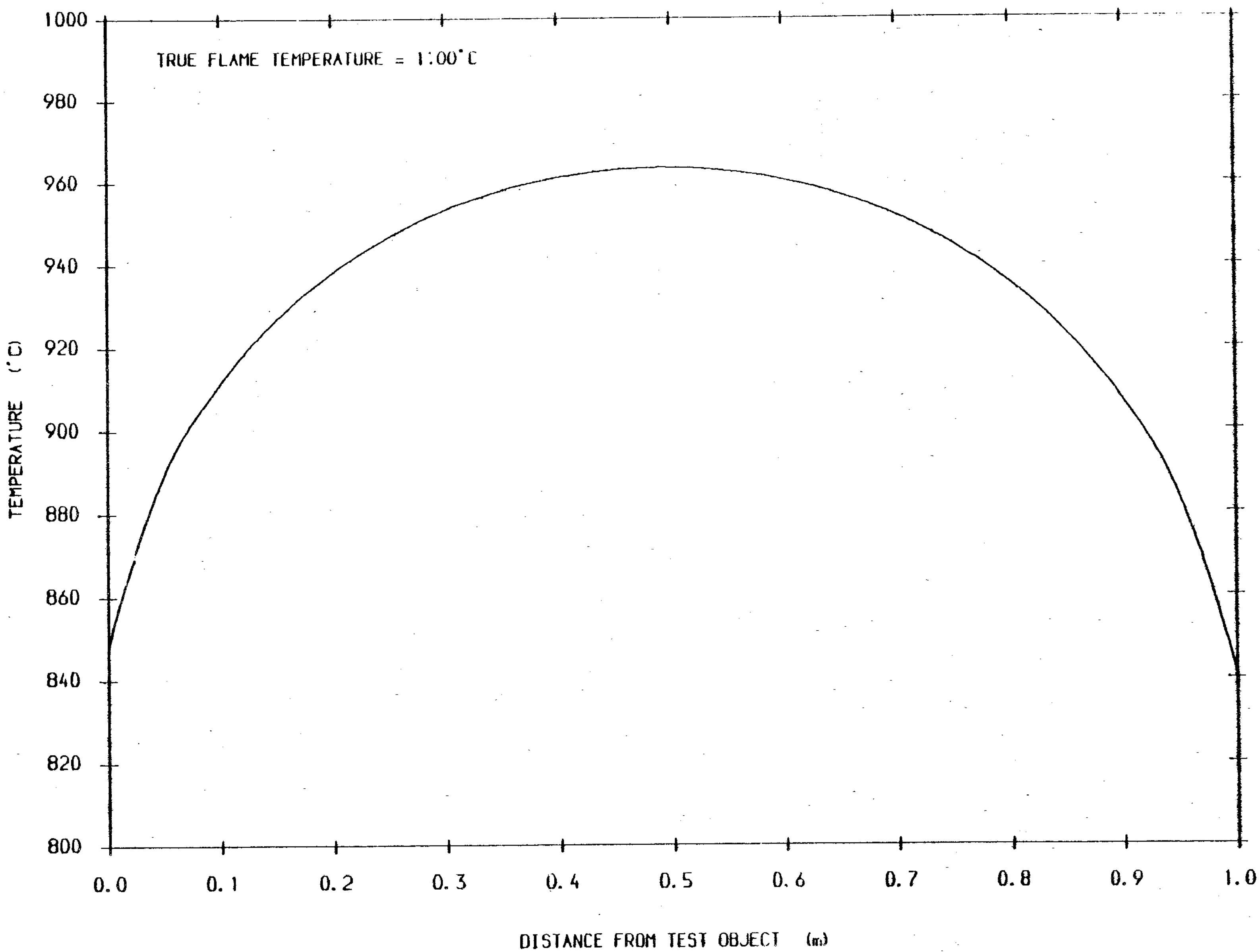

FIG. 3 TEMPERATURE MEASURED BY THERMOCOUPLE IN FIRE

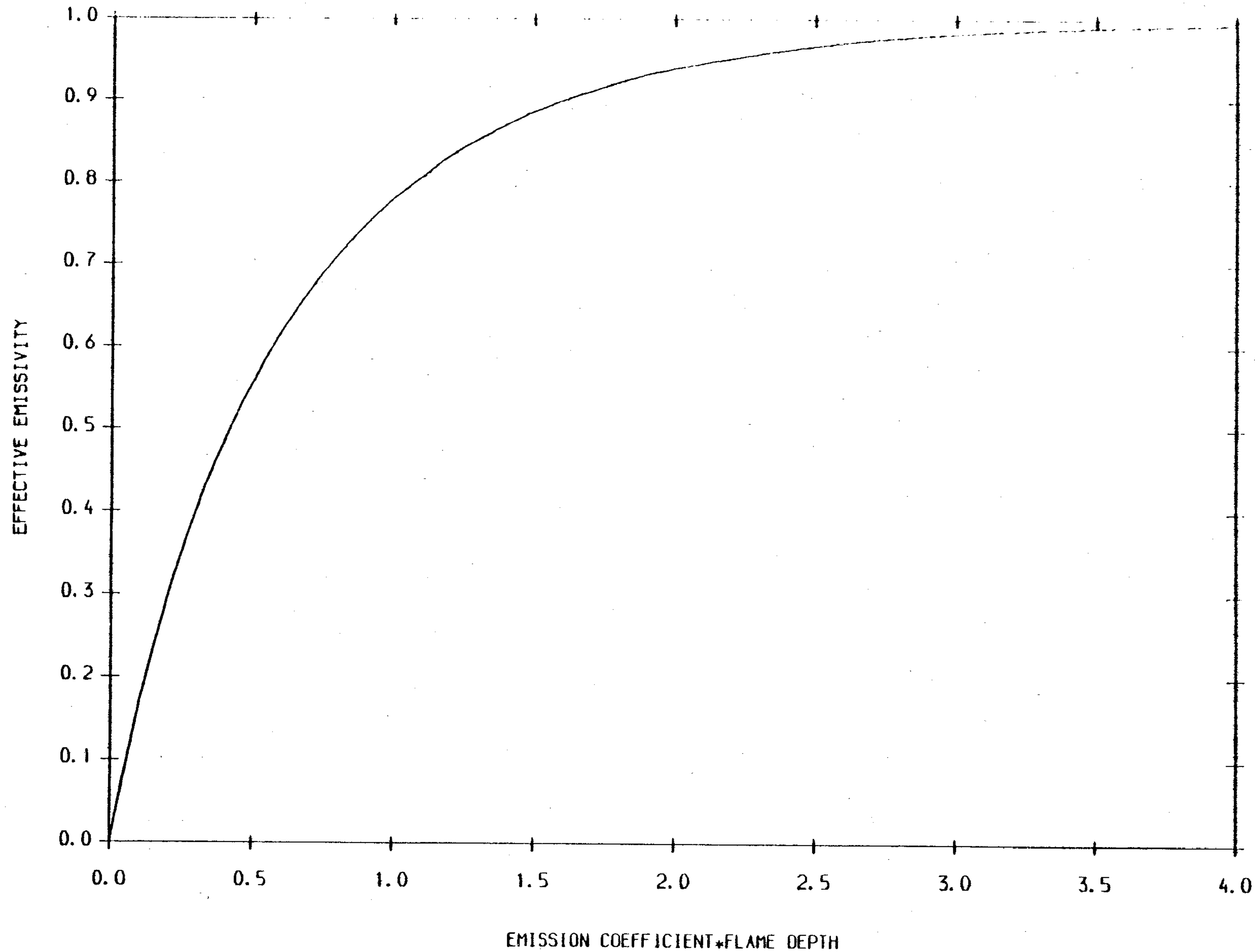

FIG. 2 VARIATION OF EFFECTIVE EMISSIVITY WITH FLAME DEPTH