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RANGE OF ALLOWABLE OPERATING
CONDITIONS FOR FFTF CLOSED LOOPS

October 1969

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ERRATA

Change Figure A-1 on Page A-1a to show at equally spaced intervals on the ascending vertical scale the numbers 500, 600, 700, 800, and 900, representing inadvertently omitted designations of minimum allowable loop coolant temperature in degrees Fahrenheit.

CHAPTER II

The first part of the book is devoted to a general survey of the history of the subject. It begins with a discussion of the early attempts to explain the origin of life, and then proceeds to a more detailed examination of the various theories which have been advanced. The second part of the book is devoted to a study of the various forms of life which have appeared on the earth, and the third part to a consideration of the various factors which have influenced the development of life.

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RANGE OF ALLOWABLE OPERATING
CONDITIONS FOR FFTF CLOSED LOOPS

By

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October 1969

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RANGE OF ALLOWABLE OPERATING
CONDITIONS FOR FFTF CLOSED LOOPS

ABSTRACT

The purpose of this report was to determine the testing capability of the closed loop system for FFTF (Fast Flux Test Facility). The closed loop system consists of an in-reactor tube, a primary heat transport system used to transfer heat from the irradiation experiment to the sodium coolant, and a secondary heat transport system for receiving and dumping heat from the primary system to the atmosphere.

The closed loop system provides an isolated circuit in the test reactor for the testing of fast reactor fuels, materials, and coolants at conditions other than those provided by the reactor environment. The range of tests that can be safely conducted in the closed loop system is limited by constraints such as coolant flow rate, stress, temperature, and material limitations. This study was undertaken to determine the effect of these constraints on testing capability and to determine the operational testing envelope for the system.

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RANGE OF ALLOWABLE OPERATING CONDITIONS FOR FFTF CLOSED LOOPS

INTRODUCTION

This report was prepared by Battelle-Northwest under Contract AT(45-1)-1830 for the Atomic Energy Commission, Division of Reactor Development and Technology, to describe results of work performed to establish the range of allowable operating conditions for the FFTF closed loops.

SUMMARY

Within limits imposed by allowable temperatures of reactor components, and allowable limits of sodium velocity, the range of allowable operating conditions for FFTF closed loops was established. Test powers from 0.5 to 2.75 MW are permissible with a coolant ΔT of 200 °F. Test powers from 0.5 to 4.3 MW are permissible with a coolant ΔT of 300 °F. Test powers from 1.0 to 5.0 MW (5 MW was the maximum power investigated) are permissible with a coolant ΔT of 400 °F.

The major limiting factor on closed loop operation was determined to be the maximum allowable temperature difference (assumed to be 500 °F) between the inlet and outlet in the top section of the closed loop. It is recommended that further analytical work be conducted to establish this allowable temperature range in a more precise manner. Bypass flow should also be investigated to determine its effect on increasing testing capability.

DISCUSSION

OBJECTIVE OF STUDY

This study was conducted for the purposes of (1) computing the effect of variations in test power, sodium flow rate and DHX air flow rate on FFTF closed loop test ΔT and bulk coolant

temperature leaving the test section and (2) using the thermal data so generated to develop estimates of probable operating limits of the FFTF closed loops.

DESCRIPTION OF FFTF CLOSED LOOPS

The closed loops have been described in the CSDD for System 61.⁽¹⁾ The loop contains a re-entrant test section, electric heater, intermediate heat exchanger, dump heat exchanger, pumps, connecting piping, sodium storage tank, and purification system. The test loop consists of a primary and secondary loop. The test in-reactor tube is an assembly of nested concentric tubes with access provided at the top. The intermediate heat exchanger is a shell and tube counter-flow exchanger. The dump heat exchanger is a counter-cross-flow exchanger. A schematic of the system is shown in Figure 1. Parameters of system components are tabulated in Appendix B.

METHOD OF INVESTIGATION

The computations were made by TAP,⁽²⁾ a stable thermal analyzer code for closed hydraulic systems.

RESULTS AND CONCLUSIONS

Initial Computer Runs

The initial results of the study in Table 1 summarize the effects on test ΔT and bulk coolant outlet temperature for variations in loop coolant flow rate and DHX air flow rate.

Test powers from 0.5 to 5.0 MW; ΔT 's of 200, 300 and 400 °F; and bulk outlet temperatures from 800 °F to 1200 °F were investigated. These data thus cover the full design range of the FFTF closed loops.

FIGURE 1. Closed Loop Heat Removal Diagram

TABLE 1. Summary of Computer Runs

Run No.	Test Power, MW	Primary and Secondary Loop Sodium Flow, lb/sec	Test ΔT , °F	Bulk Coolant Temperature Out of Test, °F	DHX Air Flow Rate, lb/sec
1	5	78.9	201	1126	44
2	5	76	204	939	80
3	5	76	203	817	130
4	5	52.6	302	1163	48
5	5	52.6	292	852	150
6	5	52.6	307	831	200
7	5	39.49	400	1222	48.75
8	5	39.49	404	917	170
9	5	39.49	404	892	220
10	3	23.7	396	1101	25
11	3	23.7	389	966	40
12	3	23.7	400	896	60
13	3	31.6	302	1162	18
14	3	31.6	298	1038	25
15	3	31.6	299	823	60
16	3	47.4	202	1188	15
17	3	47.4	199	1008	23
18	3	47.4	196	840	40
19	0.5	3.9	406	1100	3.5
20	0.5	3.9	398	999	5
21	0.5	3.9	396	885	10
22	0.5	5.26	305	1117	2.5
23	0.5	5.1	306	961	4
24	0.5	5.1	303	807	7
25	0.5	7.89	211	1044	2.5
26	0.5	7.75	206	877	4
27	0.5	7.75	206	741	6

Estimated Constraints - Major Constraints

Limiting operation conditions for the test loops were identified and values were estimated, as follows:

- Minimum sodium coolant exit temperature of 400 °F to prevent sodium plugging in the DHX
- Maximum allowable coolant velocity in the test section (based on experience with sodium erosion and corrosion) 30 ft/sec
- 1200 °F assumed maximum allowable temperature leaving the in-reactor tube
- 500 °F assumed maximum allowable temperature difference between inlet and outlet of test section top works.

DHX air flow was also investigated as a potential limiting feature, but best available information indicates that no limit is imposed by this parameter.

Range of Allowable Operating Conditions for Closed Loops

Based on the thermal data and the constraints identified in the foregoing, the range of allowable operating conditions for FFTF closed loops was computed. These conditions, shown graphically in Appendix A, are tabulated as follows:

$\Delta T = 200$ °F

- The maximum allowable test power is ≈ 2.75 MW.
- Allowable bulk coolant temperatures leaving the test range from < 800 °F to 1200 °F
- Low power tests (0.5 MW) are admissible.

$\Delta T = 300$ °F

- Admissible test powers range from 0.5 MW to 4.3 MW.
- Allowable test exit coolant temperatures range from 800 °F to 1200 °F.

$\Delta T = 400$ °F

- The maximum test power investigated is 5.0 MW
- The minimum allowable test power is 1.0 MW

- The minimum allowable bulk coolant temperature exiting the test is 900 °F
- The maximum allowable bulk coolant temperature exiting the test is 1200 °F.

Significance of Results

It was found that the major factor affecting operating limits for the closed loop 400 °F test ΔT 's was the value of 500 °F maximum allowable temperature difference estimated between the inlet and outlet in the test section top works. This limitation, representing a very serious compromise in test capability, should be thoroughly investigated to determine actual validity of the 500 °F temperature difference. Bypass flow should also be investigated to determine its effect on reducing the top works ΔT .

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APPENDIX A

DETAILS OF ALLOWED OPERATING
CONDITIONS FOR CLOSED LOOPSEVALUATION OF VARIABLES

Figure A-1 shows the minimum system coolant temperature as a function of the test fuel exit coolant temperature for given test powers and test ΔT 's. For a minimum allowable coolant temperature in the loop of 400 °F, a line is drawn perpendicularly to the ordinate representing this value. The test fuel minimum coolant exit temperature for a given test power and test ΔT is determined at the point where the line representing the limiting constraint is intersected. For example, a 5.0 MW test with 400 °F ΔT intersects the limiting constraint at a coolant exit temperature out of the fuel of 925 °F. The 5 MW of power and the test exit temperature of 925 °F represent one point on the so-called testing envelope diagram (see Figures A-2, A-3, A-4). This procedure is repeated for test powers of 3.0 MW and 0.5 MW. The three resulting points represent a line on the testing envelope for a given test ΔT describing the minimum allowable coolant temperature leaving the test fuel.

Figure A-5 shows test power as a function of top works ΔT for given test ΔT 's and test exit coolant temperature. A line representing the maximum allowable top works ΔT is drawn perpendicularly to the abscissa. The points at which the curves on the graph intersect this line represent those test conditions producing top works ΔT 's equal to the allowable limit. These points are identified in terms of fuel power, test ΔT , and test exit coolant temperature.

Figure A-6 shows the top works exit sodium coolant temperature as a function of the test coolant outlet temperature for given test ΔT 's and test powers. The line drawn

FIGURE A-1. Minimum Allowable Closed Loop Coolant Temperature as Function of Coolant Temperature Leaving Test Fuel for Given Test Powers and Test ΔT 's

FIGURE A-2. Test Power as Function of Coolant Temperature Leaving Test Fuel for 200 °F Test ΔT 's

FIGURE A-3. Test Fuel Power as Function of Coolant Temperature Leaving Test Fuel for 300 °F Test ΔT 's

FIGURE A-4. *Test Power as Function of Sodium Temperature Leaving Test Fuel for 400 °F Test ΔT 's*

FIGURE A-5. Test Fuel Power as Function of Top Works ΔT for Given Test ΔT 's and Bulk Test Outlet Coolant Temperatures

FIGURE A-6. Top Works Exit Coolant Temperature as Function of Test Fuel Coolant Exit Temperature for Given Test Powers and Test ΔT 's

perpendicular to the ordinate represents the maximum allowable sodium coolant temperature leaving the top works. The intersections of this line with the curves on the graph represent the maximum allowable test fuel exit coolant temperature for a given test ΔT and test power. These points are replotted on the appropriate test envelopes.

TESTING ENVELOPES

Testing envelopes obtained for section ΔT 's of 200 °F, 300 °F, and 400 °F are shown in Figures A-2, A-3, and A-4. The envelopes are expressed in terms of test power as a function of the bulk coolant temperature leaving the test. These testing envelopes describe a range of admissible testing compatible with the assumed values of the identified limiting system constraints.

APPENDIX B

DESCRIPTION OF SYSTEM MODELSYSTEM THERMAL MODEL DESCRIPTION

The simulation model selected included only those components that participate in 100% of the sodium flow most of the time. Such items as the sodium purification system and backup subsystem are excluded. A simplified nodal diagram of the simulation model is shown in Figure B-1. The major assumptions included:

1. Sodium convective heat transfer coefficients in the in-reactor tube are on the order of 30,000 Btu/hr-ft²-°F.
2. Reactor sodium pool is at 800 °F.
3. Air inlet to dump heat exchanger is 100 °F at standard barometric pressure.
4. The length of piping runs and their insulation thickness are as shown for System 61 of the CSDD (Table B-1).
5. The size and surface area of the pump wells as well as the heater design is as shown for System 61⁽³⁾ of the CSDD (Tables B-2, B-3, and B-4).
6. The design of the test loop test section is as shown by drawings SK-3-14466, SK-3-14602, SK-3-14465, and SK-3-14464, included at the end of this appendix.

TABLE B-1. *Piping Description*

<u>From</u>	<u>To</u>	<u>Length, ft</u>	<u>Size, in.</u>	<u>Insulation Thickness, in.</u>
In-Reactor tube	IHX	180.0	5 sch. 40	5.0
IHX	Pump	33.0	5 sch. 40	5.0
Pump	Heater	54.0	5 sch. 40	5.0
Heater	In-Reactor tube	155.0	5 sch. 40	5.0
IHX	DHX	180.0	5 sch. 40	5.0
DHX	IHX	180.0	5 sch. 40	5.0

TABLE B-2. Primary Pump*

OD, ft	5.0
Height, ft	6.0
Wall thickness, in.	0.25
Insulation thickness, in.	5.0

TABLE B-3. Secondary Pump*

OD, in.	30.0
Height, ft	5.5
Wall thickness, in.	0.25
Insulation thickness, in.	5.0

TABLE B-4. Heater

Heater output	
Design heat, kW	750
Normal operating heat, kW	30 to 500
Heater shell	
Material	304 SS
ID, in.	23.500
OD, in.	24.000
Length, ft	4.5
Heater exit temperature, °F	1200 max.
Heater rods	
Material and type	Calrod, SS sheathed
Diameter, in.	1.0
Number	100
Length, ft	5 (6 in. cold end)
Heater wells	
Material	304 SS
ID, in.	1.097
OD, in.	1.206
Internal length, ft	4.0
Insulation	
Material	Johns-Manville Superex-M block
Lagging	Asbestos cloth
Thickness, in.	5.0

* Approximated by a circular tank.

FIGURE B-1. Closed Loop Thermal Model

A node is a region of constant properties. This region may be, for example, a section of pipe wall or a specified volume of coolant. The size of the node is dictated by the accuracy desired of the model. As included in this model, the nodes may represent the average temperature of the region under consideration. Items removed from the diagram but included in the simulation model are trace heating, piping, and other components providing heat leakage paths to ambient conditions.

Component sizes, nuclear fuel description, coolant flow rate, fuel power, heater power, and material properties such as thermal conductivity and specific heat are entered in the code separately from the node connections (network description). This network description as coded for this model in the TAP code is, therefore, usable for any sized loop with a similar interconnection scheme embodying such components as inter-connecting piping, counterflow intermediate heat exchanger, and counter-cross flow dump heat exchanger.

DESCRIPTION OF SIMULATION MODEL COMPONENTS

The computer model of the in-reactor tube is a re-entrant design with five sodium fluid nodes in the downcomer and four sodium fluid nodes in the riser. The gas gap above the core is insulated with argon gas. Both conduction and radiation are considered as modes of heat transfer in this region. The double walls in the core region and below the core region are insulated with a material of presumed thermal conductivity of 2 Btu/hr-ft-°F. This insulation allows the removal of decay heat during loop shutdown (i.e., no flow) without excessively high temperatures in the test. Convective heat transfer coefficients are disregarded inside the in-reactor tube because the resistance to heat flow induced by convection is small compared to the resistance induced by conduction through the insulated double walls. The sodium film coefficient on the in-reactor tube surface in contact with the sodium pool was assumed to be

1000 Btu/hr-ft²-°F. The reactor sodium pool temperature is assumed to be 800 °F. Gamma heating of 8 W/g for the materials in the core region during operation yields a power output of 0.38 MW.

The computer model of the counterflow intermediate heat exchanger uses four sodium fluid nodes on both the primary and secondary sides. The computer code also considers parasitic heat losses through the insulation on the IHX.

The cross counterflow dump heat exchanger is modeled using four fluid nodes on both the sodium side and the air side. Fin efficiency is used to evaluate the thermal performance of the extended surfaces on the tubes. Air-side film coefficients on the tube fins are obtained from information recommended by the manufacturers of extended surface tubes.⁽⁴⁾ Sodium film coefficients are neglected.

One sodium fluid node is used in modelling the primary heater, the interconnecting piping, and the primary and secondary pumps. Parasitic heat losses through the insulation are transferred to an ambient sink at a temperature of 70 °F.

PARAMETRIC STUDY

The parametric study considers, in the flow ratio primary to secondary (mp/ms = 1), the following combinations of variables:

<u>Fuel Power</u>	<u>Test ΔT</u>	<u>Coolant Temperature Out of Test Fuel</u>
5.0 MW	400 °F	1200 °F
3.0 MW	300 °F	1000 °F
0.5 MW	200 °F	800 °F

These variables result in the 27 computer runs shown previously in Table 1.

HEAT EXCHANGERS

A baseline design of 5.0 MW with a 400 °F coolant temperature rise across the test and a bulk sodium coolant temperature of 1200 °F leaving the test fuel was selected as the design point for this study. The primary and secondary closed loop flow rates were assumed to be equal. The IHX has a log mean temperature difference of 100 °F. The dump heat exchanger has design air inlet and exit temperatures of 100 °F and 500 °F, respectively, and a design heat rejection rate of 5 MW.

To achieve heat exchanger sizes compatible with the overall system, Kays and London's number of transfer units (NTU) and heat exchanger effectiveness relationship was used. The NTU relationship (AU/mpCp) is a measure of the heat transfer size of the exchanger.

A counterflow IHX with a heat transfer effectiveness of 80% is required to achieve a log mean temperature difference of 100 °F when primary and secondary flow rates are equal. The desirability of the counterflow IHX is based on the least amount of heat transfer area required to transfer the 5 MW of heat. The number of transfer units is 4.0.⁽⁵⁾

The required heat transfer area can be determined directly from the NTU relationship once the overall heat transfer coefficient is calculated (see sample calculations). Table B-5 describes the IHX design.

TABLE B-5. *IHX Design*

Type	Counterflow
Heat Transfer Area, ft ²	146.8
Tube OD, in.	5/8
Tube Thickness, in.	0.049
Tube Length, ft	14.70
No. of Tubes	61
Material	Type 304 SS
Shell Side ΔP psi	0.4
Tube Side ΔP psi	1.5

The dump heat exchanger is a cross counterflow design with an effectiveness of 40%.* The number of transfer units is equal to 0.6. The overall heat transfer coefficient is:

$$U = \frac{1}{\frac{A}{n_{to} h_o} + R_w A}$$

U = overall heat transfer coefficient

A = heat exchanger area

R_w = thermal resistance of the wall

n_{to} = total efficiency of the outer surface

h_o = average unit conductance for outer surface

Once the overall heat transfer coefficient is calculated, the DHX heat transfer area can be determined from the NTU relationship.* $NTU = AU/C_{min}$: $C_{min} = \frac{M_s C_p}{M_{air} C_p}$. Table B-6 summarizes the DHX design.

TABLE B-6. DHX Design

<u>Type</u>	<u>Cross-Counterflow</u>
Tube arrangement	inline design
No. of pipes/row	
across bank	9.0
Pipe length in bank, ft	5
No. of parallel passes	4
Pipe OD, in.	1.9
Fin size, in.	0.75 x 0.05
Fin pitch fins/in.	5
Heat transfer area on	
air side, ft ²	2156
Material	Type 304 SS

* See sample calculations.

DHX NTU'S CALCULATIONS

The following are the sample calculations:

Mp = Primary Loop Flow Rate, lb/sec

Ms = Secondary Loop Flow Rate, lb/sec

Ma = Air Flow Rate, lb/sec

T = Temperature, °F

$$\epsilon_{IHX} = \frac{MsCp (T_4 - T_3)}{MpCp (T_1 - T_3)} \quad (1)$$

$$\frac{MsCp}{MpCp} = 1.0 \quad (2)$$

$$\epsilon_{IHX} = \frac{T_4 - T_3}{T_1 - T_3} \quad (3)$$

$$\epsilon_{DHX} = \frac{MsCp (T_4 - T_3)}{MaCp (T_4 - T_5)} \quad (4)$$

From Equation (3)

$$T_4 = T_3 + \epsilon_{IHX} (T_1 - T_3) \quad (5)$$

Substitute Equation (5) into (4)

$$\epsilon_{\text{DHX}} = \frac{\text{MsCp} (T_4 - T_3)}{\text{MaCp} [T_3 + \epsilon_{\text{IHX}} (T_1 - T_3) - T_5]} \quad (6)$$

for $\frac{\text{MpCp}}{\text{MsCp}} = 1.0$

$$\text{LMTD} = 100 \text{ } ^\circ\text{F} = (T_1 - T_4) = (T_2 - T_3)$$

$$T_1 = 1200 \text{ } ^\circ\text{F}$$

$$T_2 = 800 \text{ } ^\circ\text{F}$$

$$T_3 = 700 \text{ } ^\circ\text{F}$$

$$T_4 = 1100 \text{ } ^\circ\text{F}$$

Substituting temperatures referenced in the foregoing into Equation (6) yields $\epsilon_{\text{DHX}} = 40\%$. Since $\frac{\text{MsCp}}{\text{MaCp}} = 1$,⁽⁵⁾ NTU ≈ 0.6 .

OVERALL IHX HEAT TRANSFER COEFFICIENT CALCULATIONS

Design dimensions and conditions are as follows:

Tube size, OD, in.	= 5/8
Wall thickness, in.	= 0.049
P/D	= 1.4
P, in.	= 0.875
No. tubes	= 61
LMTD, $^\circ\text{F}$	= 100
Shell ID, in.	= 8.125
K_{Na} , Btu/hr-ft- $^\circ\text{F}$	= 37.8
CP_{Na} , Btu/lbm- $^\circ\text{F}$	= 0.3
M_{Na} , lbm/sec-ft	= 0.00014
ρ_{Na} , lbm/ft ³	= 51.2
M_p , lbm/hr	= 142,166
M_s , lbm/hr	= 142,166
A shell, ft ²	= 0.2299
A tube, ft ²	= 0.1298

Calculations of the overall IHX heat transfer coefficient are as follows:

$$V = \frac{M}{\rho A} \quad (1)$$

$$V_s = \frac{142,166}{(3600) (51.2) (0.2299)} = 3.35 \text{ ft/sec}$$

$$V_t = \frac{142,166}{(3600) (51.2) (0.1298)} = 5.9 \text{ ft/sec}$$

$$D_{eq \text{ Shell}} = \frac{4 A_s}{\pi [D_{shell} + (N) D_{tubes}]} \quad (2)$$

$$= \frac{4(0.2299) (12)}{\pi (8.125 + 61) (0.625)} = 0.0759 \text{ ft}$$

$$D_{eq \text{ Tubes}} = 0.0439 \text{ ft}$$

$$Re = \frac{(D_{eq}) (V) (\rho)}{u} \quad (3)$$

$$Re_{shell} = \frac{0.0439 (5.9) (51.2)}{0.00014} = 93094$$

$$Re_{tube} = \frac{0.0759 (3.35) (51.2)}{0.00014} = 94723$$

$$Pe_{shell} = (Re)(Pr) = 93094 (0.004) = 372.9 \quad (4)$$

$$Pe_{tube} = 94723 (0.004) = 378$$

$$NU_{shell} = 0.625 Pe^{0.4} = 6.67 \quad (5)$$

$$NU_{tube} = 6.7$$

$$h_{shell} = \frac{NU(K)}{D_{eq}} = 3322 \quad (6)$$

$$h_{tube} = 5769$$

$$R_w = \text{Tube Wall Resistance} = \frac{s}{K} \left(\frac{2.0(\text{OD})}{12} \right) = \frac{0.049}{12(12.5)} \left(\frac{2.0(0.625)}{0.625 + 0.527} \right) \quad (7)$$

$$= 3.54 \times 10^{-4} \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}$$

$$\frac{1}{U} = \left(\frac{1(\text{OD})}{K_1(\text{ID})} + R_w + \frac{1}{R_o} \right) \quad (8)$$

$$\frac{1}{U} = \left(\frac{0.625}{5769(0.527)} + 3.54 \times 10^{-4} + \frac{1}{3322} \right)$$

$$U = 1161$$

IHX SIZE CALCULATION

$$\epsilon_{\text{IHX}} = 80\%$$

$$\frac{M_p C_p}{M_s C_p} = 1$$

For a Counter Flow Heat Exchanger, the Ntu's = 4⁽⁵⁾

$$4 = \frac{AU}{M_p C_p}$$

$$146.8 = \frac{4(142166)(0.3)}{1161} = \text{area (ft}^2\text{)}$$

$$\text{Length} = \frac{146.8(12)}{61(3.14)(0.625)} = \frac{A}{(N)(\pi)(D)} = 14.2 \text{ ft}$$

IHX PRESSURE DROP CALCULATION

$$K_1 = \text{Expansion Coefficient}$$

$$K_2 = \text{Contraction Coefficient}$$

$$Re_t = 94723$$

$$f_{\text{smooth tube}} = 0.0186$$

$$\Delta P_{\text{tube}} = \frac{\rho V t^2}{2g} \left(\frac{f(L)}{ID} + K_1 + K_2 \right)$$

$$\Delta P_{\text{tube}} = \frac{(51.2)(5.9)^2}{288(32.2)} \left(\frac{0.0186(14.7)}{0.0439} + 1.0 + 0.5 \right)$$

$$= 1.48 \text{ psi}$$

$$\Delta P_{\text{shell}} = \frac{\rho V_{\text{shell}}^2}{2g} \left(2K_1 + 2K_2 + \frac{fL}{D_{\text{eq}}} \right)$$

$$\Delta P_{\text{shell}} = \frac{51.2(3.35)^2}{144(64.4)} \left(2(1) + 2(0.5) + \frac{0.0186(14.7)}{0.0759} \right)$$

$$= 0.4 \text{ psi}$$

DHX OVERALL HEAT TRANSFER COEFFICIENT CALCULATION

$C_{\min} = \frac{M_p C_p}{M_a C_p}$	= 1.0
A_{Fins}	= 5.49 ft ² /ft
M_a	= 175,514 lb/hr
A_{Tube}	= 0.497 ft ² /ft
Tube OD, in.	= 1.9
Fin Height, in.	= 0.75
Fin Thickness, in.	= 0.05
Pitch, fin/in.	= 5.0
Number of Parallel Passes	= 4.0
Length of Tube, ft	= 5.0
DHX Effectiveness	= 0.40
Distance Between Tube Centers, in.	= 4.4

NOMENCLATURE

A	= Air Side Heat Transfer Area
F	= Tube Length in Banks
G	= Mass Flow Rate, lb/hr-ft ²
h _o	= Air Side Film Coefficient of Heat Transfer, Btu/hr-ft ² -°F
L	= Fin Height
N	= Number of Tubes
NFA	= Net Free Air of Tube Banks, ft ²
P	= Fin Pitch
S	= Distance Between Tube Centers
U	= Overall Heat Transfer Coefficient, Btu/hr-ft ² -°F

ϵ = Fin Efficiency

$$\begin{aligned} \text{NFA} &= F(N) [0.0833 (S-\text{do}) - 0.167 (L)(N)(P)] \\ &= 5(9) [0.0833 (4.4 - 1.9) - 0.167 (0.75)(0.05)(5)] \\ &= 7.96 \end{aligned}$$

$$G = \frac{m}{\text{NFA}} = \frac{175,514}{7.96} = 22,048$$

$$h = 29^{(4)}$$

$$h_o = h \left[\left(\frac{K_g}{M_g^{0.6}} \right) \left(\frac{C_g M_g}{K_g} \right)^{1/3} / \left(\frac{K}{M^{0.6}} \right) \left(\frac{C_M}{K} \right)^{1/3} \right]$$

$$\begin{aligned} h_o &= 29 \left[\left(\frac{0.0174}{(0.0214)^{0.6}} \right) \left(\frac{0.241(0.0214)}{0.0174} \right)^{1/3} / \right. \\ &\quad \left. \left(\frac{0.016}{(0.019)^{0.6}} \right) \left(\frac{0.244(0.019)}{0.016} \right)^{1/3} \right] \end{aligned}$$

$$h_o = 29.5$$

FIND FIN EFFICIENCY⁽⁵⁾

$$\text{ro-ri} \sqrt{\frac{2h_o}{K_g}} = \frac{0.75}{12} \sqrt{\frac{2(12)(29.5)}{0.05(12)}}$$

$$= 2.149$$

$$\epsilon = 38\%$$

$$A_t = A_{\text{Fin}} + A_{\text{Tube}}$$

$$A_t = 5.49 + 0.497 = 5.99 \text{ ft}^2/\text{ft}$$

$$U = \frac{h_o}{A_t} (\epsilon A_F + A_o)$$

$$U = \frac{29.4}{5.99} [0.38 (5.49) + 0.497] = 12.67$$

DHX SIZE CALCULATION

$$\epsilon_{\text{DHX}} = 0.40$$

For a Cross-Counter flow exchanger without mixing the NTU's = 0.60⁽⁵⁾

$$\text{NTU} = \frac{A(U)}{C_{\min}}$$

$$\frac{0.6(175,514)(0.243)}{12.91} = \text{Area ft}^2$$

$$\underline{2018} = \text{Area ft}^2$$

- GENERAL NOTES:
1. UNLESS OTHERWISE SPECIFIED, DIMENSIONS ARE IN INCHES
TOLERANCES: FRACTIONS $\pm \frac{1}{32}$ DECIMALS $\pm .005$
ANGLES $\pm 1^\circ$ FINISH $\frac{32}{\sqrt{R}}$
 2. UNLESS OTHERWISE SPECIFIED, MATERIAL IS AISI 316 SST.
 3. APPLY HARD FACING $\frac{3}{16}$ THICK AND MACHINE TO $\frac{1}{16}$ THICK
WITH $\frac{1}{16}$ FINISH, (HAYNES STELLITE NO. 6B) PER
MANUFACTURERS INSTRUCTIONS.
 4. SURFACE ROUGHNESS IN ACCORDANCE WITH ASA B 46.1-1962
 5. BREAK ALL SHARP EDGES AND REMOVE ALL BURRS.
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