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SUBJECT: Heat Transfer and ΔP Design of MSRE Primary Heat Exchanger

TO: Distribution

FROM: J. H. Westsik

Abstract

A heat exchanger with a nominal transfer capacity of 10 Mw at design point has been specified for the MSRE. Upon investigation of several alternative configurations a cross-baffled geometry appears to be satisfactory. There are 165 U-tubes of 1/2-in. OD in a single shell. An over-all length of about 8 ft with a shell dia of 17 in. will be required. The fuel salt holdup is estimated as 5.5 ft^3. An outline of calculations is appended.

Introduction

The design of the primary heat exchanger for the MSRE was undertaken with the goal of specifying a component conservative in performance, compact in appearance, one that is structurally rugged and requires a minimum fuel volume. By means of analyses contained in the body of this report it will be demonstrated that the cross-baffled U-tube configuration satisfies these criteria best and is therefore the recommended selection.

Molten Fluoride Salts as Heat Transfer Fluids

The behavior of molten salts as heat transfer fluids has been investigated in connection with the ARE project. Results of a variety of experiments have been reported.[1] There is good agreement between these data and those published for other nonmetal agents under comparable conditions. Tube side film coefficients measured by Amos, MacPherson and Senn[2] agree with curves of Sieder and Tate[3] in the transitional flow range. Fig. 3 of the Yarosh[1] report compares heat transfer with water and fluoride salts in the shell side of the same heat exchanger. It was assumed, on the basis of this evidence, that the similarity extends into configurations that have not yet been investigated (e.g., cross-baffled shells).

There is only limited information available on friction factors. Measured friction factors are in apparent agreement with data of Standards of TEMA.[4] No significant change of pressure drop was observed over 1000 hr of operation with molten-salt exchangers.

Process Conditions in the Primary Heat Exchanger

Both primary and secondary salts enter the heat exchanger after a pumping step. It is desirable that the secondary salt be kept at slightly higher pressure than the fuel salt, thus assuring a dilution of the fuel in case of leakage rather than contamination of the secondary loop.

The design of the heat exchanger is based on the conditions listed in Table 1.

Table 1. Design Conditions

	Fuel Salt	Fuel Coolant
Composition, mole %:		
LiF	70	66
BeF_2	23	34
ThF_4	1	
ZrF_4	5	
UF_4	~1	
Temperature, °F:		
Inlet	1225	1025
Outlet	1175	1100
Flow Rates:		
gpm	1200	830
cfs	2.67	1.85
Average Physical Properties:		
Specific heat, Btu/lb-°F	0.46	0.57
Thermal conductivity, Btu/ft^2-hr-°F/ft	2.75	3.5
Viscosity, lb/ft-hr	17.9	20.0
Density, lb/ft^3	154.3	120
Prandtl number	3.00	3.26
Thermal conductivity of INOR-8, Btu/ft^2-hr-°F/ft		12.2

Mean Δt for true counter current heat exchange is 137°F. Applying a correction factor of 0.965 to include the effects of a single shell pass the effective Δt_m is reduced to 133°F (standards of TEMA).

In calculating the heat exchanger performance average physical properties were used. It was felt that the uncertainties in these properties would make more accurate methods meaningless.

Choice of Flow Paths

In all cases investigated the fuel salt was passed through the shell side. Aside from a somewhat higher fuel salt holdup this appears to be the preferable arrangement. An estimate was made to determine the expected fuel savings if the flow paths were reversed.

For the purposes of these calculations film coefficients and ΔP were assumed constant on both sides and merely the fluids were interchanged. Without charge for additional piping that would be required if the fuel passed through the tube side a volume saving of about 1 ft^3 would result. It is about 2% of the total reactor inventory and does not seem to constitute a compelling reason to sacrifice the straightforward design attainable with the fuel-in-shell alternative.

Heat Exchanger Geometry

The tube bundle of the exchanger is made of 0.500-in. OD x 0.042-in. wall INOR-8 tubes bent to a U-shape. There are 165 U-tubes in the bundle. A row of dummy rods in the center plane of the exchanger fills the space not usable for heat transfer surfaces because of the flow separating baffle in the coolant header.

The U-tube arrangement eliminates the problems caused by differential expansion of the shell and tubes. The resulting assembly is shorter, making the layout of the reactor more convenient. By means of the cross-baffled shell design a good matching of film coefficients is possible without disturbing the tube side flow.

The number of tubes specified is somewhat arbitrary except that about 165 U-tubes can be fitted into the shell without much void space. The secondary salt ΔP in the tube side is in the vicinity of 30 psi.

Shell side heat transfer with three configurations was investigated:

1. Unbaffled shell with tube pitch of 0.600 in.
2. Unbaffled shell with tube pitch of 0.750 in.
3. (a) Cross-baffled shell, variable tube pitch.
   (b) Cross-baffled shell, variable baffle pitch.

Heat Transfer and ΔP Analysis of Tube Side (Secondary Salt)

Heat transfer on the tube side is not affected by variations in shell side geometry of the cases under investigation, thus it will be treated separately and will be combined with the shell side figures at appropriate places. Table 2 shows the tube side characteristics.

Table 2. Tube Side Heat Transfer and ΔP

Number of U-tubes	165
Tube size, in.	0.500 OD x 0.042 wall
Flow area, ft^2	0.153
Flow velocity, fps	12.1
Reynolds modulus	9060
Film coefficient, Btu/ft^2-hr-°F	4940
Friction factor	0.040
Pressure drop per unit length, psi/ft	2.2

Shell Side Performance

Heat transfer on the shell side is strongly affected by the tube spacing. Close spacing is desirable. It is, however, limited by manufacturing considerations.

Case 1, of the shell side layout, represents about the closest spacing that would permit the use of tube spacers. At the tube-to-tube sheet joint one could overlap the trepan grooves or the tube ends could be offset to a wider spacing.

Tube spacing of Case 2 would permit a more convenient assembly at the tube sheet and the use of more rugged spacer design in the shells.

By varying the tube and baffle spacings in Case 3, one can combine the advantages of both previous cases. Results of calculations for Cases 1 and 2 are summarized in Table 3. A separate treatment of Case 3 appears to be necessary to indicate the effects of variable spacings.

Analysis of Cross-Baffled Heat Exchanger (Cases 3A and 3B)

In the cross-baffled geometry the shell side heat transfer is affected by both the tube and the shell spacings. Heat exchanger characteristics as functions of both variables are shown on graphs of Figs. 1 and 2. These characteristics were calculated with correlations of Kern.[5]

In theory a large number of combinations of both spacings is possible within the limits of accuracy of the correlations. However, the physical layout requires that there be an even number of cross baffles. It is assumed

Table 3. Heat Transfer and ΔP in Cases 1 and 2

Number of U-tubes:	165	
	Case 1	Case 2
Tube pitch, in.	0.600	0.750
Shell ID, in.	12	15
Flow area, ft^2	0.308	0.760
Equivalent, diameters:		
Reynolds, ft	0.0252	0.0610
Nusselt, ft	0.0286	0.0664
Reynolds modulus	6740	6660
Shell side film coefficient, Btu/ft^2-hr-°F	2720	1160
Over-all coefficient of heat transfer, Btu/ft^2-hr-°F	1010	655
Friction factor in shell side	0.046	0.046
Shell side pressure drop per unit length, psi/ft	2.39	0.156
Heat transfer surface area, ft^2	263	392
Active shell length, ft	6.10	9.07
ΔP in active section:		
Tube side, psi	31.7	44.3
Shell side, psi	17.0	1.67
Fuel inventory, ft^3	2.2	8.1

that the shell side ΔP must stay below 30 psi to hold the total primary loop ΔP to the 50 psi limit.

From the graphs of Fig. 1 the configuration of 6 baffles spaced at 12 in. apart with the tubes spaced at 0.775 in. on a triangular pitch gives a ΔP of 20.5 psi on the shell side and 29.0 psi in the tube side. The fuel holdup is 5.5 ft^3, or about 10% of the total inventory. The tube spacing is wide enough to make assembly operations and tube-to-tube sheet welds conveniently attainable. An over-all length of 8 ft is expected.

An inside shell diameter of 16 in. is required to hold the tube bundle. It is expected that the shell would be made of 1/2 in. thick plate rolled to this diameter and a short section near the tube sheet would be turned to 16-12 in. ID, and be joined to the flange of the tube sheet with identical ID. In this manner generous fillet may be allowed at the tube sheet flange.

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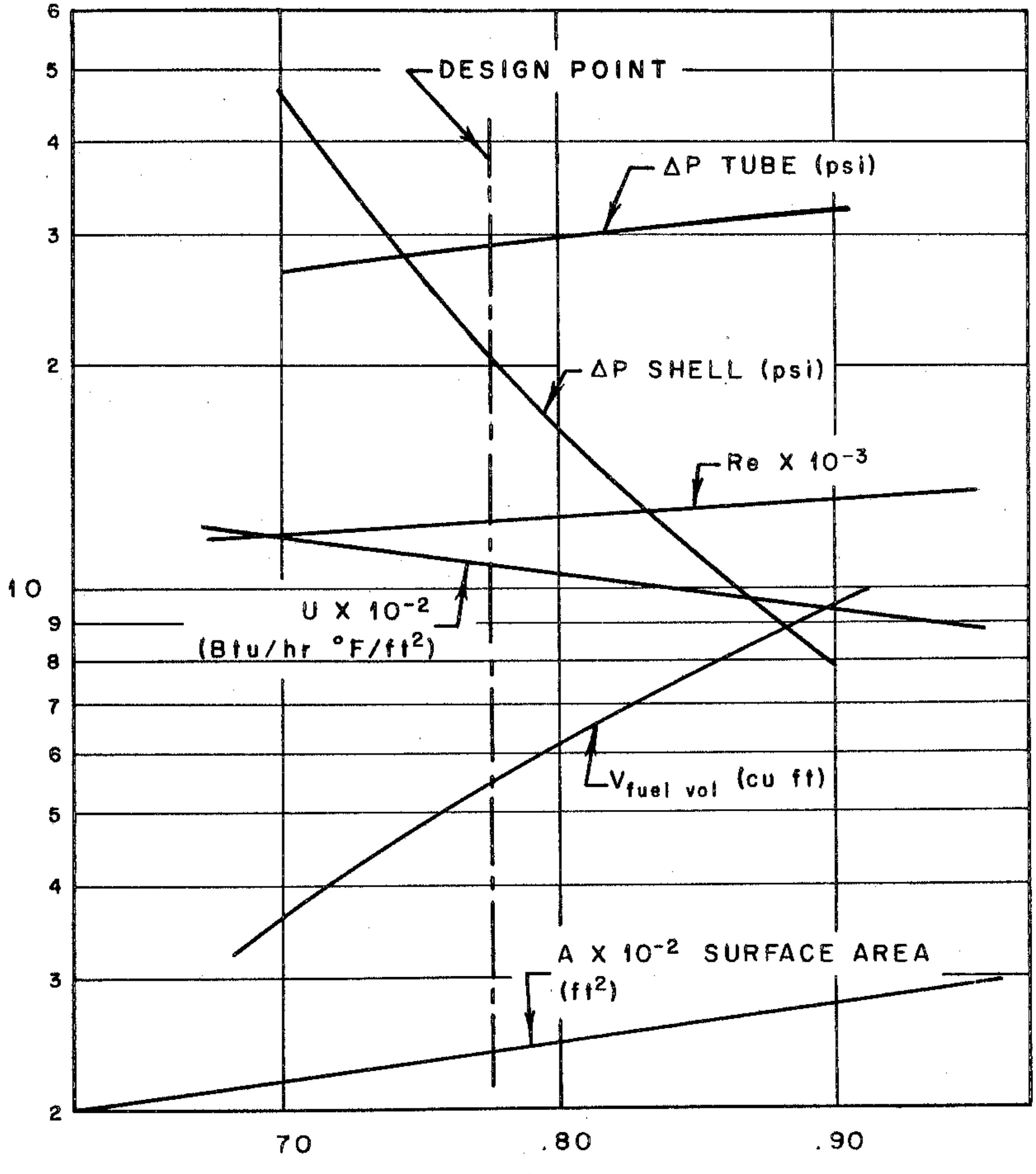

Fig. 1. Cross Baffled Heat Exchanger. Baffle Pitch 12", Tube Pitch Variable.

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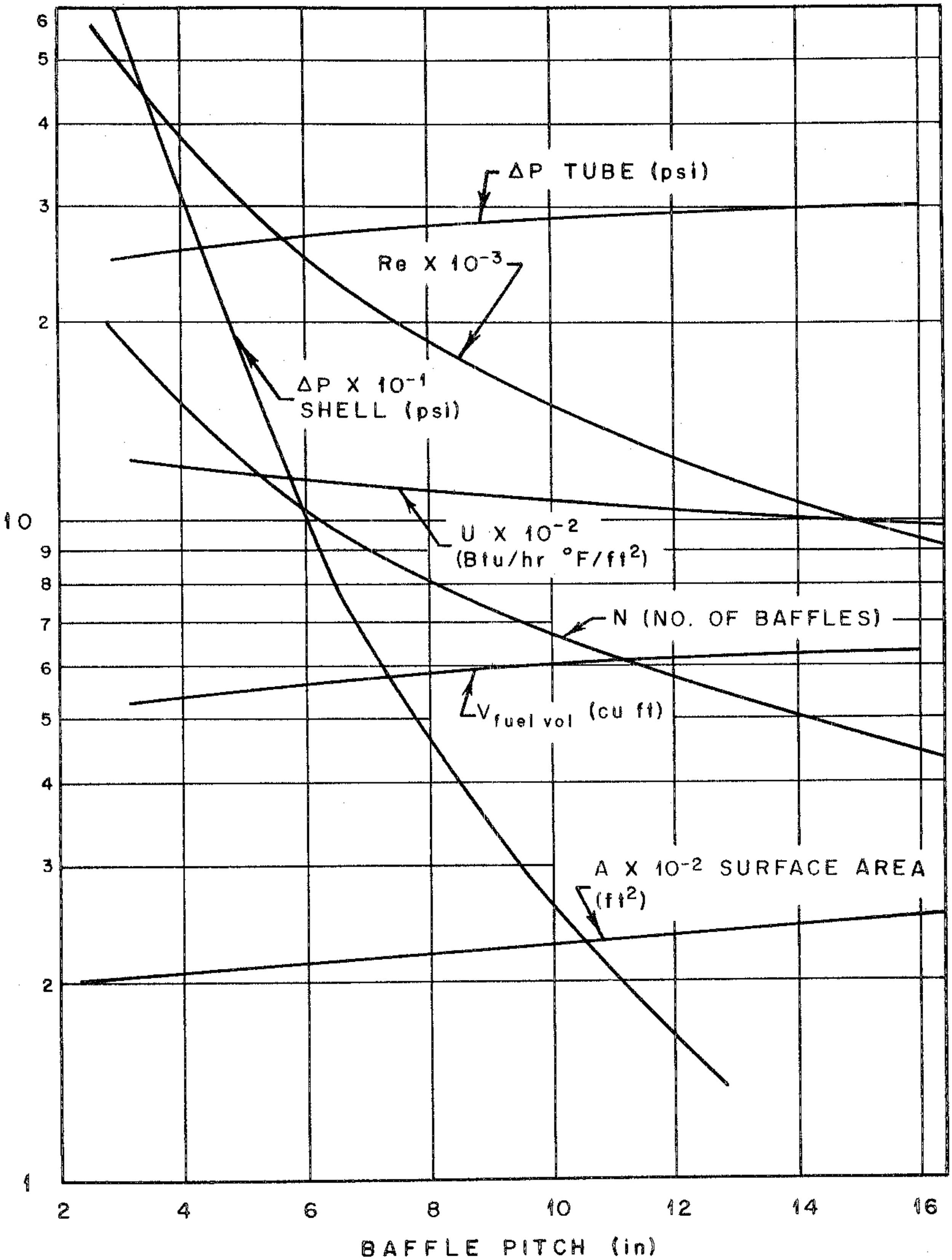

Fig. 2. Cross Baffled Heat Exchanger. Tube Pitch 8", Baffle Pitch Variable.

Rating of the Proposed Heat Exchanger

The active length of the shell is taken as the length of the straight portion of the tubes between the thermal barrier plate and the last baffle. Experience with U-tube exchangers indicates that heat transfer at the bends is significantly less than that predicted for straight tubes.

The active shell section contains 259.2 ft^2 heat transfer surface, or about 8% more than the calculated requirement. In the heat transfer coefficient allowance was made for possible scale deposits through increasing the resistance to heat flow by 10.7%. In all a margin of better than 20% was allowed.

Heat Transfer Below Design Point Power

It is of some interest to trace the performance of the heat exchanger when the power extraction is below the design level.

If the reduction of power level occurs by reducing the temperature change in each fluid, the film coefficients remain essentially constant and the mean Δt required to transfer the thermal energy changes linearly with the power level. In the second case investigated, the flow of the fuel stream was so regulated that the temperature change remained constant at 50°F, i.e., the flow rate was made proportional to the power level. The flow of the secondary coolant remained constant as would the case be with a constant speed secondary pump.

Fig. 3 shows the curves of Δt_m vs power level for both cases. Corresponding inlet and outlet temperatures for both fluids are shown on Figs. 4 and 5.

Thermal Convection in the Primary Loop

The layout of the primary loop permits a limited amount of power removal through circulation induced by thermal convection.

It was assumed that 5% of the design power would have to be removed. The elevation difference between the heated and cooled volumes was estimated as 8 ft.

Because of the complexity of the loop a summation of ΔP section by section was necessary. Flow resistances in each segment were expressed in terms

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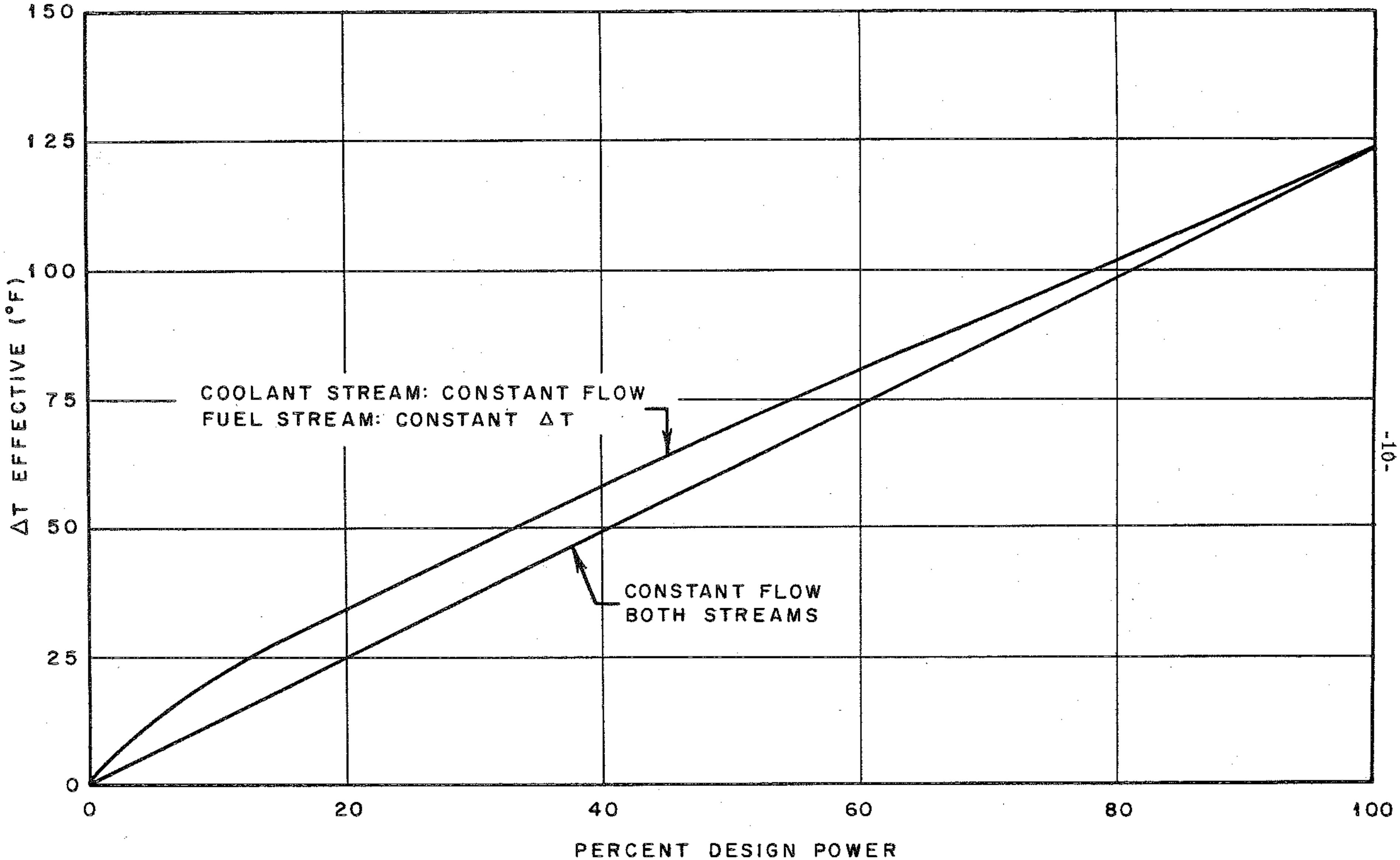

Fig. 3. Effective Δt_m Requirement at Reduced Power Operation.

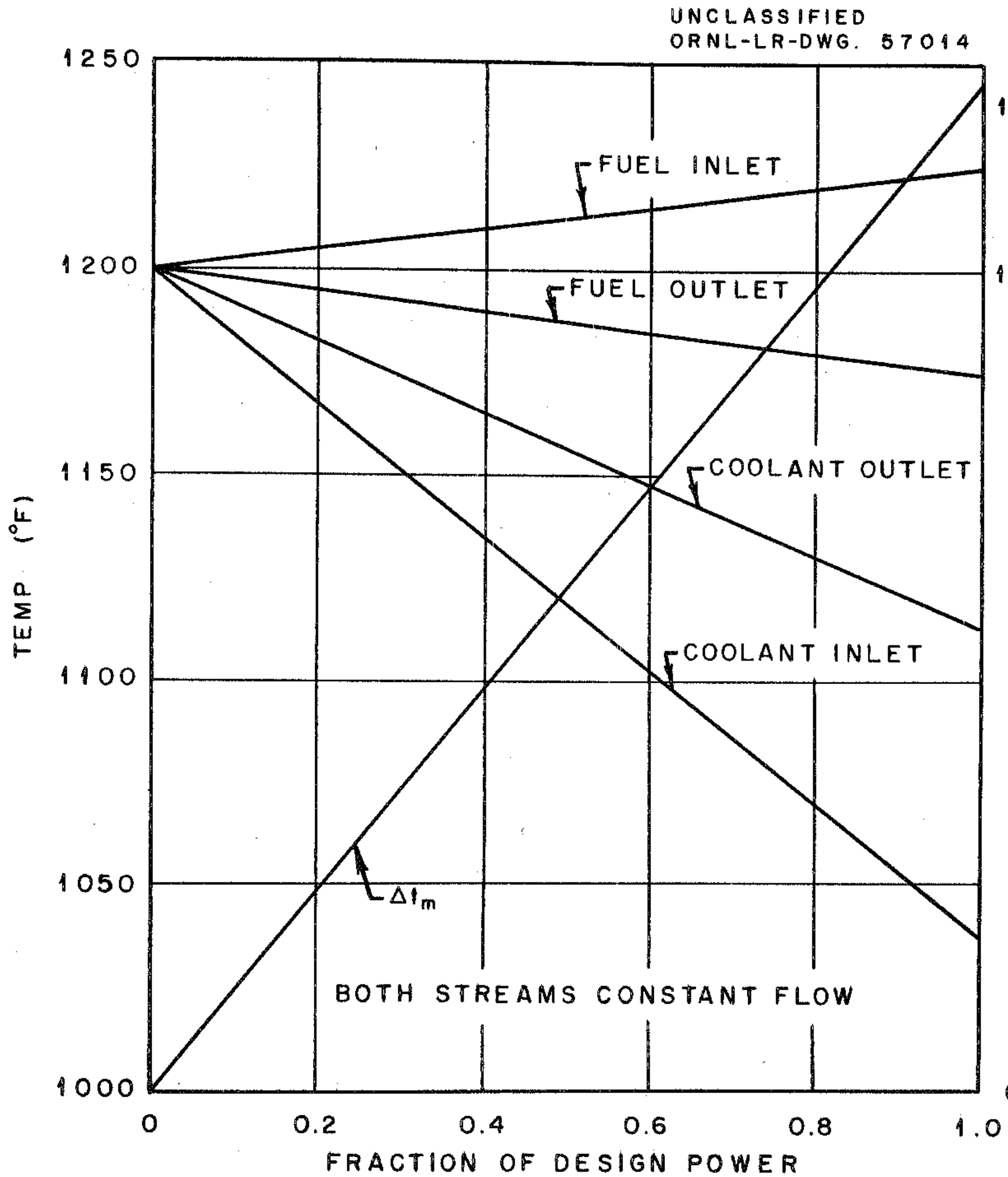

Fig. 4. MSRE Heat Exchanger Temperatures at Reduced Loads.

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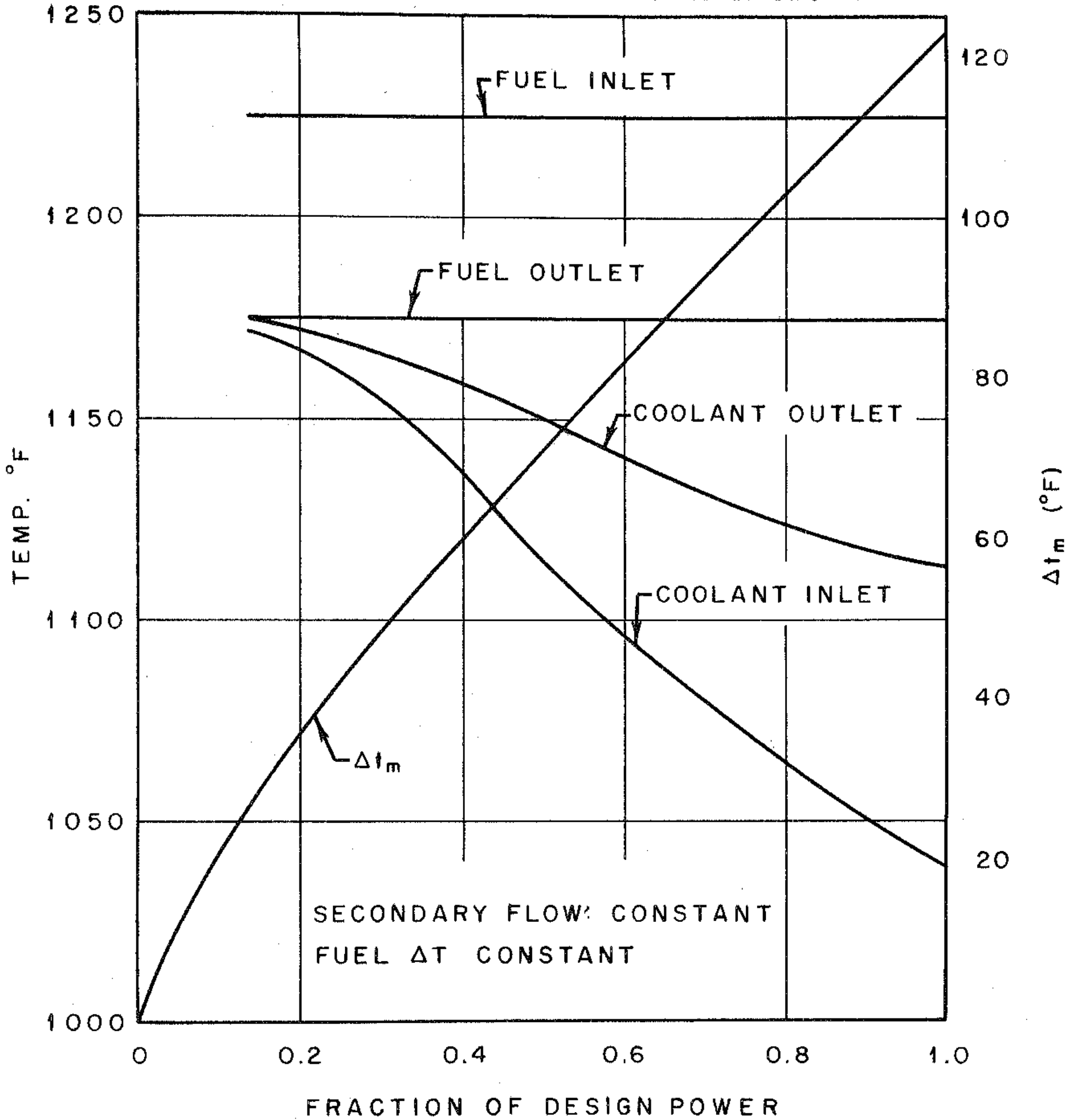

Fig. 5. MSRE Heat Exchanger Temperatures at Reduced Loads.

of the temperature driving force requirement. Friction factors were taken as $\frac{64}{Re}$ in laminar region and $.16(\frac{1}{Re})^{.25}$ in turbulent and transition regions. The total loop resistance was then equated to the temperature driving force to give an equation of the form

$$\Delta t_D = \frac{a}{(\Delta t_D)^{1.75}} + \frac{b}{\Delta t_D}$$

where Δt_D = temperature difference between the hot and cold leg (°F)

$$\left.\begin{array}{lcl} a & = & 7.56 \times 10^3 \\ b & = & 1.463 \times 10^3 \end{array}\right\} \text{system resistance constants}$$

Graphical solution of the equation yielded Δt_D = 43.5°F. If the average fuel temperature is 1200°F and Δt_m = 11.0°F and the temperature change of the secondary loop is 3.75°F (design flow rate maintained in secondary coolant loop), secondary salt terminal temperatures would be approximately 1181°F and 1178°F. One must note that these figures are dependent upon the configuration of the loop and should be reevaluated when the layout is finalized.

REFERENCES

1. R. E. MacPherson and M. M. Yarosh, Development Testing and Performance Evaluation of Liquid Metal and Molten-Salt Heat Exchangers, ORNL CF-60-3-164 (Mar. 17, 1960).

2. J. C. Amos, R. E. MacPherson, and R. L. Senn, Preliminary Report of Fused Salt Mixture 130 Heat Transfer Coefficient Test, ORNL CF-58-4-23 (Apr. 2, 1958).

3. D. Q. Kern, Process Heat Transfer, p 834, Fig. 24, McGraw-Hill, New York (1950).

4. Standards of Tubular Exchanger Manufacturers Association, 2d ed., Tubular Exchanger Mfrs. Assn., Inc., New York (1949).

5. D. Q. Kern, Process Heat Transfer, pp 137-139, McGraw-Hill, New York (1950).

NOMENCLATURE

a = heat transfer surface per unit shell length (ft^2/ft)

A = total heat transfer surface (ft^2)

A_f = shell side flow area (ft^2)

a_s = equivalent shell side flow area in cross-baffled designs (ft^2)

B = baffle pitch (in.)

C' = tube clearance (in.)

d = tube diameter (in.)

D = tube diameter (ft)

D_e = equivalent diameter (ft)

D_s = shell diameter (ft)

f = friction factor (dimensionless)

f' = friction factor (ft^2/in^2)

g = proportionality constant (ft/sec^2)

G = mass velocity (lb/ft^2-hr)

h = heat transfer film coefficient (Btu/ft^2-hr-°F)

k = thermal conductivity (Btu/ft^2-hr-°F/ft)

ℓ = straight length of tube (in.)

L = active tube length (ft)

L_T = active shell length (ft)

m = variable factor (dimensionless)

n = number of U-tubes in bundle (dimensionless)

N = number of cross baffles (dimensionless)

Nu = Nusselt modulus (dimensionless)

p = power (watts)

ΔP = pressure drop (psi)

Pr = Prandtl modulus (dimensionless)

P_T = tube pitch (in.)

q = heat transfer rate (Btu/hr)

Q = volumetric flow rate (ft^3/sec or ft^3/hr as specified)

ρ = density (lb/ft^3)

Re = Reynolds modulus (dimensionless)

s = specific gravity (dimensionless)

NOMENCLATURE (continued)

t = coolant temperature (°F)

T = fuel temperature (°F)

Δt = coolant temperature rise (°F)

ΔT = fuel temperature rise (°F)

Δt_m = mean temperature difference (°F)

U = overall coefficient of heat transfer (Btu/ft^2-hr-°F)

V = fluid velocity (ft/sec)

v_f = fuel holdup (ft^3)

W = mass rate of flow (lb/hr)

x = tube wall thickness (ft)

μ = viscosity (lb/ft-hr)

Subscripts

a = average

c = coolant side

DP = design point

f = fuel side

i = inside

m = where the variable factor applies

o = outside

s = shell side

Appendix I

<u>Heat transfer in tube side</u>

1. Reynolds modulus at 10 Mw operation

$$Re = \frac{V\, d_i \rho}{\mu}$$

$$= \frac{12.1 \times 0.416 \times 120}{20 \times 12} \times 3600$$

$$= 9060$$

2. ℓ/d ratio:

assume ℓ = 72 in. (undisturbed length!)

$$\ell/d_i = \frac{72}{0.416}$$

$$= 173$$

3. Tube side film coefficient

from Fig. 24 of Kern: <u>Process Heat Transfer</u>

@ Re = 9060 and ℓ/d = 180

D = 0.0347 ft

$$\frac{h\, D}{k} \times \left(\frac{1}{Pr}\right)^{-1/3} = 33.0$$

$$h = \frac{3.5}{0.0347} \times (3.26)^{1/3} \times 33$$

$$= 4940\ \text{Btu/ft}^2\text{-hr-°F}$$

<u>Pressure drop in tube side</u>

$$\frac{\Delta P}{L} = \frac{Psf}{Psi} \times f\, \frac{\rho V^2}{2g} \times \frac{1}{D}$$

@ Re = 9600

f = 0.040

$$= \frac{1}{144} \times 0.040 \times 120 \times \frac{(12.1)^2}{2 \times 32.2} \times \frac{1}{0.0347}$$

$$= 2.2\ \text{Psi/ft}$$

Appendix II

Heat transfer in shell side, Cases 1 and 2

Case 1. Geometry

Tube pitch = 0.600 in.

Shell diameter: (assume 20 tube pitches) = 12 in.

Flow area:

$$A_f = \frac{\pi}{4} D_s^2 - (2n + 20) \frac{\pi}{4} D_o^2$$

$$= \frac{3.14}{4} \times \left(\frac{12}{12}\right)^2 - (2 \times 165 + 20) \frac{3.14}{4} \times \left(\frac{0.5}{12}\right)^2$$

$$= 0.785 - 0.477$$

$$= 0.308 \text{ ft}^2$$

Equivalent diameters (D_e)

Reynolds $\left[D_e\right]$

$$= \frac{4 A_f}{\text{Wetted perimeter}}$$

$$= \frac{4 \times 0.308}{(2 \times 165 + 20)\ 3.14 \times \frac{0.5}{12} + 3.14 \times \frac{12}{12}}$$

$$= 0.0252 \text{ ft}$$

Nusselt $\left[D_e\right]$

$$= \frac{4 A_f}{\text{Heated perimeter}}$$

$$= \frac{4 \times 0.308}{(2 \times 165) \times 3.14 \times \frac{0.5}{12}}$$

$$= 0.0286 \text{ ft}$$

Heat transfer coefficients

Flow velocity

$$V = \frac{Q}{A_f}$$

$$V = \frac{2.67}{0.308}$$

$$= 8.67 \text{ ft/sec}$$

Reynolds modulus

$$Re = \frac{V\ d\ \rho}{\mu} \times 3600$$

$$= \frac{8.67 \times 0.0252 \times 154.3}{17.9} \times 3600$$

$$= 6770$$

Heat transfer film coefficient

from CF 60-3-164, p 23

equation of composite curve of 12 heat exchangers:

$$\frac{Nu}{Pr^{0.4}} = 0.01015\ (Re)^{0.85}$$

$$h = \frac{k}{D_e} \times (Pr)^{0.4}\ 0.01015\ (Re)^{0.85}$$

$$= \frac{2.75}{0.0286} \times (3.00)^{0.4} \times 0.01015 \times (6770)^{0.85}$$

$$= 2720 \text{ Btu/ft}^2\text{-hr-°F}$$

Pressure drop

$$\frac{\Delta P}{L} = \frac{1}{144}\ f\ \rho\ \frac{V^2}{2g}\ \frac{1}{D_e}$$

@ $Re = 6770$

from Fig. 9, p 29 of CF 60-3-164

$$f = 0.046$$

$$\frac{\Delta P}{L} = \frac{1}{144} \times 0.046 \times \frac{154.3}{2 \times 32.2} \times (8.67)^2 \times \frac{1}{0.0252}$$

$$= 2.28 \text{ Psi/ft}$$

<u>Overall heat transfer coefficient</u>

$$\frac{1}{U} = \frac{1}{h_f} + \frac{d_o}{d_i}\,\frac{1}{h_c} + \frac{1}{h_{scale}} + \left[\frac{x}{k}\right]_{INOR} \times \frac{d_o}{dm}$$

$$= \frac{1}{2720} + \frac{0.500}{0.416} \times \frac{1}{4940} + \frac{1}{10{,}000} + \frac{0.042}{12 \times 12.2} \times \frac{0.500}{0.458}$$

$$= (3.68 + 2.44 + 1.00 + 3.13) \times 10^{-4}$$

$$= 10.25 \times 10^{-4}$$

$$U = 976 \text{ Btu/ft}^2\text{-hr-°F}$$

<u>Heat transfer surface area</u>

$$A = \frac{q}{U\,\Delta t_m}$$

$$= \frac{3.413 \times 10^7}{9.76 \times 1.33 \times 10^4}$$

$$= 263 \text{ ft}^2$$

Heat transfer area per unit length

$$a = 2 \times 165 \times 3.14 \times \frac{0.5}{12} \times 1$$

$$= 43.2 \text{ ft}^2\text{/ft}$$

Active heat exchanger length

$$L_T = \frac{A}{a}$$

$$= \frac{263}{43.2}$$

$$= 6.1 \text{ ft}$$

Shell side pressure drop

$$\Delta P = \left(\frac{\Delta P}{L}\right)_s \times (L_T + 1)$$

$$= 2.28 \times (6.1 + 1)$$

$$= 16.2 \text{ psi}$$

Tube side pressure drop

$$\Delta P = \left(\frac{\Delta P}{L_C}\right) \times (2L_T + 2)$$

$$= 2.20 \times (12.2 + 2)$$

$$= 31.7 \text{ psi}$$

Approximate fuel volume

$$v_f = A_f (L_T + 1)$$

$$= 0.308 (6.1 + 1)$$

$$= 2.19 \text{ ft}^3$$

Case 2. Geometry

Tube pitch = 0.750 in.

Shell diameter: (assume 20 tube pitches) = 15 in.

Flow area:

$$A_f = \frac{\pi}{4} D_s^2 - (2n + 20) \frac{\pi}{4} D_o^2$$

$$= \frac{3.14}{4} \left(\frac{15}{12}\right)^2 - (2 \times 165 + 20) \frac{3.14}{4} \left(\frac{0.5}{12}\right)^2$$

$$= 1.227 - 0.477$$

$$= 0.750 \text{ ft}^2$$

Equivalent diameters

Reynolds $\left[D_e\right]$

$$= \frac{4\ A_f}{\text{Wetted perimeter}}$$

$$= \frac{0.750 \times 4}{3.14\ (\frac{15}{12}) + 3.14\ (2 \times 165 + 20) \times \frac{0.5}{12}}$$

$$= 0.0604 \text{ ft}$$

Nusselt $\left[D_e\right]$

$$= \frac{4\ A_f}{\text{Heated perimeter}}$$

$$= \frac{4 \times 0.750}{3.14\ (2 \times 165 + 20)\ \frac{0.5}{12}}$$

$$= 0.0655 \text{ ft}$$

Heat transfer coefficient

Reynolds modulus

$$Re = \frac{V\ D_e\ \rho}{\mu} \times 3600$$

$$V = \frac{Q}{A_f}$$

$$= \frac{2.67}{0.750}$$

$$= 3.56 \text{ ft/sec}$$

$$Re = \frac{3.56 \times 0.0604 \times 154.3}{1.79} \times 3600$$

$$= 6670$$

Film coefficient from CF 60-3-164, p 123

$$\frac{Nu}{Pr^{0.4}} = 0.01015\ (Re)^{0.85}$$

$$= 0.01015\ (6670)^{0.85}$$

$$= 18.07$$

$$h = \frac{k}{D_e} \times (Pr)^{0.4} \times 18.07$$

$$= \frac{2.75}{0.0655} \times (3.00)^{0.4} \times 18.07$$

$$= 1176\ \text{Btu/ft}^2\text{-hr-°F}$$

Pressure drop per unit length

$$\frac{\Delta P}{L} = \frac{1}{144} \times f\ \frac{\rho V^2}{2g} \times \frac{1}{D_e}$$

from Fig. 9, p 29 of CF 60-3-164

@ Re = 6670

f = 0.046

$$\frac{\Delta P}{L} = \frac{1}{144} \times 0.046 \times \frac{154.3}{2 \times 32.2} \times (3.56)^2 \times \frac{1}{0.0604}$$

$$= 0.161\ \text{Psi/ft}$$

Overall coefficient of heat transfer

$$\frac{1}{U} = \frac{1}{h_f} + \frac{1}{c}$$

$\frac{1}{c}$ = resistances other than shell side film

$$= 6.57 \times 10^{-4}\ \text{ft}^2\text{-hr-°F/Btu}$$

$$\frac{1}{U} = \frac{1}{1176} + 6.57 \times 10^{-4}$$

$$= (8.50 + 6.57) \times 10^{-4}$$

$$= 15.07$$

$$U = 664 \text{ Btu/ft}^2\text{-hr-°F}$$

Heat transfer surface area

$$A = \frac{q}{U \, \Delta t_m}$$

$$= \frac{3.413 \times 10^7}{664 \times 133}$$

$$= 386 \text{ ft}^2$$

Active exchanger length

$$L_T = \frac{A}{\frac{A}{L_T}}$$

$$\frac{A}{L_T} = 43.2 \text{ ft}^2\text{/ft}$$

$$= \frac{386}{43.2}$$

$$= 8.94 \text{ ft}$$

Tube side pressure drop

$$\Delta P = (2\, L_T + 2) \left(\frac{\Delta P}{L_C}\right)$$

$$= (2 \times 8.94 + 2)\, 2.2$$

$$= 43.7 \text{ psi}$$

Shell side pressure drop

$$\Delta P = (L_T + 1) \left(\frac{\Delta P}{L}\right)_s$$

$$= (8.94 + 1)(0.161)$$

$$= 1.60 \text{ psi}$$

Fuel holdup (approximate)

$$v_f = (L_T + 1) A_f$$

$$= (8.94 + 1)(0.750)$$

$$= 7.45 \text{ ft}^3$$

Appendix III

Heat transfer in shell side, Cases 3A and 3B

Case 3A.

25% cut cross baffle, constant tube pitch, variable baffle pitch

Let the tube pitch

$$P_T = 0.800 \text{ in.}$$

and the baffle pitch

$$0.200\ D_s < B < 1.00\ D_s$$

Using relations 7.1 - 7.5 of Kern (pp 138-139)

Geometry

Flow cross section (Eq. 7.1)

$$a_s = \frac{D_s \times C' B}{P_T \times 144} \qquad D_s = 20\ P_T \text{ (in.)}$$

$$C' = P_T - d_o \text{ (in.)}$$

$$= \frac{20\ P_T \times (P_T - d_o) \times 20\ P_T m}{144\ P_T} \qquad B = m\ D_s = 20\ P_T\ m \text{ (in.)}$$

$$= \frac{400\ m\ P_T\ (P_T - d_o)}{144} \qquad P_T = 0.800$$

$$d_o = 0.500$$

$$= 2.78 \times 0.800 \times 0.300 \times m \qquad 0.20 < m < 1.00$$

$$= 0.667 \times m$$

Equivalent diameter (Eq. 7.5)

$$D_e = \frac{1}{12} \times 4 \times \frac{(\frac{1}{2} P_T \times 0.86\ P_T - \frac{1}{2} \frac{\pi}{4} d_o^2)}{\frac{1}{2} \pi\ d_o}$$

$$D_e = \frac{1}{3} \times \frac{0.86\ P_T^2 - \frac{3.14}{4} d_o^2}{3.14\ d_o}$$

$$= \frac{1}{3} \times \frac{0.86 \times (0.80)^2 - 0.785 \times (0.50)^2}{3.14 \times 0.50}$$

$$= 0.0752 \text{ ft}$$

Mass velocity (Eq. 7.2)

$$G = \frac{W}{a_s}$$

$$W = \frac{3.413 \times 10^7}{0.46 \times 50} \quad \frac{\frac{\text{Btu}}{\text{hr}}}{\frac{\text{Btu}}{\text{lb °F}} \times \text{°F}}$$

$$= 1.484 \times 10^6 \frac{\text{lb}}{\text{hr}}$$

$$G = 2.22 \times 10^6 \times \frac{1}{m} \quad \frac{\text{lb}}{\text{hr}}$$

Reynolds modulus

$$Re = \frac{G\ D_e}{\mu}$$

$$= \frac{2.22 \times 10^6 \times 0.0752}{17.9 \times m}$$

$$= 9.35 \times 10^3 \frac{1}{m}$$

Heat transfer film coefficient

$$h = \frac{k}{D_e} \times (Pr)^{1/3} \times f\ (Re)$$

$$= \frac{2.75}{0.0752} (3.00)^{1/3} \quad f\ (Re)$$

$$= 52.6\ f\ (Re)$$ f (Re) to be read from Fig. 28 of Kern

Overall coefficient of heat transfer

$$\frac{1}{U} = \frac{1}{h_f} + \frac{1}{c}$$

where $\frac{1}{c} = 6.57 \times 10^{-4}$ (See Appendix II, p 20)

Heat transfer surface area

$$A = \frac{q_{DP}}{\Delta t_m} \times \frac{1}{U}$$

$$= \frac{3.1413}{1.33} \times 10^5 \frac{1}{U}$$

$$= 2.57 \times 10^5 \frac{1}{U}$$

$$= 2.57 \times 10^5 \frac{1}{U} \text{ ft}^2$$

Active heat transfer length of shell

$$L_T = \frac{A}{a}$$

$$= \frac{A}{43.2}$$

$a = 43.2 \text{ ft}^2/\text{ft}$

(See Appendix II, p 20)

$$= 0.0232 \text{ A ft}$$

Fuel holdup

$$v_f = A_f \times (L_T + 1)$$

$$= \left[(20 \ P_T)^2 \times \frac{3.14}{4} - (2 \times 165 + 20) \frac{3.14}{4} \times (0.5)^2 \right] \frac{1}{144} \times (L_T + 1)$$

$$= 0.92 \times (L_T + 1)$$

Shell side pressure drop

$$\Delta P = f \frac{G^2 \times D_s \ (N + 1)}{5.22 \times 10^{10} \times D_e \times s}$$

$N \;=\; \text{variable}$

$D_s \;=\; 20\ P_T \times \frac{1}{12} \;=\; 20 \times 0.8 \times \frac{1}{12} \;=\; 1.333\ \text{ft}$

$D_e \;=\; 0.752\ \text{ft}$

$s \;=\; \frac{154.3}{62.4} \;=\; 2.48$

$f \;=\; \phi\ (Re)$

$$\Delta P \;=\; \phi\ (Re)\ \frac{(2.22 \times 10^6\ \frac{1}{m})^2 \times 1.333 \times (N + 1)}{5.22 \times 10^{10} \times 0.0752 \times 2.48}$$

$$= 6.75 \times 10^2 \times (\frac{1}{m})^2 \times \phi\ (Re) \times (N + 1)$$

Table 1. Heat Transfer Calculations

m	$\frac{1}{m}$	$(\frac{1}{m})^2$	Re	f(Re)	$\frac{Btu}{ft^2\ hr\ °F}$ h	$\frac{1}{h} \times 10^4$	$\frac{1}{U} \times 10$	$\frac{Btu}{ft^2\ hr\ °F}$ U	ft^2 A_T	ft L
0.2	5.00	25	46,600	128	6740	1.48	8.05	1240	207	4.78
0.4	2.50	6.25	23,300	88.0	4640	2.16	8.73	1145	224	5.20
0.6	1.67	2.78	15,550	70.0	3680	2.72	9.29	1078	238	5.52
0.8	1.25	1.56	11,650	60.0	3160	3.16	9.73	1028	250	5.80
1.0	1.00	1.00	9,320	53.5	2820	3.55	10.12	988	260	6.00

Table 2. Pressure Drop Calculations

m	in. B	N	ϕ(Re)	(N+1)ϕ(Re)	$(\frac{1}{m})^2$ (N+1)ϕ(Re)	Psi ΔP	ft^3 v_f
.20	3.2	17.9	0.0016	0.0302	0.755	510	5.31
.40	6.4	9.74	0.0018	0.0194	0.0121	81.6	5.70
.60	9.6	6.90	0.00195	0.0154	0.0428	28.9	6.00
.80	12.8	5.42	0.00205	0.0132	0.0206	13.9	6.25
1.00	16.0	4.40	0.00215	0.0116	0.0116	7.83	6.31

Case 3B.

25% cut cross baffles, constant baffle pitch, variable tube pitch

Let the baffle pitch = 12 in.

and the tube pitch 0.700 in. $\leqslant P_T \leqslant$ 1.000 in.

using relations 7.1 - 7.5 of Kern (pp 138-139)

Geometry

Flow cross section (Eq. 7.1)

$$a_s = \frac{D_s \times C' \times B}{P_T \times 144}$$

$$= \frac{20\ P_T \times (P_T - 0.500) \times 12}{P_T \times 144}$$

$$= 0.834\ (m - 1)$$

$$D_s = 20\ P_T$$

$$C = P_T - 0.500$$

$$B = 12 \text{ in.}$$

$$P_T = m \times d_o$$

$$= 0.500\ m$$

$$1.4 \leqslant m \leqslant 1.8$$

Equivalent diameter (Eq. 7.5)

$$D_e = \frac{4}{12}\ \frac{0.86\ P_T^2 - 0.785\ d_o^2}{3.14\ d_T} = \frac{1}{3}\ \frac{0.86\ m^2\ d_o^2 - 0.785\ d_o^2}{3.14\ d_o}$$

$$= 0.106\ \frac{(0.86\ m^2 - 0.785)\ d_o^2}{d_o}$$

$$= 0.0531\ (0.86\ m^2 - 0.785)$$

$$= 0.0456\ (m^2 - 0.912)$$

Mass velocity

$$G = \frac{W}{a_s}$$

$$G = 1.484 \times 10^6 \times \frac{1}{0.834\ (m-1)}$$

$$= 1.78 \times 10^6\ \frac{1}{m-1}$$

$W = 1.484 \times 10^6$ lb/hr

(See Appendix III, p 27)

Reynolds modulus

$$Re = \frac{G\ D_e}{\mu}$$

$$= \frac{1.78}{1.79 \times 10} \times 10^6 \times \frac{1}{m-1} \times 4.56 \times 10^{-2} \ (m^2 - 0.912)$$

$$= 4.53 \times 10^3 \ \frac{m^2 - 0.912}{m-1}$$

Heat transfer film coefficient in shell side

$$h_f = \frac{k}{D_e} \times (Pr)^{1/3} \times f(Re)$$

$$= \frac{2.75 \times (3)^{1/3}}{0.0456\ (m^2 - 0.912)} \ f(Re)$$

$$= 87 \left(\frac{1}{m^2 - 0.912} \right) f(Re)$$

f(Re) to be read from Fig. 28, p 838 of Kern.

Overall coefficient of heat transfer

$$\frac{1}{U} = \frac{1}{h_f} + \frac{1}{c}$$

where $\frac{1}{c} = 6.57 \times 10^{-4}$ (See Appendix II, p 20)

Heat transfer surface area

$$A = 2.57 \times 10^5 \ \frac{1}{U}$$ (See Appendix III, p 28)

Active heat transfer length of shell

$$L_T = 0.0232\ A$$ (See Appendix II, p 20)

Fuel holdup

$$v_f = A_s \times (L + 1)$$

$$A_s = \frac{\pi}{4} D_s^2 - (2 \times 165 + 20) \frac{\pi}{4} d_o^2$$

$$= \frac{\pi}{4} \times (20 \text{ m } d_o)^2 - 350 \frac{\pi}{4} d_o^2$$

$$= \frac{\pi}{4} d_o^2 (400 \text{ m}^2 - 350)$$

$$= 0.785 \times \frac{(0.5)^2}{144} \times 400 (\text{m}^2 - 0.875)$$

$$= 0.545 (\text{m}^2 - 0.875)$$

$$v_f = 0.545 (\text{m}^2 - 0.875) (L + 1)$$

Shell side pressure drop

$$\Delta P = \frac{f \times G^2 \times D_s (N + 1)}{5.22 \times 10^{10} \times D_e \times s}$$

$$f = \phi (Re)$$

$$G = 1.78 \times 10^6 \frac{1}{m-1}$$

$$D_s = 20 P_T = 20 \times 0.500 \text{ m} \times \frac{1}{12}$$

$$= 0.834 \text{ m}$$

$$D_e = 0.0456 (\text{m}^2 - 0.912)$$

$$s = 2.48$$

$$\Delta P = \phi(Re) \frac{(1.78 \times 10^6 \times \frac{1}{m-1})^2 \times 0.834 \text{ m} \times (N + 1)}{5.22 \times 10^{10} \times 0.0456 (\text{m}^2 - 0.912) \times 2.48}$$

$$= \phi(Re) \times \frac{264}{0.589} \times \left(\frac{1}{m-1}\right)^2 \times \frac{m\ (N+1)}{(m^2 - 0.912)}$$

$$= 449 \quad \phi(Re) \times \left(\frac{1}{m-1}\right)^2 \times \frac{m\ (N+1)}{(m^2 - 0.912)}$$

Table 3. Heat Transfer

m	m-1	m^2	$\frac{1}{m^2-0.912}$	m^2-0.912	$\frac{Re}{4.53\times10^3}$	Re	$\frac{87}{m^2-0.912}$	f(Re)	h $\frac{Btu}{ft^2\ hr\ °F}$	$\frac{10^4}{h}$	$\frac{10^4}{U}$
1.4	0.40	1.96	0.954	1.05	2.62	11,900	82.8	60.0	4970	2.01	8.58
1.5	0.50	2.25	0.748	1.34	2.68	12,130	65.0	61.0	3970	2.52	9.09
1.6	0.60	2.56	0.606	1.65	2.75	12,450	52.8	62.0	3280	3.06	9.62
1.65	0.65	2.72	0.554	1.81	2.78	12,600	48.0	62.5	3000	3.33	9.90
1.70	0.70	2.89	0.505	1.98	2.83	12,820	44.0	63.0	2770	3.61	10.18
1.75	0.75	3.08	0.465	2.15	2.89	13,100	40.5	64.0	2590	3.86	10.43
1.80	0.80	3.24	0.429	2.33	2.91	13,200	37.3	64.5	2410	4.15	10.72

m	P_T in.	U $\frac{Btu}{ft^2\ hr\ °F}$	A ft^2	L ft	m^2-0.875	0.545 x (L+1)	v_f ft^3
1.4	0.70	1164	221	5.13	1.085	3.34	3.62
1.5	0.75	1100	234	5.43	1.375	3.50	4.81
1.6	0.80	1040	247	5.73	1.685	3.67	6.18
1.65	0.825	1010	254	5.90	1.845	3.76	6.94
1.70	0.85	982	262	6.08	2.015	3.86	7.77
1.75	0.875	958	268	6.22	2.185	3.94	8.61
1.80	0.90	933	276	6.40	2.365	4.03	9.54

Table 4. Pressure Drop

m	$\left(\frac{1}{m-1}\right)^2$	$m(N+1)$	$\frac{m(N+1)}{m^2 - 0.912}$	$f(m)$	$\times 10^{13}$ $\phi(Re)$	$449\ \phi(Re)$	psi ΔP
1.4	6.25	8.57	8.15	50.95	2.05	0.921	46.9
1.5	4.00	9.65	7.20	28.8	2.03	0.911	26.2
1.6	2.78	10.75	6.52	18.14	2.02	0.898	16.3
1.65	2.37	11.38	6.28	14.9	2.00	0.898	13.4
1.70	2.04	12.02	6.07	12.4	1.99	0.894	11.1
1.75	1.78	12.64	5.88	10.47	1.98	0.890	9.32
1.80	1.56	13.32	5.71	8.90	1.97	0.885	7.87

Appendix IV.

Temperature differentials at reduced power operation

Power reduction at constant flow conditions:

Assumed conditions:

$$P = m\, P_{DP}$$

$$V_{fuel} = \text{constant}$$

$$V_{coolant} = \text{constant}$$

$$A = \text{constant}$$

Deduced conditions:

$$\Delta t_{fuel} = m\Delta T$$

$$\Delta t_{coolant} = m\Delta t$$

$$U = \text{constant} = U_{DP}$$

$$\Delta t_m = m\, \Delta t_{m\ DP}$$

$$\Delta t_m = \frac{(T_o - t_i) - (T_i - t_o)}{\ln \dfrac{T_o - t_i}{T_i - t_o}}$$

$$T_o = T_i - \Delta T$$

$$t_i = t_o - \Delta t$$

$$\Delta t_m = \frac{(T_i - \Delta T) - (t_o - \Delta t) - (T_i - t_o)}{\ln \dfrac{(T_i - \Delta T) - (t_o - \Delta t)}{T_i - t_o}}$$

$$\Delta t_m = \frac{\Delta t - \Delta T}{\ln \left(1 + \dfrac{\Delta t - \Delta T}{T_i - t_o}\right)}$$

$$\ln \left(1 + \frac{\Delta t - \Delta T}{T_i - t_o}\right) = \frac{\Delta t - \Delta T}{\Delta t_m}$$

$$1 + \frac{m\Delta t - m\Delta T}{T_i - t_o} = e^{\frac{m\Delta t - m\Delta T}{m\Delta t_m}}$$

$$1 + m\,\frac{\Delta t - \Delta T}{T_i - t_o} = e^{\frac{\Delta t - \Delta T}{\Delta t_m}}$$

but by deduced assumptions

$$\left(e^{\frac{\Delta t - \Delta T}{\Delta t_m}} \right) - 1 = \text{constant}$$

therefore $T_i - t_o = \frac{m}{B} (\Delta t - \Delta T)$

and $T_o = T_i - m\Delta T$

$$t_i = t_o - m\Delta t$$

Thus Δt_m and all terminal temperatures vary linearly with power level.

<u>Power reduction at constant °T (fuel Δt) and constant coolant flow</u>

$$P = P_{DP} \times m$$

$$U = f(m)$$

$$\Delta t_m = \phi(m)$$

$$\Delta t_m = m\,\Delta t_{DP} = 75\ m$$

$$\Delta T = \text{const} = 50°F$$

$$T_m = \text{const} = 1200°F$$

$$V_{fuel} = m\,V_{DP}$$

$$q = U\,A\,\Delta t_m$$

$$\frac{q}{A} = U\,\Delta t_m$$

Let $\left(\frac{q}{A}\right)_m = m \left(\frac{q}{A}\right)_{DP}$

$$= m \times \frac{3.413 \times 10^7}{6 \times 43.2}$$

$$= 1.32 \times 10^5 \; m$$

Variation of U with power variation

$$\frac{1}{U} = \frac{1}{h_f} + 6.57 \times 10^{-4}$$ (See Appendix II, p 23)

$$h_f = \frac{k}{D_e} (Pr)^{1/3} \times f(Re)$$

$$f(Re) = 0.436 \; (Re)^{0.525}$$

if $2000 < Re < 15{,}000$

from Kern, Fig. 28.

The selected configuration of $P_T = 0.775$ in. gives

$$D_e = 0.0456 \left[\left(\frac{0.775}{0.500} \right)^2 - 0.912 \right]$$

$$= 0.0680 \text{ ft}$$

$$\frac{k}{D_e} (Pr)^{1/3} = \frac{2.75}{0.0680} (3.0)^{1/3}$$

$$= 58.4 \text{ Btu/ft}^2\text{-hr-°F}$$

at des. pt

$$Re = 12{,}300$$

$$f(Re) = 61.5$$

$$h_f = 3590 \text{ Btu/ft}^2\text{-hr-°F}$$

$$\frac{\left.h_f\right]_m}{h_f} = \frac{\left[\frac{k}{D_e} \times (Pr)^{1/3} \times 0.0436\ (Re)^{0.525}\right]_i}{\left[\frac{k}{D_e} \times (Pr)^{1/3} \times 0.0436\ (Re)^{0.525}\right]_{des}}$$

but $k_m = k_{DP}$

$$D_{e_m} = D_{e\ DP}$$

$$Pr_m = Pr_{DP}$$

$$\frac{\left.h_f\right]_m}{h_f} = \frac{Re_m^{\ 0.525}}{Re_{DP}}$$

$$Re_m = (m)\ Re_{DP}$$

$$\left.h_f\right]_m = h_f\ (m)^{0.525}$$

$$\frac{1}{U} = \frac{1}{h_f} \times \left(\frac{1}{m}\right)^{0.525} + 6.57 \times 10^{-4}$$

and $\Delta t_m = 1.32 \times 10^{5-4} \times m \left[2.78 \times \left(\frac{1}{m}\right)^{0.525} + 6.57 \right]$

$$= 36.7\ m \left[\left(\frac{1}{m}\right)^{0.525} + 2.36 \right]$$

$$\Delta t_m = \frac{\Delta t - \Delta T}{\ln\left(1 + \dfrac{\Delta t + \Delta T}{T_i - t_o}\right)}$$

$$1 + \frac{\Delta t - \Delta T}{T_i - t_o} = e^{\frac{\Delta t - \Delta T}{\Delta t_m}}$$

$$T_i - t_o = \frac{\Delta t - \Delta T}{e^{\left(\frac{\Delta t - \Delta T}{\Delta t_m}\right.} - 1\Big)}$$

Table 5. Mean Δt for $0.1\ P_{des} < P < P_{des}$

m	$\left(\frac{1}{m}\right)^{0.525}$	$2.36 + \left(\frac{1}{m}\right)^{0.525}$	$36.7\ m$	°F Δt_m
1.0	1.00	3.364	36.7	124
0.8	1.124	3.488	29.4	102
0.6	1.308	3.672	22.0	80.8
0.4	1.618	3.982	14.7	58.5
0.2	2.325	4.689	7.34	34.4
0.1	3.350	5.714	3.67	21.0

Table 6. Terminal Temperatures

m	$\Delta t-\Delta T$	$\frac{\Delta t-\Delta T}{\Delta t_m}$	$e^{\frac{\Delta t-\Delta T}{\Delta t_m}}$	$e^{\frac{\Delta t-\Delta T}{(\Delta t_m-1)}}$	T_i-t_o °F	T_i °F	t_o °F	T_o °F	t_i °F
1.0	25	0.202	1.224	0.224	112	1225	1113	1175	1038
0.8	10	0.0975	1.102	1.025	97.5	1225	1127.5	1175	1067.5
0.6	- 5	-0.0619	0.940	-0.060	83.3	1225	1141.7	1175	1096.7
0.4	-20	-0.342	0.710	-0.290	69.0	1225	1156	1175	1126
0.2	-35	-1.02	0.362	-0.638	54.8	1225	1170	1175	1155
0.1	-42.5	-2.02	0.133	-0.861	49.0	1225	1176	1175	1168.5

Distribution

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