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SUGGESTIONS FOR WORK ON DESIGN OF HEAT EXCHANGERS IN LIQUID METAL FUELLED REACTORS

by

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SUMMARY

The report discusses the low heat transfer film coefficient on the secondary coolant side of tubular exchangers used in the calculations in AAEC/E.1. It suggests the types of heat exchangers which should be investigated in order to improve this coefficient and in turn to reduce the relative hold-up.

In AAEC/E. 1, the value assumed for the heat transfer film coefficient on the secondary coolant side is 1.70 watts/(cm^2)(^{o}C) equivalent to 3000 B. Th. U/(hr)(ft^2)(^{o}F). This is the value used in (1). At first sight this value appears to be low. In (2) it is stated that there are few experimental data for the prediction shell side heat transfer coefficients for liquid metals and because of its complexity, no satsifactory analytical solution has been found. A generalized expression

$$\frac{h\ D_t}{k} \approx S\left(\frac{W}{D_t\ \mu}\right)^b$$

is given where

b = approximately 0.6 but no value for S is reported

h = shell side heat transfer coefficient

D_t = outside diameter of tubes

k = thermal conductivity

W = weight rate of flow

μ = dynamic viscosity

Using generalizations obtained from data with ordinary fluids, such as that of Donahue (3), leads to predictions of Nusselt modulus which may be as much as a factor of 10 above the values found by experiment.

The values of the overall coefficient of heat transfer arising from the calculations reported in AAEC/E. 1 range,

(a) between 0.56 and 1.47 watts/(cm^2)(^{o}C) for Na/Stainless steel,

(b) between 0.94 and 1.97 watts/(cm^2)(^{o}C) for Na/Niobium

(c) between 0.43 and 1.17 watts/(cm^2)(^{o}C) for Bi/Croloy.

In (4), the following table for the performance of liquid metal to liquid metal heat exchangers is given.

	Description	Duty B. Th. U	U B. Th. U/(hr) (ft^2) (^oF)	U watts / (cm^2) (^oC)
1.	Shell and tube - unbaffled shell concentric tubes - SS tubes Na third fluid	3,700,000	1,000	0.57
2.	Shell and tube - unbaffled shell - single wall tubes - SS tubes - very small exchangers	90,000	2,200	1.25
3.	Shell and tube - baffled shell - single wall - SS tubes - very small exchanger	100,000	2,500	1.42
4.	Shell and tube - baffled shell - single wall - SS tubes	3,500,000	1,900	1.08
5.	"Hockey Stick" exchanger - flattened nickel tubes - Na third fluid	4,000,000	2,000	1.14
6.	"Hockey Stick" exchanger - round SS tubes - NaK third fluid	7,000,000	400	0.23

From this it is inferred that the assumption for the value of the heat transfer film coefficient on the secondary coolant is not far from the truth.

The following table taken from (4) compares the designs for liquid metal to liquid metal heat exchangers.

	Type	Advantages	Disadvantages
1.	Shell and tube unbaffled shell	a. low shell side pressure drop	a. poor tube bundle penetration b. low shell side coefficient c. requires "U" tubes or expansion joint
2.	Shell and tube baffled shell	a. High shell side coefficient	a. high shell side pressure drop b. requires "U" tubes or expansion joint
3.	Shell and Tube concentric tubes	a. as for 1 and 2 b. prevents inter-leakage	a. high tube wall resistance b. maintenance on inner tube sheet difficult
4.	"Hockey Stick" flat tubes	a. low pressure drop b. prevents interleakage c. reasonable overall coefficient 3. all tube sheets are accessible by removing covers	a. expensive fabrication b. large overall size
5.	"Hockey Stick" round Tubes	a. low pressure drop b. prevents interleakage c. all tube sheets are accessible by removing covers d. cheaper construction than 4	a. overall coefficients lower than 4 b. expensive fabrication c. large overall size

For tubes of 1.0 cm i.d. and 0.10 cm tube wall thickness the following information has been extracted from the calculations reported in AAEC/E.1.

	ν = 100 cm/sec	ν = 300 cm/sec	ν = 450 cm/sec	ν = 800 cm/sec
Na/SS				
Overall coefficient U	0.93	0.96	–	1.00
Temperature drop –				
on primary coolant side	80°C	6°C	–	5°C
– through tube wall	19°C	20°C	–	21°C
– on secondary coolant side	23°C	24°C	–	24°C
Na/Nb				
Overall coefficient U	1.22	1.28	–	1.35
Temperature drop				
– on primary coolant side	11°C	9°C	–	7°C
– through tube wall	9°C	10°C	–	10°C
– on secondary coolant side	30°C	31°C	–	33°C
Bi/Croloy 5-Si				
Overall coefficient U	0.66	0.79	0.83	–
Temperature drop –				
– on primary coolant side	22°C	17°C	15°C	–
– through tube wall	12°C	14°C	15°C	–
– on secondary coolant side	16°C	19°C	20°C	–

All the evidence shows that if tubular heat exchangers are to be used, improvement in performance can be best achieved by concentrating on the secondary coolant side.

The possible designs of hewt exchangers for improving the overall heat transfer coefficient and in turn the relative holdup are –

(i) Parallel plate type

(ii) Annular type

(iii) Primary coolant flowing down a sheet in a thin film and transferring heat to gas such as nitrogen by convection and radiation

(iv) Extended surfaces on secondary coolant side of heat exchanger

For design (i), Seban (5) gives, for parallel plates with heat through one side only, the following approximate solution in the case of uniform heat flux.

$$\frac{hD_{eq}}{k} = 5.8 + 0.02\left(D_{eq} \cdot \frac{\nu c \rho}{k}\right)^{0.8}$$

Where D_{eq} = equivalent diameter

= 2b, for wide parallel plates a distance b apart.

For design (ii), Werner, King and Tidball (6) and Lyon (7) give the expression

$$\frac{h_{ann}\, D_{eq}}{k} = 0.75\ \frac{h_{tube}\, D_{eq}}{k} \cdot R_o^{0.30}$$

Where h_{ann} = Heat transfer coefficient for annulus

D_{eq} = $D_o - D_i$ where D_o = outside diameter

and D_i = inside diameter

R_o = D_o/D_i

and h tube is calculated from $\frac{hD}{k} = 7.0 + 0.025\left(\frac{D\nu c\rho}{k}\right)^{0.8}$

For design (iii), it appears that the normal heat transfer correlations for gases flowing past hot surfaces will enable sufficiently accurate calculations to be made.

Design (iv) at first sight does not appear to have much merit, in that one will be displacing liquid metal by tube material of equal or even lower thermal conductivity.

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