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EFFECT OF SLURRY PHYSICAL PROPERTIES ON HEAT EXCHANGER
AND PUMP CHARACTERISTICS

I. SUMMARY

Design calculations were made for a system consisting of a pump, one hundred feet of pipe and a heat exchanger to remove 1 Mw of heat from various aqueous thorium oxide slurries. The rheological properties of the slurries were varied over a range of yield stresses from 0 to 1.5 lb/sq ft and of coefficients of rigidity from 1/2 to 2 centipoise. Two different cases were studied.

1. A heat exchanger having fixed axial and radial ΔT in which the tube length was allowed to vary.
2. A heat exchanger having fixed tube length in which the axial and radial ΔT were allowed to vary.

It was shown that the pump power must be increased by a factor of 15 to 30 in order to maintain satisfactory operation of the heat exchanger as the slurry yield stress is increased from 0 to 1.5 lb/sq ft. However the pump power is essentially independent of heat exchanger tube diameter for any given slurry. The rated capacity of a slurry heat exchanger is essentially independent of slurry yield stress and coefficient of rigidity, provided that the tube velocity can be suitably increased as the slurry yield stress is increased.

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II INTRODUCTION

The use of thorium oxide slurries in homogeneous reactor systems adds several new variables (e.g., yield stress and coefficient of rigidity) that must be considered in the design of different components. In some cases these variables may be the controlling factors in the design of a particular piece of equipment. In order to define possible limitations, it was suggested^a that the pump head and pump capacity be determined for a loop consisting of one-hundred feet of pipe, a pump and a heat exchanger capable of removing 40 Mw of heat from a flocculated thorium oxide slurry containing 1000 g Th/kg H₂O. The most complicated design calculations for such a system are those involved in the specification of the heat exchanger.

Design studies of a heat exchanger for use in homogeneous solution reactor systems, such as the HRT,^{1,2} are usually made by specifying:

1. Inlet temperature to heat exchanger,
2. Outlet temperature from heat exchanger,
3. Total heat removed by heat exchanger,
4. Heat exchanger tube diameter,
5. Volumetric flow rate,
6. Steam pressure,

and calculating the total heat transfer surface area as a function of tube velocity. The tube length and number of tubes followed immediately from the area-velocity relationship. Typical parameters³ for homogeneous reactor heat exchangers are given in Table I.

^a R. B. Briggs, personal communication, October 25, 1956.

TABLE 1

TYPICAL PARAMETERS FOR HOMOGENEOUS REACTOR HEAT EXCHANGERS³

Reactor	Fuel Inlet Temp., °C	(°F)	Temp. Drop Through Exchanger, °F	Tube Velocity, ft/sec	Total Flow, gpm	Power Removed in Exchanger, kw	Heat Exchanger Flux, kw/sq ft
HRT	300	(572)	77.5	11	400	5,000	10.6
TBR (core)	300	(572)	90	17.8	26,000	357,000	19.3
TBR (blanket)	315	(599)	126	11.8	4,400	82,000	18.5
HRR	275	(527)	90	15	35,300	500,000	10.0

However in the design of a heat exchanger to handle flocculated^a thorium oxide slurries, it is believed that the flow in the tubes must be turbulent at all times. Usually this means that the velocity is above that considered to give the optimum heat exchanger design, therefore velocity is not one of the independent variables in a slurry heat exchanger design study.

The systems which were studied contained a heat exchanger, 100 ft of pipe and an additional resistance equivalent to 300 L/D for elbows, expansions and contractions. Two different heat exchanger systems were studied.

1. A heat exchanger having fixed axial and radial ΔT in which the tube length was allowed to vary.
2. A heat exchanger having fixed tube length in which the axial and radial ΔT were allowed to vary, i.e., what would happen to a particular heat exchanger if the slurry properties vary during the test?

The heat exchanger tubes were 1-3/4-in., 10 Ga; 1-in., 14 Ga and 1/2-in., 16 Ga. The case 1 axial and radial ΔT were 40°F and 72°F respectively (572°F inlet, 532°F outlet and 500°F steam temperature). The case 2 tube length was 40 ft. The range of slurry yield stresses was from 0 to 1.5 lb/sq ft and coefficients of rigidity was from 1/2 to 2 centipoise. The different combinations of slurry properties used in the calculations are given in Table 4 in the Appendix.

^a In a flocculated slurry, the particles stick together in the form of loose irregular clusters in which the original particles can still be recognized. Such a slurry is called a non-Newtonian material and may be defined by its yield stress and coefficient of rigidity. In a deflocculated slurry, the particles repel each other and the original particles are not in contact. Such a slurry is called a Newtonian material and may be defined by viscosity alone.

III METHOD OF CALCULATION

As pointed out in the Introduction, the critical velocity for the onset of turbulence is controlling in the calculation of heat exchangers for use with flocculated ThO_2 slurries. The critical velocity was determined by first calculating the Hedstrom number for the chosen slurry-system combination. The critical Reynolds number (and hence the critical velocity) was determined by the intersection of the Hedstrom line with the appropriate turbulent flow friction-factor curve. Tube velocity was then arbitrarily taken as approximately twice the critical velocity. The slurry film coefficient was calculated using a version of the Dittus-Boelter equation modified for transition effects in slurries.⁵ The overall heat transfer area for the case 1 heat exchangers was calculated in a manner similar to that described by J. A. Lane in "Principles of Nuclear Reactor Engineering"¹³ (see Ref. 1 and 2 also). The case 2 heat exchangers were calculated using the same slurry side coefficients obtained in the case 1 study but using a constant steam side coefficient of $2000 \text{ BTU}/(\text{hr})(\text{sq ft})(^\circ\text{F})$. The steam side heat-transfer-coefficient temperature-drop relationship used in both cases was evaluated from the data of Cichelli and Bonilla¹⁴ (see Appendix). The pump head, pump capacity and hydraulic horsepower were calculated for each particular heat exchanger and for each heat exchanger plus an external system consisting of 100 ft of pipe and an additional resistance equivalent to $300\text{-L}/\text{D}$ for elbows, expansions and contractions, etc.

IV. RESULTS

The results of the calculations, given in Figures 1 to 10 and Tables 2 and 3 may be summarized as follows for the particular systems studied:

Case 1. Constant Axial and Radial Temperature Drop

1. The required velocity in the heat exchanger tubes increases from 10 to 30 ft/sec as the slurry yield stress increases from 0 to 1.5 lb/sq ft.
2. The pump power required:
 - A. increases by a factor of 20 to 30^a as the yield stress is increased from 0 to 1.5 lb/sq ft.
 - B. is essentially independent of tube diameter.
 - C. increases by a factor of 1.3 to 1.7 as the coefficient of rigidity is increased from 1/2 to centipoise.
3. The pump head required for heat exchanger alone:
 - A. increases by a factor of 20 to 30 as the yield stress is increased from 0 to 1.5 lb/sq ft.
 - B. is essentially independent of tube diameter.
 - C. increases by a factor of 1.3 to 1.7 as the coefficient of rigidity is increased from 1/2 to 2 centipoise.
4. The number of heat exchanger tubes:
 - A. decreases by a factor of 3 as the yield stress increases from 0 to 1.5 lb/sq ft.

^a This factor depends on the square of the ratio of tube velocity to critical Hedstrom velocity; a value of two was arbitrarily chosen for this study. If the value chosen were to be doubled (i.e., tube velocity four times the critical velocity) then the factor given above would be four times as great.

- B. is independent of coefficient of rigidity.
- 5. The heat exchanger tube length:
 - A. increases by a factor of 3 to 5 as the yield stress increases from 0 to 1.5 lb/sq ft.
 - B. decreases as the tube diameter is increased.
 - C. increases by a factor of 1.3 to 1.7 as the coefficient of rigidity increases from 1/2 to 2 centipoise.

Case 2. Constant Tube Length, Variable Axial and Radial Temperature Drop

- 1. The required velocity in the heat exchanger tubes increases from 10 to 30 ft/sec as the slurry yield stress increases from 0 to 1.5 lb/sq ft.
- 2. The pump power required:
 - A. increases by a factor of 15 to 20 as the slurry yield stress increases from 0 to 1.5 lb/sq ft.
 - B. essentially independent of tube diameter.
 - C. essentially independent of slurry coefficient of rigidity.
- 3. The pump capacity required:
 - A. increases by a factor of 3 as the slurry yield stress increases from 0 to 1.5 lb/sq ft.
 - B. increases with increasing tube diameter.
 - C. essentially independent of slurry coefficient of rigidity.
- 4. The total pump head required:
 - A. increases by a factor of 5 to 7 as the yield stress increases from 0 to 1.5 lb sq ft.
 - B. decreases with increasing tube diameter.

- C. is essentially independent of coefficient of rigidity.
5. The number of tubes and total area of the heat exchanger were constant for each tube diameter, essentially independent of slurry properties. The results are summarized in Table 1.
6. The axial ΔT through the heat exchanger:
- A. decreases by a factor of 3 as the slurry yield stress increases from 0 to 1.5 lb/sq ft.
 - B. decreases with increasing tube diameter.
 - C. is essentially independent of slurry coefficient of rigidity.
7. The radial ΔT across the heat exchanger:
- A. increases by a factor of 1.1 to 1.5 as the slurry yield stress increases from 0 to 1.5 lb/sq ft.
 - B. increases with increasing tube diameter.
 - C. increases by a factor of 1.1 to 1.5 as the slurry coefficient of rigidity increases from 1/2 to 2 centipoise.
8. The product of tube diameter, tube velocity and axial ΔT is a constant for all yield stress and coefficient of rigidity.
9. The product of tube length, overall heat transfer coefficient and radial ΔT is a constant for all yield stress and coefficient of rigidity.

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TABLE 2

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HEAT EXCHANGER DESIGN CALCULATIONS FOR AQUEOUS THORIUM OXIDE SLURRIES
FOR CONSTANT TUBE LENGTH AND VARIABLE AXIAL AND RADIAL TEMPERATURE DROP

Case Number	Yield Stress, τ_y , lb/sq ft	Coefficient of Rigidity, Centipoise	Tube Diameter, Inches	Tube Velocity, ft/sec	Overall Heat Transfer Coefficient, BTU/(hr)(sq ft)(°F)	Axial ΔT , °F	Radial ΔT , °F	Weight Rate of Flow per Mw, (lb/hr) $\times 10^{-5}$	Pressure Drop, ft of Fluid	gpm/Mw
IE	0	1/2	1/2	10	940	86	32	8.0	43	100
IF	0	2	1/2	10	750	78	36	8.7	47	110
IIE	1/2	1/2	1/2	20	980	44	31	16	108	200
IIF	1/2	2	1/2	20	660	39	41	18	102	220
IIIE	1-1/2	1/2	1/2	30	830	26	33	26	226	320
IIIF	1-1/2	2	1/2	30	600	26	45	26	219	320
IA	0	1/2	1	10	900	30	27	2.3	14	285
IC	0	2	1	10	690	30	39	2.3	18	285
IIA	1/2	1/2	1	20	590	15	35	4.5	43	560
IIC	1/2	2	1	20	520	15	46	4.5	42	560
IIIA	1-1/2	1/2	1	30	650	11	41	6.3	92	780
IIIC	1-1/2	2	1	30	480	10	50	6.9	88	860
IB	0	1/2	1-3/4	10	625	14	40	4.9	7	610
ID	0	2	1-3/4	10	490	14	48	4.9	9	610
IIB	1/2	1/2	1-3/4	20	500	7	45	10	22	1250
IID	1/2	2	1-3/4	20	380	7	61	10	21	1250
IIIB	1-1/2	1/2	1-3/4	30	450	5	53	15	45	1870
IIID	1-1/2	2	1-3/4	30	340	5	69	15	45	1870

TABLE 3

SPECIFICATIONS FOR HEAT EXCHANGER TO REMOVE 1 MW OF HEAT
FROM THORIUM OXIDE SLURRIES^a

Tube Diameter, Inches	Tube Length, Ft.	Number of Tubes	Total Surface Area	Heat Exchanger Flux, kw/sq ft.
1/2 - 16 Ga	40	31	160	6.3
1 - 14 Ga	40	17	180	5.6
1-3/4 - 10 Ga	40	11	200	5.0

^a These heat exchanger specifications are independent of slurry properties for any given tube diameter. Only the tube velocity and total rate of flow vary as the slurry properties vary. See Fig. 9.

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Fig. 1. Effect of Slurry Yield Stress on
Number of Heat Exchanger Tubes

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Fig. 2. Effect of Slurry Yield Stress on Heat Exchanger Tube Length

Fig. 3. Effect of Slurry Properties on Pump Head for Heat Exchanger (No External Piping)

Fig. 4. Effect of Slurry Properties on Pump Hydraulic Horsepower per MW for Heat Exchanger (No External Piping)

Fig. 5. Effect of Slurry Properties on Radial ΔT /Axial ΔT for 40 Feet Tube Length Heat Exchangers

Fig. 6. Effect of Slurry Properties on Axial Temperature Drop for 40 Ft. Tube Length Heat Exchanger (Variable Axial and Radial ΔT .)

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Fig. 7. Effect of Slurry Properties on Radial Temperature Drop for 40 ft. Tube Length Heat Exchanger (Variable Axial and Radial ΔT)

Fig. 8. Effect of Slurry Properties on Pump Head for 40ft. Tube Length. Heat Exchanger and for Total System (Variable Axial and Radial ΔT)

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Fig. 9. Effect of Slurry Properties on Pump Flow Rate for 40 Ft. Tube Length Heat Exchanger (Variable Axial and Radial ΔT)

Fig.10. Effect of Slurry Properties on Pump Hydraulic HP per MW for 40 Ft. Tube Length Heat Exchangers and for Total System (Variable Axial and Radial ΔT)

V. DISCUSSION OF RESULTS

For a yield stress of 1.5 lb/sq ft the electrical pump horsepower corresponds to about 5% of the heat power produced by the blanket for the heat exchanger alone and from 8 to 15% for the total system, depending on the tube diameter (and consequently on the axial temperature drop). If, instead of two, a critical velocity four times the Hedstrom crition is used, then the pump horsepower for the heat exchanger alone is increased to 20% of the heat power produced by the blanket, and from 30 to 60% for the total system, depending on the tube diameter. However this situation would be alleviated somewhat by the increase of the slurry heat transfer coefficient to essentially that of a Newtonian fluid of the same viscosity and density. Reduction of the yield stress from 1.5 to less than 0.1 lb/sq ft gives better than a 20 fold reduction in the pump power requirements.

The design curves presented in Figures 8, 9 and 10 may be used directly to determine the characteristics of systems containing heat exchangers having tube lengths different from the 40 ft assumed for these calculations. The particular relations required are:

$$\Delta P = K_1 L \quad (1)$$

$$\Delta T_{\text{Axial}} = K_2 L \quad (2)$$

$$\frac{\Delta T_{\text{Radial}}}{\Delta T_{\text{Axial}}} = \frac{Wc_p}{UA} \quad (3)$$

$$W = K_3/L \quad (4)$$

$$WAP = K_4 \quad (\text{From Fig. 3}) \quad (5).$$

The values for different tube lengths are determined by taking the ratio of the new tube length to the old tube length according to the relations given by Equations 1 to 5.

The results of Tables 2 and 3 indicate that, for any given tube diameter, the physical dimensions of the heat exchanger need not vary to maintain the same power output as the slurry yield stress and coefficient of rigidity vary. Only the tube velocity and correspondingly the pump power must be varied as the slurry properties vary. Comparison of the value for the heat transfer per unit of surface given in Table 3 with the values for heat exchangers in solution reactors given in Table 1 show that the slurry heat exchangers have values only about 30% less than the HRT and HRR, but are ~70% less than the TBR. The radial temperature drops in the slurry heat exchanger are less than those in solution heat exchangers, therefore the steam temperature and pressure can be significantly higher in the slurry system.

It should be pointed out that pumps are being developed^{11,12} that correspond closely to at least one of the extreme situations illustrated in figures 8 and 9. These pumps have a capacity of the order of 15,000 to 17,000 gpm and develop a head of 360 ft of fluid. The head required to handle a slurry having a yield stress of 1-1/2 lb/sq ft in a system containing a heat exchanger with 1/2-in., 40 ft long tubes is 230 ft of fluid and the capacity for 40 Mw of heat is 13,000 gpm. Since the product of pump head and capacity is a constant for systems having different heat exchanger tube

diameters and constant tube length (the slurry yield stress being constant), apparently any tube diameter could be accommodated by suitable alterations of this particular pump.

One additional variable that should be kept in mind as an "ace in the hole" is the slurry concentration. It has been shown that the slurry yield stress varies as the cube of the concentration.⁸ Thus if a particular slurry proves to be "unpumpable" with existing pumps, reduction of the slurry concentration should quickly reduce the yield stress to a tolerable value.

VI. CONCLUSIONS

1. For a slurry having a yield stress of 1.5 lb/sq ft the electrical pump horsepower required for the total slurry system may be 30 to 60% of the total electrical energy generated by the blanket circuit. Reduction of the yield stress from 1.5 to less than 0.1 lb/sq ft gives better than a 20 fold reduction in the pump power requirements.
2. The results of the case 2 calculation show that, if the heat exchanger tube velocity is maintained at twice that calculated from the critical Hedstrom number as the slurry properties are changed, the heat exchanger is not the controlling factor in slurry reactor design. Instead the pump power must be increased by a factor of 15 to 30 as the yield stress increases from 0 to 1.5 lb/sq ft.
3. The product of pump head and pump capacity is essentially constant for heat exchangers having different tube diameters and constant tube length, the slurry yield stress being constant.
4. The first and second conclusions requires that a high head pump be available to attain the velocities required for highly flocculated slurries,

with pumps developing of the order of 500 ft of head and having a capacity of 300 gpm per Mw being required for a slurry having a yield stress of 1.5 lb/sq ft when using 1/2-in. heat exchanger tubes; alternatively pumps developing 150 ft of head and 1800 gpm per Mw being required for a slurry having a yield stress of 1.5 lb/sq ft when using 1-3/4-in. heat exchanger tubes.

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APPENDIX A

STEAM SIDE HEAT TRANSFER COEFFICIENTS

The data of Cichelli and Bonilla¹⁴ were used to evaluate the steam side nucleate boiling heat transfer coefficients, the curves being fitted by the relation

$$q/A = K_1 (\Delta T)^3 \quad (1)$$

which has a semi-theoretical basis according to Rohsenow.¹⁷ The natural convection coefficients were extrapolated from the Cichelli and Bonilla data using a relation

$$q/A = K_2 \Delta T^{1.31}, \quad (2)$$

the slope being taken from the data of Rinaldo.¹⁶ The smoothed curves of heat transfer coefficient as a function of film temperature drop for pressures from 14.7 to 1015 psia are shown in Fig. 11.

Fig. 41. Effect of Film Temperature Drop on Steam Side Heat Transfer Coefficient

APPENDIX B

CALCULATIONS

An outline of the calculation procedure was given in the body of the report. The particular relations used in the calculations are given below.

Slurries having six different combinations of yield stress and coefficient of rigidity were chosen for these calculations. The slurry physical properties are summarized in Table 4.

TABLE 4

SLURRY PHYSICAL PROPERTIES USED IN DESIGN CALCULATIONS

Specific Heat, (BTU/lb) 275-300°C	0.5					
Density (lb/cu ft) 275-300°C	100					
Thermal Conductivity (BTU/(hr ft °F) 275-300°C	0.45					
Yield Stress (lb/sq ft)	0	0	0.5	0.5	1.5	1.5
Coefficient of Rigidity (Centipoise)	0.5	2	0.5	2	0.5	2

The critical velocity for the onset of turbulence was determined by the intersection of the two parametric lines N_{He} and η/μ on a modified Hedstrom plot (see references 7 and 8). The operating velocity in heat exchanger tubes and external piping was arbitrarily taken as about twice the critical velocity. The pressure drop through the tubes was calculated from the usual Fanning friction factor relation using the friction factor from the Hedstrom plot.

The slurry heat transfer coefficient was determined using the relation⁵

$$\frac{h_{sl} D}{k} = 0.023 \left(\frac{DV\rho}{\eta} \right)^{0.8} \left(\frac{c_p \eta}{k} \right)^{1/3} \left(1 - \frac{\tau_y}{\tau_w} \right)^3 \quad (1)$$

The case 1 tube area was determined from the relation:

$$A = Wc_p \left[\frac{1}{\frac{D_i}{D_o} h_i} \ln \left(\frac{\left(\frac{Q}{A_o} \right)_i}{\left(\frac{Q}{A_o} \right)_o} \right) + \frac{1}{\frac{D_m}{D_o} h_w} \ln \left(\frac{\left(\frac{Q}{A_o} \right)_i}{\left(\frac{Q}{A_o} \right)_o} \right) - 0.0161 \left[\left(\frac{Q}{A_o} \right)_i^{0.237} - \left(\frac{Q}{A_o} \right)_o^{0.237} \right] \right], \quad (2)$$

with the heat fluxes at the inlet and outlet being evaluated from the relation:

$$T - T_B = \frac{\frac{Q}{A_o}}{\frac{D_i}{D_o} h_i} + \frac{\frac{Q}{A_o}}{\frac{D_m}{D_o} h_w} + \frac{\frac{Q}{A_o}}{10^3 \Delta T^{0.31}} \quad (3)$$

Equation 2 and 3 are similar to those described by Lane¹³ however, the steam side relations were evaluated from the data of Appendix A. The case 2 overall heat transfer coefficient was determined in the usual way using a steam side coefficient of 2000.

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A	Area	sq/ft
c_p	Specific heat	BTU/lb
D	Diameter	ft
D_i	Inside diameter of pipe	ft
D_m	Mean of inside and outside pipe diameters	ft
D_o	Outside diameter of pipe	ft
f	Fanning friction factor	Dimensionless
g_c	Conversion factor	$\left(\frac{\text{lb force}}{\text{lb mass}}\right) \left(\frac{\text{ft}}{(\text{sec})(\text{sec})}\right)$
h_{sl}	Slurry heat transfer coefficient	BTU/(hr)(sq ft)(°F)
h_{st}	Steam heat transfer coefficient	BTU/(hr)(sq ft)(°F)
k	Slurry thermal conductivity	BTU/(hr)(ft)(°F)
k_m	Metal thermal conductivity	BTU/(hr)(ft)(°F)
L	Tube length	ft
N_{He}	Hedstrom number	dimensionless
N_{Re}	Reynolds number	dimensionless
Q	Total heat transferred	BTU/M
ΔT	Temperature differential	°F
T	Steam temperature	°F
T_B	Slurry bulk mean temperature	°F
U	Overall heat transfer coefficient	BTU/(hr)(sq ft)(°F)
V	Velocity	ft/sec
W	Weight rate of flow	lb/hr

η	Coefficient of rigidity	lb/(ft)(sec)
μ	Viscosity	lb/(ft)(sec)
ρ	Density	lb/cu ft
τ_w	Shear at wall	lb/sq ft
τ_y	Yield stress	lb/sq ft