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# DESIGN OF CONCENTRIC TUBULAR REACTOR FUEL ELEMENTS FOR UNIFORM COOLANT CONDITIONS

by

IAN M. BINNS
R. LICCIARDO

ABSTRACT

Concentric tubular reactor fuel element geometries to give equal coolant outlet temperatures are presented. Oscillations from tube to tube in thickness and temperatures generally occur but it is possible to eliminate them by choice of the centre element. This may be a fuel rod or a non-heat-producing rod with or without a surrounding annulus of fuel. The geometries and temperatures are dependent on the voidage and on a non-dimensional parameter equivalent to a Biot number based on the channel equivalent diameter.

## CONTENTS

## 1. INTRODUCTION

In design of nuclear fuel elements it is difficult to make compatible the temperature distribution, heat conduction, and surface heat transfer, so a trial-and-error method is commonly used to obtain a configuration of fuel elements involving concentric tubes to give a practical temperature distribution. However a more efficient method which would lead to a better understanding of the problem would be the evolution of a set of simultaneous equations whose solution for specified conditions would uniquely determine the geometry of the fuel element.

In the following analysis the specified condition is that the coolant channels should produce equal coolant outlet temperatures from equal inlet temperatures.

The analysis involves a study of the dimensions of individual tubes, relations between neighbouring tubes, and the effect of boundaries.

## 2. GENERAL EQUATION AND ITS SOLUTION

For the analysis the fuel elements, a series of concentric tubes, are identified as in Figure 1.

The following assumptions are made;

1. The fuel element material is homogeneous having uniform heat generation and thermal properties, both independent of temperature.
2. The model including its boundaries is axially symmetric, with constant heat flux at the boundaries.
3. Steady state conditions have been achieved.
4. There is negligible conduction longitudinally in the fuel elements.

Using the notation given in Section 7 the resulting general equation for the rate of increase in coolant temperature in the n-th channel is as follows:

$$\frac{dT_{fn}}{dl} = \frac{q}{m_n C_{pn}}\left[\frac{r_{mn}^2 - r_{m(n-1)}^2}{A_n/\pi} - 1\right] + \frac{2K}{m_n C_{pn}} \; \frac{1}{A_n/\pi}\left[a_{(n-1)}\left(T_{f(n-1)} - T_{fn}\right) + a_n\left(T_{f(n+1)} - T_{fn}\right)\right]. \quad (1)$$

where

$$r_{mn}^2 = a_n K\left[\frac{r_{on}}{h_{on}} + \frac{r_{in}}{h_{in}} - \frac{(r_{on}^2 - r_{in}^2)\left(\frac{1}{r_{on}h_{on}} + \frac{1}{r_{in}h_{in}}\right)}{2\ln\left(\frac{r_{on}}{r_{in}}\right)}\right] + \frac{r_{on}^2 - r_{in}^2}{2\ln\left(\frac{r_{on}}{r_{in}}\right)} \quad (2)$$

$$a_n = \frac{1}{\ln\frac{r_{on}}{r_{in}} + K\left(\frac{1}{r_{on}h_{on}} + \frac{1}{r_{in}h_{in}}\right)} \quad (3)$$

and

$r_{mn}$ is the radius at which the temperature gradient in the n-th fuel element is zero if surrounding coolant temperatures $T_{f(n+1)}$ and $T_{fn}$ are equal.

**NOTE**: The derivation of equation 1 is given in Appendix 1.

If the coolant temperatures are equal then

$$\frac{dT_{fn}}{dl} = \left[\frac{q}{m_n C_{pn}} \; \frac{r_{mn}^2 - r_{m(n-1)}^2}{A_n/\pi} - 1\right] = k \quad . \quad (4)$$

and this can be made equal for all values of n by design choice.

It is shown in Appendix 2 that for equal mass flow per unit area the successive values of $r_{mn}$ are given by

$$r_{mn}^2 = r_{m(n-1)}^2 + \frac{r_{in}^2 - r_{o(n-1)}^2}{C} \tag{5}$$

where C is the voidage defined as flow cross-section area divided by the sum of the flow and fuel cross-section areas.

For similar gas conditions in each channel the equivalent diameter De is made equal for all n.

Therefore

$$r_{in} = r_{o(n-1)} + \frac{De}{2} \tag{6}$$

and the dimensions of a tube bundle can be calculated using equations 2, 3, 5, and 6 assuming a known boundary.

A major simplification can be made by assuming the heat transfer coefficients $h_{on}$ and $h_{in}$ to be equal.

Equation 2 and 3 then give

$$r_{mn}^2 = r_{in}^2 X_n \left[ \frac{1 + r_{in} \frac{h}{K} \frac{X_n - 1}{2}}{1 + r_{in} \frac{h}{K} \frac{X_n \ln X_n}{X_n + 1}} \right] \tag{7}$$

where $X_n = \frac{r_{on}}{r_{in}}$

The solution can be made non-dimensional by making the following substitutions:

$$D = \frac{Deh}{K}, \quad R = \frac{rh}{K}, \quad A' = A\left(\frac{h^2}{K}\right).$$

The above equations now become

$$R_{in} = R_{o(n-1)} + \frac{D}{2} \tag{8}$$

$$R_{mn}^2 = R_{m(n-1)}^2 + \frac{1}{C}\left(R_{in}^2 - R_{o(n-1)}^2\right) \tag{9}$$

$$R_{mn}^2 = R_{in}^2 \; X_n \left[ \frac{1 + R_{in} \frac{X_n - 1}{2}}{1 + R_{in} \frac{X_n \ln X_n}{X_n + 1}} \right] \tag{10}$$

Equation 10 is transcendental and the following method of solution is suggested:

$$\text{Since} \quad fn\,(X_n) = \frac{1 + R_{in} \frac{X_n - 1}{2}}{1 + R_{in} \frac{X_n \ln X_n}{X_n + 1}}$$

which is approximately equal to one,

$$\text{let} \quad (X_n)_1 = \frac{R_{mn}^2}{R_{in}^2}.$$

Then

$$(X_n)_2 = \frac{(X_n)_1}{f[(X_n)_1]} \quad .$$

$$(X_n)_3 = \frac{(X_n)_1}{f[(X_n)_2]} \quad .$$

until the required accuracy is obtained.

### 2.1 Surface and Maximum Fuel Temperatures

The surface temperature is given relative to the fluid bulk temperatures in equations (v) and (vi) of Appendix 1. Substituting $r_{mn}$ for $r_{pn}$ the non-dimensional equations are

$$\frac{T_{in} - T_f}{\frac{D_e q}{h}} = \left[\frac{1}{2}\ \frac{R_{mn}^2}{R_{in}^2} - 1\right] \frac{R_{in}}{D} \quad . \tag{11}$$

$$\frac{T_{on} - T_f}{\frac{D_e q}{h}} = \frac{1}{2}\left[1 - \frac{R_{mn}^2}{R_{in}^2}\ \frac{1}{X_n^2}\right] \frac{X_n R_{in}}{D} \quad . \tag{12}$$

These equations in the above form can be used in conjunction with the geometric analysis.

The maximum fuel element temperature occurs at $r = r_{pn} = r_{mn}$ and from equations (iii) and (iv) of Appendix 1 ,

$$\frac{T_m - T_f}{\frac{D_e q}{h}} = \frac{1}{4}\ \frac{R_{in}^2}{D}\left[\left(\frac{R_{mn}^2}{R_{in}^2}\right)\ln\left(\frac{R_{mn}^2}{R_{in}^2}\right) - \frac{R_{mn}^2}{R_{in}^2} + 1\right] + \frac{T_{ni} - T_{fn}}{\frac{D_e q}{h}} \quad . \tag{13}$$

The limiting surface temperature parameters derived in Appendix 3 both equal $\frac{R_{on} - R_{in}}{2D}$ .

The similar limit on the maximum fuel element parameter is found to equal

$$\frac{R_{on} - R_{in}}{2D}\left[1 + \frac{R_{on} - R_{in}}{4}\right].$$

## 3. DISCUSSION OF THE ANALYSIS

### 3.1 The Equality of Coolant Temperatures

This results from the assumption of uniform inlet conditions. If there is a coolant temperature gradient in the radial direction, the effect of the temperature terms in equation 1 is to adjust the coolant temperature increase until the temperature is equal to that in the two neighbouring channels.

### 3.2 Coolant Mass Flow Per Unit Area

It is desirable to have the pressure drop equal for each channel and this could be very closely achieved by specifying equal equivalent diameters. Secondary variations result from surface temperature variations.

As a result of this, or by using entrance constrictions, equal mass flows per unit area can be assumed.

### 3.3 Equality of Heat Transfer Coefficients

In the simplification of equations 2 and 3 heat transfer coefficients at the inside and outside surfaces of the fuel element are assumed equal. It seems generally accepted that the heat transfer coefficient at the outer boundary of an annular coolant channel is that given by the normal tube equation using the equivalent diameter while for the inner boundary the equation is usually modified by a factor depending on the curvature of the surface. No consistently accurate equation seems to be available in the literature but a simple illustrative example is that given by Monrad and Pelton.*

For the outer boundary on the inside surface of tubes Monrad and Pelton justify the following:

$$Nu = 0.023\ Re^{0.8}\ Pr^{0.4} \quad .$$

and for the inner boundary on the outside surface of tubes,

$$Nu = 0.02\left(\frac{d_{outer}}{d_{inner}}\right)^{0.53} Re^{0.8}\ Pr^{0.4}$$

It seems likely that for gas–cooled systems $\frac{d_{outer}}{d_{inner}}$ will be within the range 1.0 to 2.0 yielding ratios of heat transfer coefficients from 1.0 to 1.255. Except for the centre the ratio should be very close to 1.0 and so it is reasonable to assume that the heat transfer coefficients are equal.

## 4. RESULTS OF CALCULATIONS

Calculations were made for voidages of 15, 30, and 45 per cent. and values of D from 0.01 to 100.0, to determine the geometric configurations and corresponding temperatures. The centre boundary consisted of a channel of a non–dimensional diameter equal to D. The results indicated in Figure 2 showed oscillations in tube thickness about a mean of $\frac{1-C}{C} \cdot \frac{De}{2}$ as predicted by the approximate analysis in Appendix 3. The magnitude of these oscillations increased with increasing D and decreasing voidage. Similar results are obtained for the temperatures, Figures 3 to 5.

It was possible to eliminate these thickness variations by determining the centre boundary values of $r_{mo}$ and $r_{oo}$ to give a tube thickness of $\frac{1-C}{C} \cdot \frac{De}{2}$. Eliminating $r_{mn}$ from equations 10 and 11 and putting $n = 1$ and $r_{o1} - r_{i1} = \frac{1-C}{C} \cdot \frac{De}{2}$ .

$$r_{mo}^2 = r_{i1}^2\ X_1 \left[\frac{1 + \frac{r_{i1}b}{K}\ \frac{X_1 - 1}{2}}{1 + \frac{r_{i1}b}{K}\ \frac{X_1 \ln X_1}{X_1 + 1}}\right] - \frac{r_{i1}^2 - r_{oo}^2}{C} \quad .$$

and

$$r_{i1} = r_{oo} + \frac{De}{2} \quad .$$

$$r_{o1} = r_{oo} + \frac{1}{C}\ \frac{De}{2} \quad .$$

Hence this gives a direct relation between $r_{mo}^2$ and $r_{oo}^2$ in the form shown in Figure 8. The portion AB of this curve corresponding to $r_{mo} > r_{oo}$ has no physical significance since for equal coolant temperatures $r_m$ is the radius at which the temperature gradient is zero within the fuel material. (Refer equation 4).

The point B corresponds to $r_{mo} = r_{oo}$; that is the point of zero temperature gradient in the element occurs at the surface and therefore there is no heat transfer at that surface. The region within $r_{mo}$ can hence be considered a non–heat–producing rod.

It follows that the portion BC of the curve corresponding to $r_{mo} < r_{oo}$ gives solutions for a non–heat–producing rod surrounded by an annulus of fuel, and point C where $r_{mo} = 0$ gives an all fuel rod.

The portion CD where $r_{mo}^2$ is negative gives imaginary configurations, point D is equivalent to point C, and DE is equivalent to BC.

* Lapides, M.E. and Goldstein, M.B. (1957)
Heat Transfer Source File Data APEX 425

The real solutions, as shown in Figure 6 where curves have been plotted for various voidages, therefore include the following cases

(a) a central non-heat-producing rod, i.e. $r_{mo} = r_{oo}$ .

(b) a central fuel rod, i.e. $r_{mo} = 0$ , $r_{oo} > 0$ ,

(c) a central non-heat-producing rod of radius $r_{mo}$ surrounded by an annulus of fuel,

i.e. $r_{mo} \neq 0$

$r_{mo} < r_{oo}$

Case (a) occurs only in the region of small sizes.

The corresponding tube temperature behaviour is indicated in Figure 7 for a voidage of 30 per cent. and D = 1.0. The temperatures including those of the centre element ($n = o$) are presented for a range of sizes of the centre element beginning at a fuel rod where $\frac{r_{mo}}{De/2} = 0$ and $\frac{r_{oo}}{De/2} = 2.19$ up to a fuel annulus $\frac{r_{oo}}{De/2} = 15$ surrounding a non-heat-producing rod. The corresponding values of $r_{mo}$ can be obtained from Figure 6. All temperatures are presented with respect to the bulk coolant temperature in a non-dimensional form.

These boundary configurations gave no thickness variations for the series of concentric tubes.

## 5. DISCUSSION OF RESULTS OF CALCULATIONS

The tube cluster involving a centre channel represents the general case of concentric tubular fuel elements of constant coolant channel equivalent diameter and uniform voidage to give uniform coolant conditions. The results for this case indicate the significance of the non-dimensional equivalent diameter $D = \frac{De\,h}{K}$ ; to achieve the above specification at high values of D, large oscillations from one tube to the next in thickness and temperature must result. These oscillations decrease significantly for lower values of D and also for higher voidages.

The variations of the surface temperatures seem to invalidate the assumption that the heat transfer coefficients are the same in each channel since it is necessary to include a temperature term in the heat transfer coefficient equation. The variation in the coefficient would not amount to more than approximately 5 per cent. for gas-cooled systems which operate at values of D from 0.1 to 1.0 but at low voidages. High values of D are usually associated with fast reactors having high voidages, and therefore giving lower temperature variations.

For the boundary conditions for clusters of tubes of uniform thickness, since $r_{mo}$ must be positive and within the fuel material, any solutions involving negative or unreal values of $r_{mo}$ or values of $r_{mo} > r_{oo}$ have no physical significance. It should also be noted that no attempt was made to start with defined outside boundaries because of the possibility of obtaining unreal configurations at the centre.

Only the solutions having physical significance are presented in the results (Figure 6). They are in two separate ranges and except at low voidages and high values of D one of these ranges would not be practical because of the very small sizes involved. Within this range the solution where $r_{mo} = r_{oo}$ , that is a central non-heat-producing rod, would require some modification to the heat transfer assumption since, unlike the other channels, heat is only being transferred at one surface.

The more practical range begins with values of $r_{oo}$ a little less than the tube thickness of $\frac{1-C}{C} \cdot \frac{De}{2}$ and $r_{mo} = 0$ . As $r_{oo}$ increases $r_{mo}$ becomes finite and the difference between them, that is the thickness of the fuel annulus surrounding the central non-heat-producing rod, asymptotically approaches $\frac{1-C}{C} \cdot \frac{De}{2}$ .

The temperatures do not have the oscillatory effect present in the general case and the temperature drops across the fuel elements are markedly decreased.

It is evident from all the results that the larger the tube the closer it approaches the limiting geometry and temperatures.

## 6. CONCLUSIONS

Concentric tubular reactor fuel elements to give equal coolant outlet temperatures can be designed if the voidage and Biot number based on the channel equivalent diameter are known.

If the oscillations from tube to tube in thickness and temperature, which generally result, cannot be tolerated, they can be eliminated by specifying a particular centre fuel element.

## 7. NOTATION

| | |
|---|---|
| a | as defined in equation 3 |
| A | area of coolant channel |
| $A^1$ | non-dimensional area $= A\left(\frac{h}{k}\right)^2$ |
| c | coolant flow voidage with respect to fuel element area |
| $c_1$, $c_2$ | constants of integration |
| $C_p$ | specific heat at constant pressure of coolant |
| d | diameter of tubes |
| D | non-dimensional equivalent diameter $= \frac{De\ h}{k}$ |
| $D_e$ | equivalent diameter of coolant channel |
| h | heat transfer coefficient |
| k | as defined in equation 4 |
| K | thermal conductivity of fuel element material |
| l | axial length |
| m | mass flow of coolant per unit area |
| n | number of channel or fuel element |
| Nu | Nusselt number |
| Pr | Prandtl number |
| q | power density in fuel element material |
| r | radius |
| R | non-dimensional radius $= \frac{rh}{k}$ |
| Re | Reynolds number |
| T | temperature |
| X | ratio of inner to outer tube radii |
| $\tau$ | fuel element tube thickness. |

**Subscripts**

| | |
|---|---|
| f | corresponding to coolant |
| i | value at inner radius of fuel element tube |
| m | value at radius defined in equation 4 |
| n | corresponding to the n-th fuel element tube or channel |
| o | value at outer radius of fuel element tube |
| p | value at radius where $\frac{dT}{dr} = o$ within fuel material. |

## APPENDIX 1

### DERIVATION OF THE GENERAL EQUATION

The longitudinal temperature equation is given by equating the heat generation in the fuel element to that gained by the coolant.

Thus

$$q\,\pi\,(r_{pn}^2 - r_{in}^2 + r_{on}^2 - r_{p(n-1)}^2) = m_n\,\pi\,(r_{in}^2 - r_{o(n-1)}^2)\,C_p\,\frac{d\,T_{fn}}{dl} \quad \text{(i)}$$

Therefore

$$\frac{d\,T_{fn}}{dl} = \frac{q}{m_n\,C_{pn}}\left[\frac{r_{pn}^2 - r_{p(n-1)}^2}{r_{in}^2 - r_{o(n-1)}^2} - 1\right] \quad \text{(ii)}$$

The steady state radial temperature distribution for the axially symmetric case is given by the solution of the relevant Poisson's equation as follows:

$$T = \frac{-q}{4K}\,r^2 + C_1 \ln r + C_2 \quad , \quad \text{(iii)}$$

where for the n–th tube,

$$C_{1n} = \frac{T_{on} - T_{in} + \frac{q}{4K}(r_{on}^2 - r_{in}^2)}{\ln \frac{r_{on}}{r_{in}}} \quad ,$$

and $C_2$ is given in terms of temperature and radius at either surface.

Now at $r_p$

$$\frac{dT}{dr} = 0 \quad ,$$

and therefore

$$r_{pn}^2 = \frac{2K}{q}\,C_{1n} \quad \text{(iv)}$$

Equating heat transfer and heat generation at the surface,

$$2 r_{on} h_{on} (T_{on} - T_{f(n+1)}) = q\,(r_{on}^2 - r_{pn}^2) \quad , \quad \text{(v)}$$

and

$$2 r_{in} h_{in} (T_{in} - T_{fn}) = q\,(r_{pn}^2 - r_{in}^2) \quad . \quad \text{(vi)}$$

Eliminating the surface temperatures from equations (iii) to (vi),

$$r_{pn}^2 = a_n\,\frac{2K}{q}(T_{f(n+1)} - T_{fn}) + r_{mn}^2 \quad ,$$

where

$$a_n = \frac{1}{\ln\left(\frac{r_{on}}{r_{in}}\right) + K\left(\frac{1}{r_{on}h_{on}} + \frac{1}{r_{in}h_{in}}\right)} \quad ,$$

and

$$r_{mn}^2 = a_n K\left[\left(\frac{r_{on}}{h_{on}}\right) + \left(\frac{r_{in}}{h_{in}}\right) - \frac{(r_{on}^2 - r_{in}^2)\left(\frac{1}{r_{on}h_{on}} + \frac{1}{r_{in}h_{in}}\right)}{2 \ln \frac{r_{on}}{r_{in}}}\right] + \frac{r_{on}^2 - r_{in}^2}{2 \ln\left(\frac{r_{on}}{r_{in}}\right)} \quad .$$

Therefore

$$\frac{d\,T_{fn}}{dl} = \frac{q}{m_n C_{pn}} \left[ \frac{r_{mn}^2 - r_{m(n-1)}^2}{A_n/\pi} - 1 \right] + \frac{2K}{m_n C_{pn}} \; \frac{1}{A_n/\pi} \left[ a_{(n-1)} \left( T_{f(n-1)} - T_{fn} \right) + a_n \left( T_{f(n+1)} - T_{fn} \right) \right] .$$

## APPENDIX 2

### DERIVATION OF EQUATION 5

From equation 4 of the text when $\frac{dT_{fn}}{dl} = k$ ,

$$r^2_{mn} = r^2_{m(n-1)} + \frac{1}{\pi}\left[\frac{m_n C_{pn}}{q} k + 1\right] A_n \quad ,$$

and by successive substitution

$$r^2_{mn} = r^2_{mo} + \frac{1}{\pi}\sum_{i=1}^{n}\left[\frac{m_i C_{pi}}{q} k + 1\right] A_i \quad ,$$

or

$$r^2_{mn} = r^2_{mN} - \frac{1}{\pi}\sum_{i=n+1}^{N}\left[\frac{m_i C_{pi}}{q} k + 1\right] A_i \quad .$$

With equal mass flow per unit area and noting that $C_p$ will also be equal for equal temperatures,

$$r^2_{mn} = r^2_{mo} + \frac{1}{\pi}\left[\frac{m\,C_p}{q} k + 1\right]\sum_{i=1}^{n} A_i$$

$$= r^2_{mN} - \frac{1}{\pi}\left[\frac{m\,C_p}{q} k + 1\right]\sum_{i=n+1}^{N} A_i \quad .$$

Therefore

$$\left[\frac{m\,C_p}{q} k + 1\right] = \frac{r^2_{mN} - r^2_{mo}}{\frac{1}{\pi}\sum_{i=1}^{N} A_i} = \frac{1}{C} \quad .$$

and also

$$\left[\frac{m\,C_p}{q} k + 1\right] = \frac{r^2_{mn} - r^2_{m(n-1)}}{A_{n/\pi}} = \frac{1}{C} \quad .$$

where C is the voidage defined as the flow cross-section area divided by the sum of the flow and fuel cross-section areas.

## APPENDIX 3

### LIMITING VALUES

**1. Limiting value of $r_{mn}$ at large radii**

When $r_{on}$ and $r_{in}$ become large, their ratio X approaches unity. The expression $\frac{X \ln X}{1+X}$ then approaches the value $\frac{X-1}{2}$ .

Substituting in equation 7 of the text

$$r_{mn}^2 \to r_{in}\, r_{on} \quad .$$

It is apparent also that as $h/k \to 0$ , the same result holds.

**2. Limiting Tube Thickness**

To obtain an approximate analysis of the tube thickness substituting in equation 9 gives

$$\tau_n - \frac{D_e}{2}\,\frac{1-C}{C} = -\left(1 - \frac{D_e}{2\, r_{in}}\right)\left(\tau_{n-1} - \frac{D_e}{2}\,\frac{1-C}{C}\right) .$$

that is for any positive difference between tube thickness and $\frac{De}{2} \cdot \frac{1-C}{C}$ , there will be a smaller but negative difference corresponding to the thickness of the adjacent tube.

**3. Temperature Limits**

Using the limiting value of $r_{mn}$ , we have for large radii

$$\ln\left(\frac{r_{mn}^2}{r_{in}^2}\right) = \ln X$$

$$\simeq X - 1 \quad .$$

Substituting in equations 11, 12, and 13 of the text,

$$\frac{T_{mn} - T_{fn}}{\frac{D_e q}{h}} = \frac{R_{on} - R_{in}}{2D}\left[1 + \frac{R_{on} - R_{in}}{4}\right]$$

$$\frac{T_{in} - T_{fn}}{\frac{D_e q}{h}} = \frac{T_{on} - T_{fn}}{\frac{D_e q}{h}} = \frac{1}{2}\,\frac{R_{on} - R_{in}}{D} \quad .$$

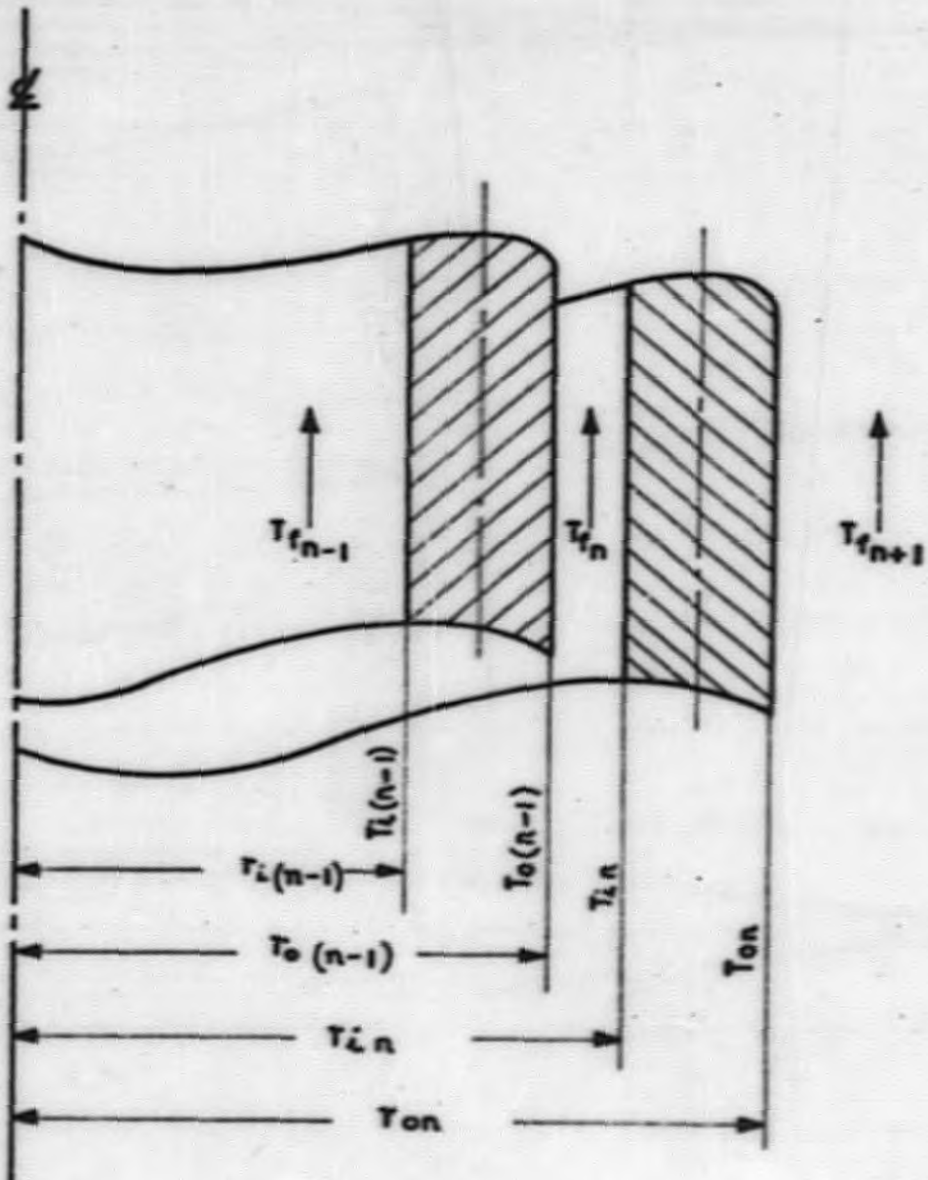


**FIGURE I. TEMPERATURES AND RADII AS USED IN CALCULATIONS.**

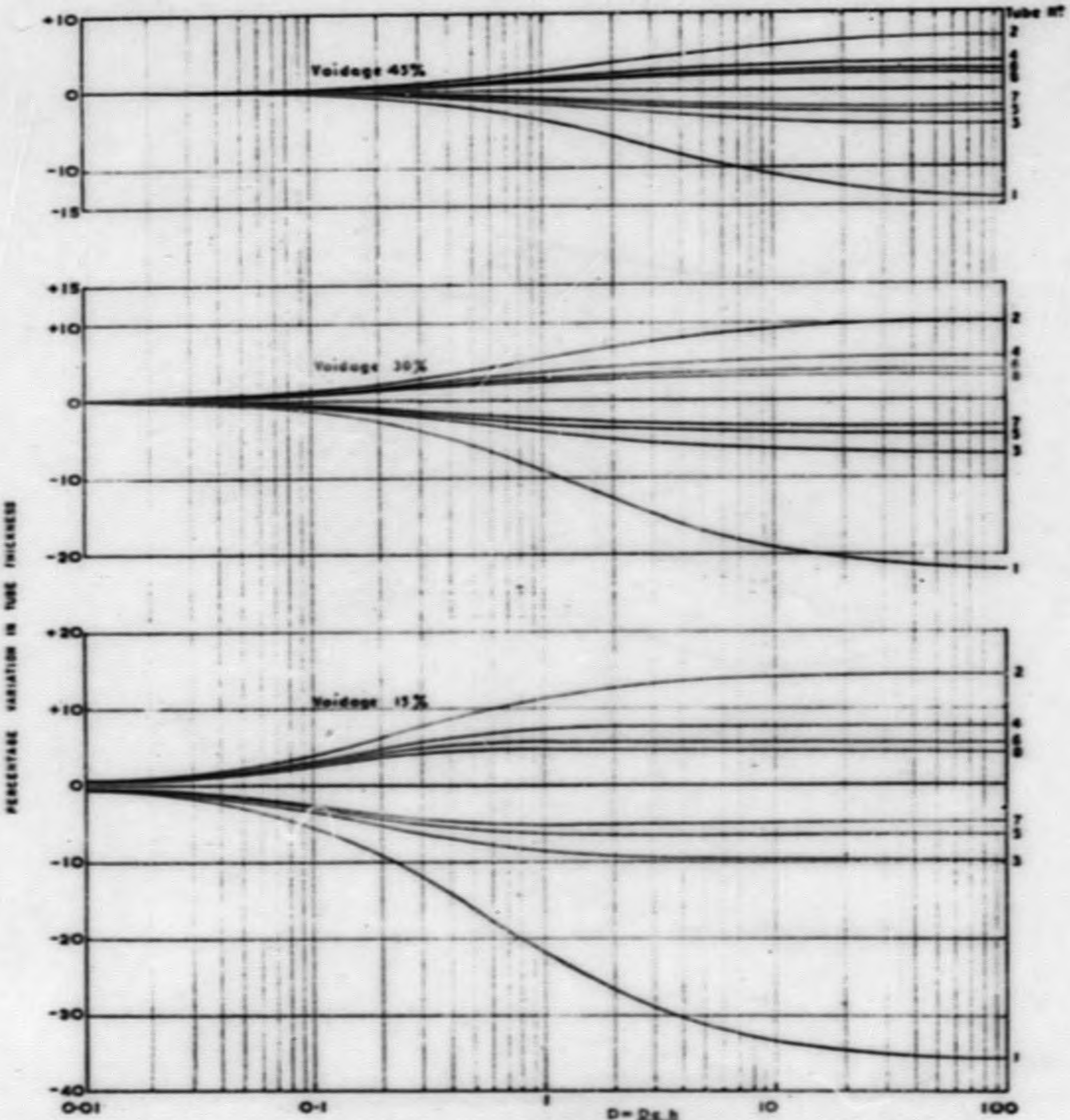


**FIGURE 2.** VARIATIONS IN TUBE THICKNESS WITHIN CLUSTERS OF CONCENTRIC TUBES HAVING CENTRE COOLANT CHANNELS.

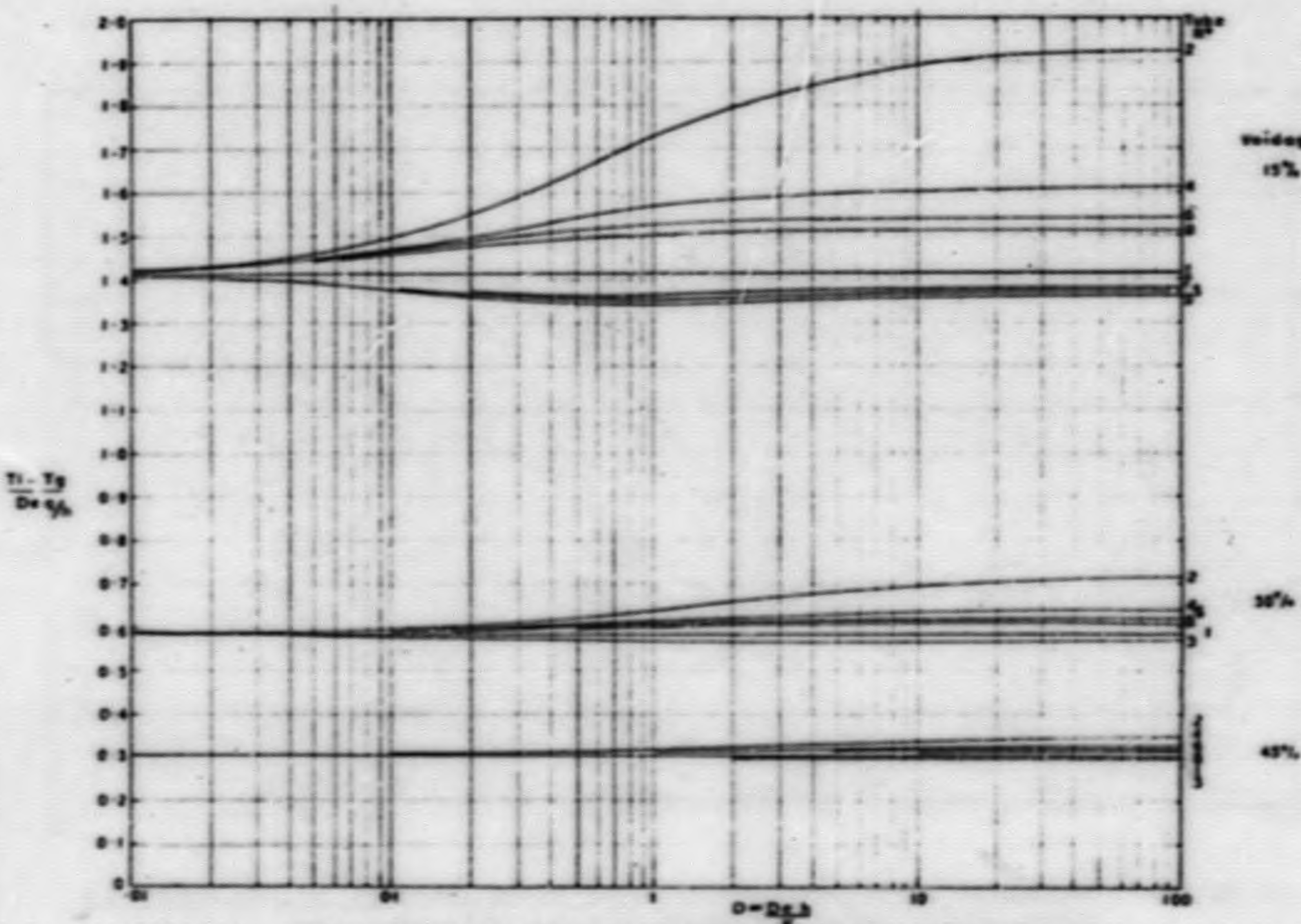


FIGURE 3. TEMPERATURES AT INNER TUBE SURFACES WITHIN CLUSTERS OF CONCENTRIC TUBES HAVING CENTRE COOLANT CHANNELS.

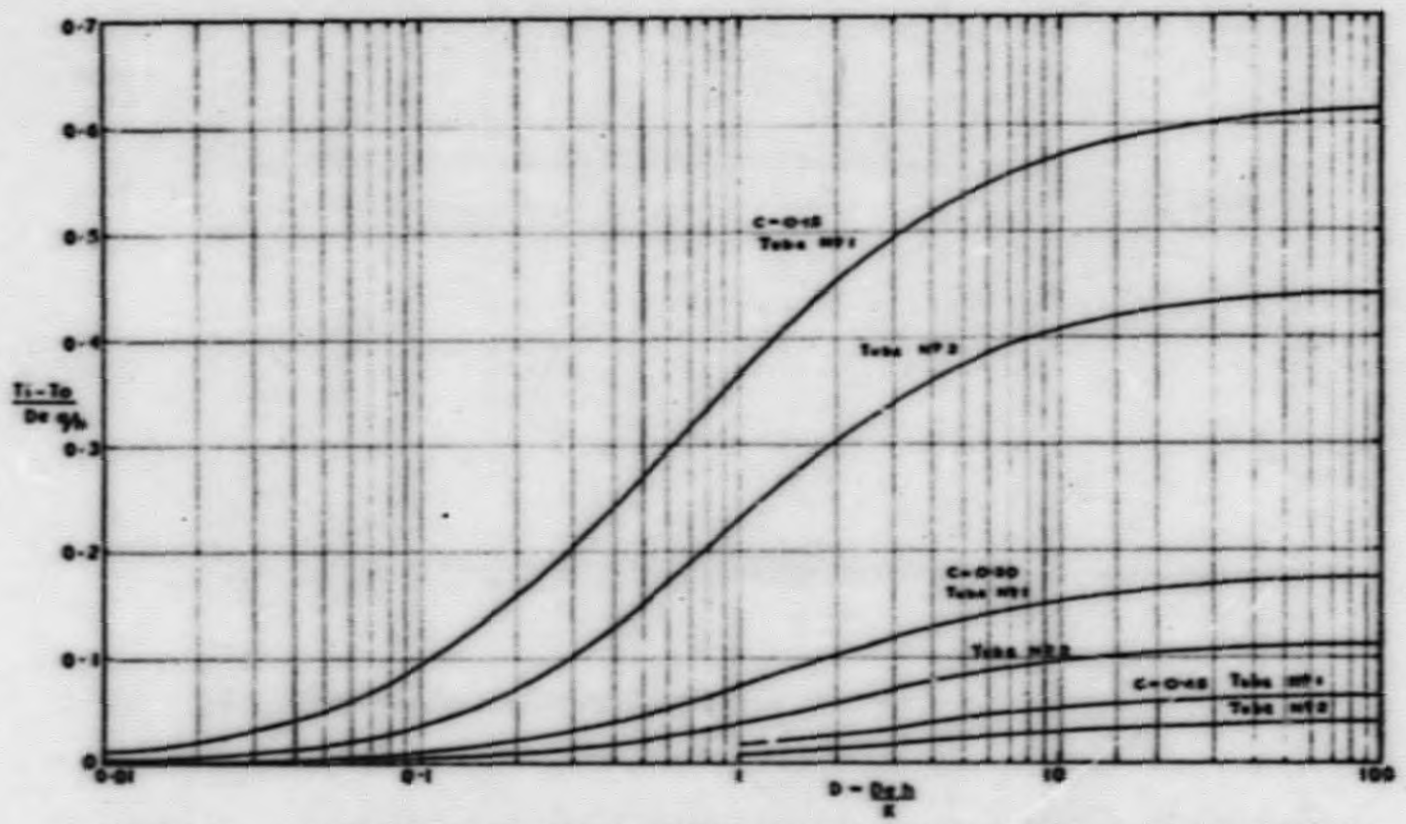


FIGURE 4. TEMPERATURE DIFFERENCES ACROSS TUBES WITHIN CLUSTERS OF CONCENTRIC TUBES HAVING CENTRE COOLANT CHANNELS

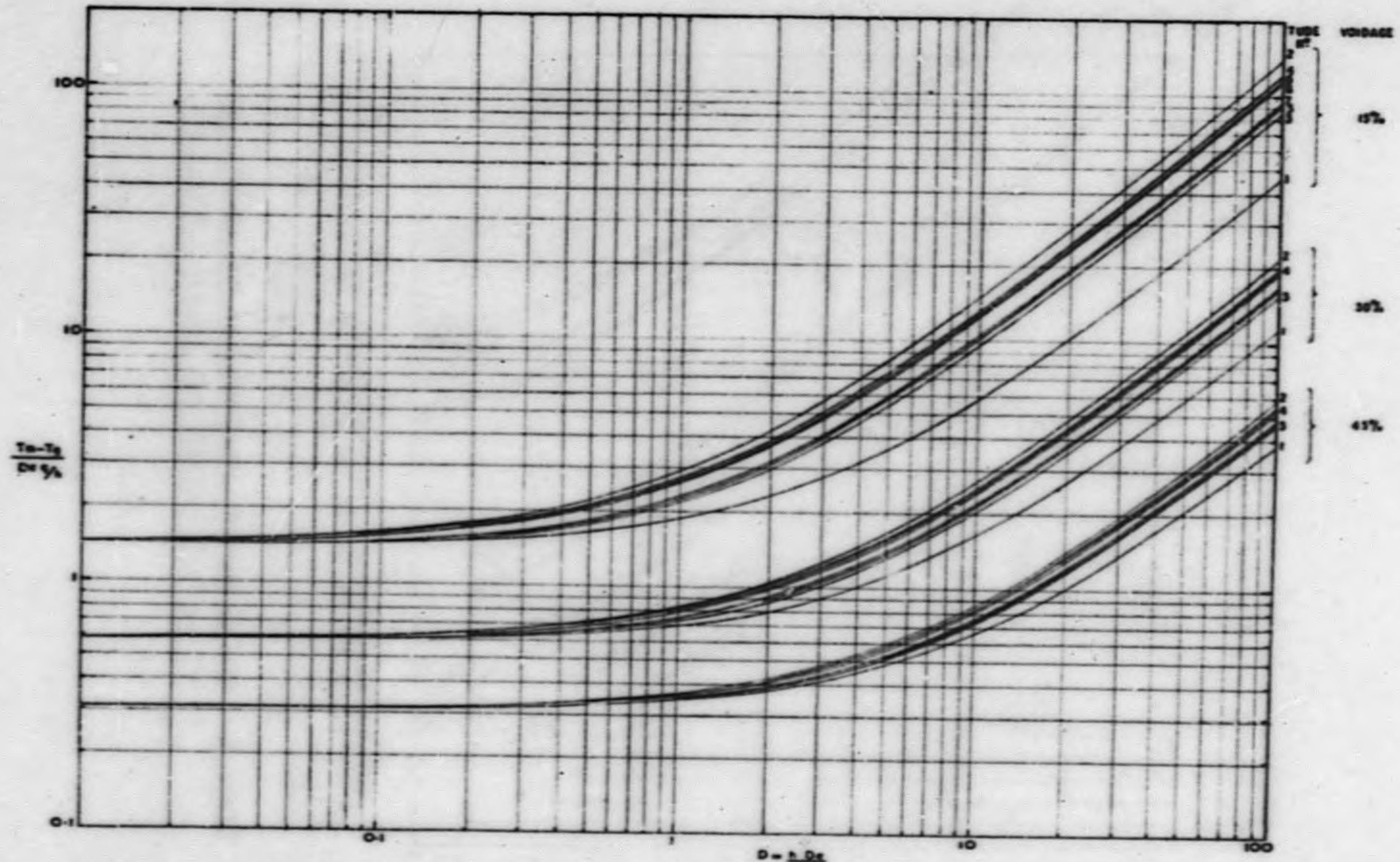


FIGURE 5. MAXIMUM TEMPERATURES OF TUBES WITHIN CLUSTERS OF CONCENTRIC TUBES HAVING CENTRE COOLANT CHANNELS

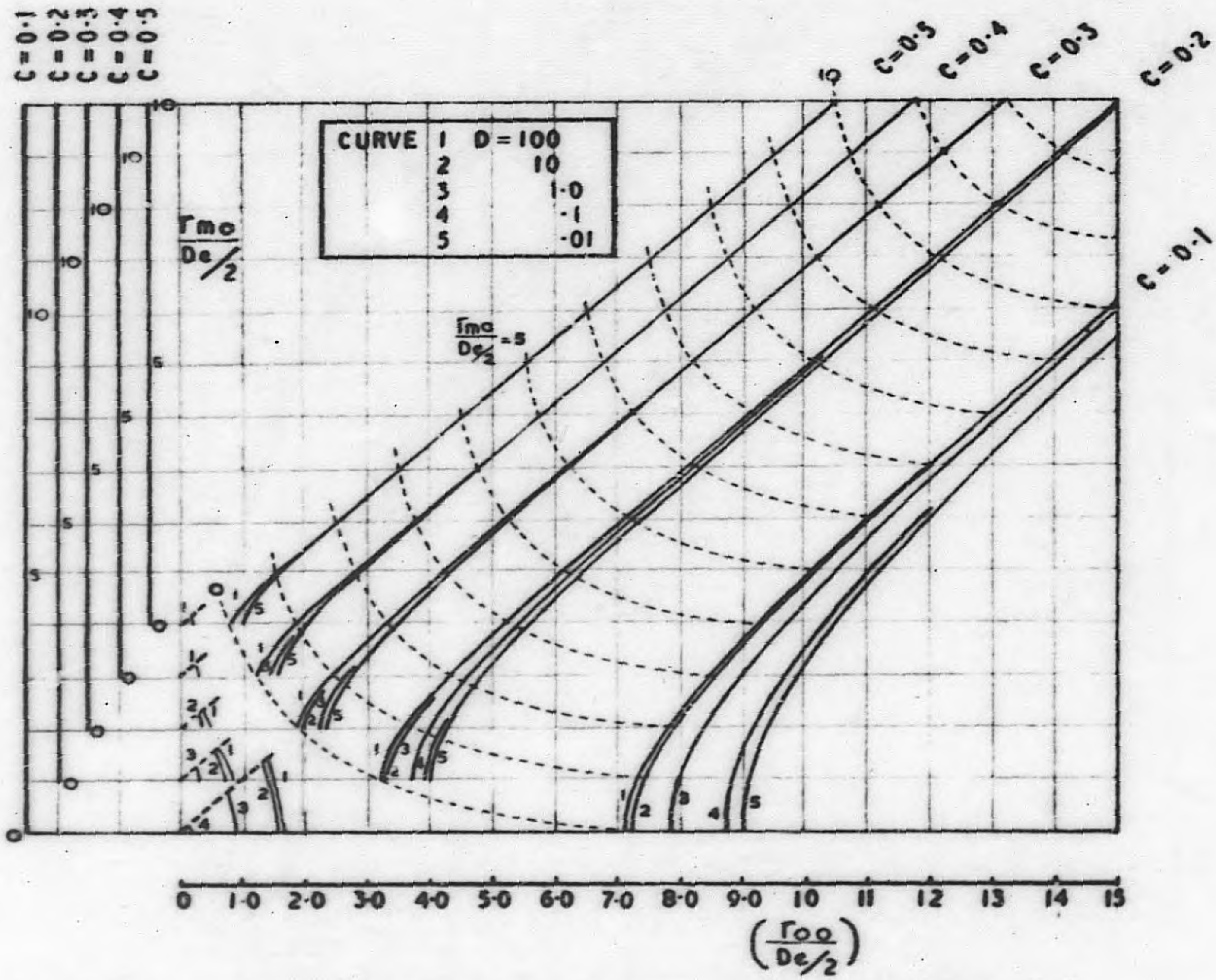


FIGURE 6. CENTRE BOUNDARY CONDITIONS FOR TUBE CLUSTERS OF UNIFORM TUBE THICKNESS AND COOLANT TEMPERATURES.

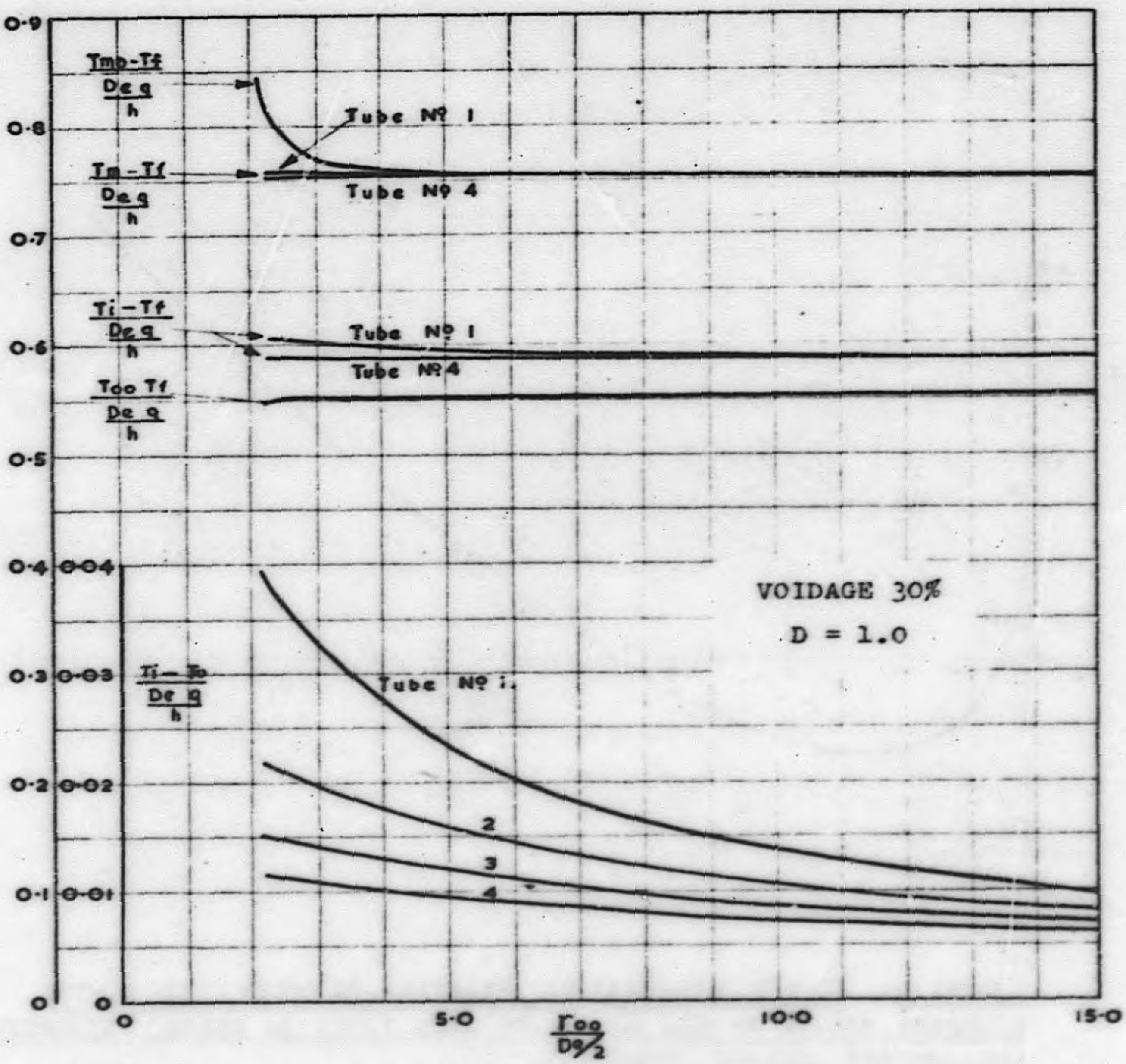


FIGURE 7. FUEL TEMPERATURES FOR VARIOUS CENTRE BOUNDARY CONDITIONS FOR TUBE CLUSTERS OF UNIFORM TUBE THICKNESS AND COOLANT TEMPERATURES.

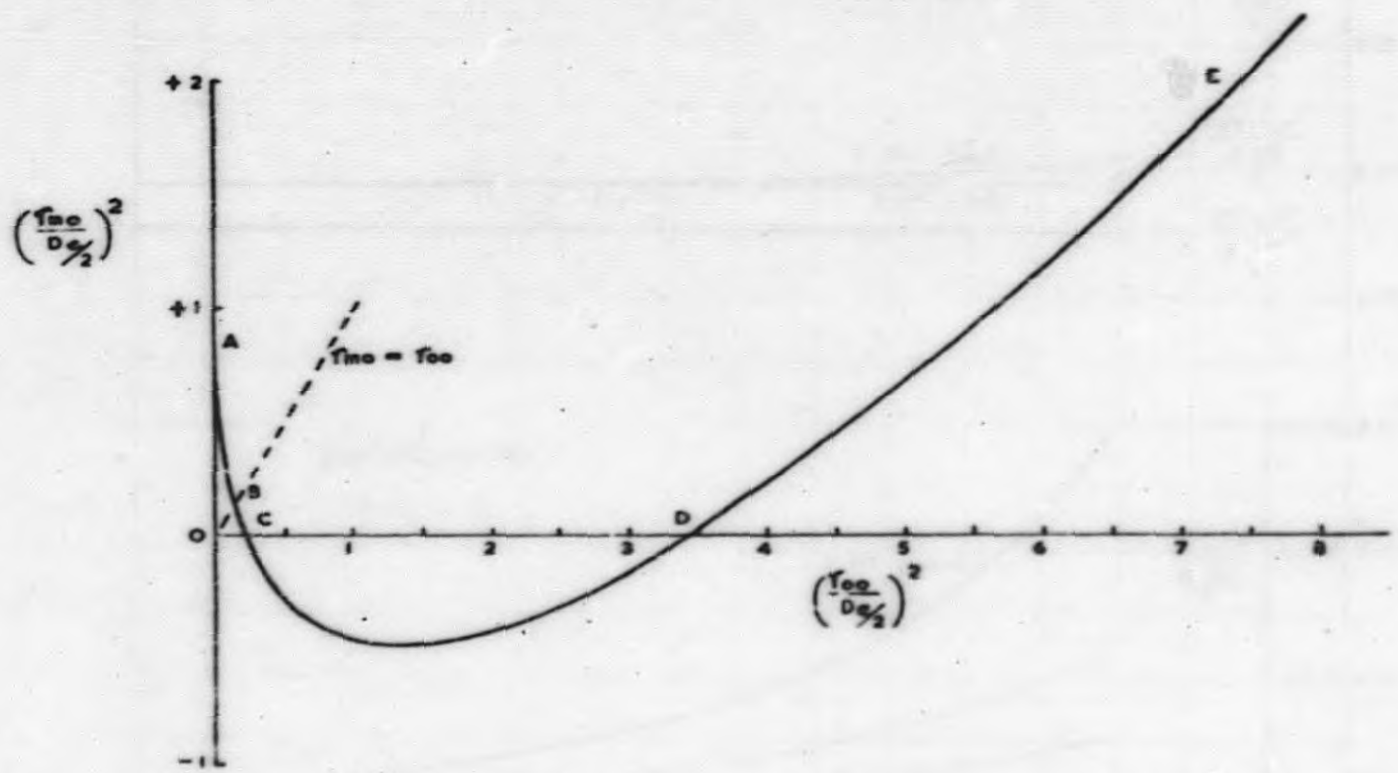


FIGURE 8. CURVE ILLUSTRATING RELATION BETWEEN THE CENTRE BOUNDARY VALUES OF $r_{mo}$ AND $r_{oo}$ TO GIVE TUBES OF EQUAL THICKNESS AND UNIFORM COOLANT CONDITIONS.

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