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September 6, 1967

To: Distribution MSR-67-72

Subject: Heat Removal From Fuel Dump Tanks

A study has been made of the problem of removing heat from the fuel dump tanks, and this report summarizes the results of this study.

The heat exchanger concept investigated consists of a cylindrical tank containing double wall bayonet type assemblies. The fuel salt would occupy the space outside the bayonets. A MSRE type coolant salt would occupy the space between the two walls and the coolant material would flow on the inside of the assembly. Table I gives the dimensions of the tube assemblies.

Table I

Bayonet Assembly Dimensions

	ID (in.)	OD (in.)
Outer wall	1.930	2.000
Inner wall	1.777	1.875
Coolant supply tube	1.450	1.500

The exchanger size was determined for the criteria shown in Table II.

Table II

Design Criteria

Coolant	= steam
Inlet coolant temperature	= 650°F
Outlet coolant temperature	= 1000°F
Inlet coolant pressure	= 540 psi
Fuel salt temperature	= 1150°F
Heat removal rate	= 24 MW
Number of exchangers	= 2

The heat transfer coefficient for the fuel salt film was based on natural convection and the following equation, given by Perry, et al.,[1] was used to calculate this coefficient.

$$\frac{hL}{k} = 0.13 \left[\frac{L^3\rho^2 gB(\Delta t)}{\mu^2} (C_p\mu/k)\right]^{1/3} ,$$

where

h = heat transfer coefficient, Btu/(hr)(ft^2)(^{o}F),

L = length of heat transfer surface, ft,

k = thermal conductivity, Btu/(hr)(ft)(^{o}F),

ρ = density, lb/ft^3,

g = acceleration due to gravity, ft/hr^2,

B = volumetric coefficient of thermal expansion, (^{o}F)$^{-1}$,

Δt = temperature difference between surface and fluid, ^{o}F,

μ = viscosity, lb/(ft)(hr),

C_p = specific heat, Btu/(lb)(^{o}F).

It was assumed that the coolant salt in the space between the two walls was stagnant and that all heat will be transferred through this region by conduction. The heat transfer coefficient for the coolant film was determined from the following equation.[2]

$$\frac{hD}{k} = 0.023(DG/\mu)^{0.8} (C_p\mu/k)^{1/3},$$

where

D = equivalent diameter of the coolant passage, ft,

G = mass velocity of coolant, lb/(ft^2)(hr)

h, k, μ, and C_p are the same as previously defined.

The individual coefficients and the overall coefficient are listed in Table III.

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[1] R. H. Perry, C. H. Chilton, S. D. Kirkpatrick, "Chemical Engineers' Handbook," 4th ed., McGraw-Hill Book Co., 1963, pp 10-10.

[2] R. H. Perry, C. H. Chilton, S. D. Kirkpatrick, "Chemical Engineers' Handbook," 4th ed., McGraw-Hill Book Co., 1963, pp 10-14.

Table III

Heat Transfer Coefficients Based on Tube Assembly Outside Area

Fuel salt film	= 130 Btu/(hr)(^{o}F)(ft^2)
Outer wall	= 4000 Btu/(hr)(^{o}F)(ft^2)
Stagnant salt	= 184 Btu/(hr)(^{o}F)(ft^2)
Inner wall	= 2180 Btu/(hr)(^{o}F)(ft^2)
Steam film	= 189 Btu/(hr)(^{o}F)(ft^2)
Overall	= 52.3 Btu/(hr)(^{o}F)(ft^2)

The resulting heat exchanger requirements are summarized in Table IV.

Table IV

Summary of Heat Exchanger Design Calculations

Number of tube assemblies	= 271
Active length	= 19 ft
Tube assembly arrangement	= Triangular spacing
Tube assembly pitch	= 2 5/8 in.
Exchanger shell inside diam.	= 48 in.
Steam flow rate (each exchanger)	= 211,000 lb/hr
Steam pressure drop	= 7 psi
Steam outlet velocity	= 72 ft/sec

A transient heat transfer case was analyzed to determine the maximum thermal stress in the tube assembly. The case considered was a startup condition when the exchanger is originally at 1000^{o}F and steam at 650^{o}F is introduced at full flow. The resulting maximum temperature drop across the inside tube wall was 19^{o}F. This value was subsequently used in the stress analysis calculations.

A complete stress analysis of this vessel has not been made. However, a brief analysis was made to determine approximate metal thickness required and to establish that a feasible mechanical design can be accomplished.

The worst stress condition for the tube assembly will exist at the inside surface of the inner tube. Table V gives the calculated stresses for this location.

Table V

Stresses at Inside Surface of Inner Tube in Bayonet Assembly

Hoop stress due to pressure	= 10,400 psi
Longitudinal stress due to pressure	= 5200 psi
Radial stress due to pressure	= -540 psi
Maximum thermal stress	= 2760 psi

The maximum primary stress intensity is 10,400 psi and is less than the allowable stress intensity of 17,000 psi. The maximum primary plus secondary stress intensity is 13,700 psi and this is also less than the allowable stress intensity.

The external tube in the bayonet assembly will be subjected to an external pressure. A critical buckling pressure of 77 psi was calculated. The maximum external pressure should be below this critical value.

Estimates were made for tube sheet and vessel wall thicknesses and these values are given in Table VI.

Table VI

Estimate of Tube Sheet and Vessel Wall Thickness

	Thickness
Tank wall exposed to steam	1 1/2 in.
Tank wall exposed to salt	1/2 in.
Top dished head	1 1/2 in.
Bottom reversed dish head	1 in.
Top tube sheet (flat)	2 in.
Middle tube sheet (dished)	2 3/4 in.
Bottom tube sheet (dished)	7/8 in.

The values for the tube sheet thicknesses were determined from formulas for solid components modified by the use of ligament efficiencies.

The use of air as a coolant instead of steam was briefly investigated. A computer program called "Dump" was written to perform the iterative calculations for this study. The results of this investigation indicate that air may be used as the coolant material if the exit air pressure is approximately 100 psi.

Results are given in Table VII for one case where air was used as the coolant with the exchanger geometry the same as in the steam coolant case.

Table VII

Calculated Performance of Heat Exchanger with Air as Coolant

Air mass velocity	= 117,000 lb/(hr)(ft^2)
Air exit velocity	= 195 ft/sec
Air inlet temperature	= 100^{o}F
Air exit temperature	≈ 1150^{o}F
Air inlet pressure	= 110 psi
Air pressure drop	= 10 psi
Heat transferred	= 12 MW
Overall heat transfer coefficient	= 39.6 Btu/(hr)(ft^2)(^{o}F)
Maximum temperature drop across inner tube wall	= 20^{o}F

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