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Boiling Heat Transfer in a Bench-Scale, Molten-Salt Thermal Energy Storage Device

R. M. Canon
J. D. Hewitt

MASTER

OAK RIDGE NATIONAL LABORATORY

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CHEMICAL TECHNOLOGY DIVISION

BOILING HEAT TRANSFER IN A BENCH-SCALE,
MOLTEN-SALT THERMAL ENERGY STORAGE DEVICE

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MOLTEN-SALT THERMAL ENERGY STORAGE DEVICE

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ABSTRACT

Overall boiling heat transfer coefficients were determined experimentally for a proposed Thermal Energy Storage (TES) salt (NaNO_3) in a vertical-tube boiler for the following range of variables:

Water flow (lb/hr-ft^2)	840-2800
Steam temperature ($^{\circ}\text{F}$)	295-475
Steam pressure (psi)	50-500
Heat flux (Btu/hr-ft^2)	2100-6700

Tube and vessel heat fluxes and energy recovery fractions were also determined.

1. INTRODUCTION

Thermal Energy Storage (TES) systems have been suggested for a variety of commercial, industrial, and utilities applications. Batch operations involving heat and cool cycles (e.g., brickmaking and glassmaking) are the most obvious industrial applications; however, thermal storage would also be required for solar heating and probably for solar power plants. Systems utilizing the latent heat of fusion are attractive because of the high storage densities attainable. For most applications, an inorganic salt or salt mixture can be found with melting temperatures in the desired range.

Low frozen-salt thermal conductivities, necessitating large heat-transfer surface areas, are one disadvantage of TES systems. Since many possible applications would use a TES system for production of steam,

boiling heat transfer from the freezing salt is of fundamental importance. It is desirable to know the boiling heat transfer coefficients obtainable with a given system to provide a design basis. This report describes heat transfer experiments with a salt (NaNO_3) on the shell side of a vertical tube boiler. Data were obtained to enable calculation of heat fluxes, energy recovery potential, and heat transfer coefficients to the boiling water. Sodium nitrate is a good candidate for TES systems because of its high heat of fusion and low cost; its physical properties are summarized in Appendix A.

2. DESCRIPTION OF THE APPARATUS

The apparatus for studying the boiling heat transfer of water from a freezing salt is shown schematically in Fig. 1. Distilled water at 25°C enters the bottom of the energy storage vessel and is vaporized on the tube side. The resulting steam is condensed, and the liquid is collected to determine the flow rate. Sodium nitrate is contained in a 4-in. pipe, approximately 40 in. long. Boiling occurs in seven vertical tubes placed inside the vessel on a 1-3/8-in. triangular pitch, as shown in Fig. 2. (Further details of the boiler construction are shown in Appendix B.) Salt temperatures are measured by Chromel-Alumel thermocouples at 12 locations along the vessel length. The salt is melted by the Calrod electrical heaters wrapped around the vessel. Power is fed from a control system with temperature sensor in the vessel wall. The vessel is insulated with 2 in. of high-temperature (Fiberfrax) insulation.

Distilled water is fed to the bottom of the boiler by a positive displacement pump capable of a 10-lb/hr delivery at 1000 psi. Water temperatures

Fig. 1. Schematic diagram of the thermal energy storage system.

Fig. 2. Vessel cross section showing the boiler tube arrangement and salt thermocouple location.

are measured inside the tubes by seven 0.040-in. stainless steel, sheathed thermocouples extending from the bottom of the vessel, as shown in Fig. 3. The temperature of the water is measured at a point 2 in. upstream from the boiler, and the remaining six thermocouples are located at 5, 10, 15, 20, 25, and 30 in. into the boiler. Additional 0.062-in. thermocouples located 32 in. into the boiler and at the boiler exit measure the temperature. The boiler tubes are fabricated of type 304 stainless steel and have a 0.375-in. OD and a 0.035-in. wall with 30 in. in the salt bath. The salt vessel is made of type 304 stainless steel, sched 10 pipe.

Temperatures are recorded by two multipoint Brown Recorders calibrated in the range of 0 to 350°C. The steam is condensed and collected to determine water flow rates.

3. OPERATING PROCEDURES

The salt temperature is raised to the desired level by use of the external electrical heaters. The control system is set to maintain a vessel-wall temperature of 600 to 640°F, which is well above the melting point of the salt (585°F). The external heaters are turned off 10 min prior to each run. At this point, salt temperatures range from 599 to 626°F. Distilled water is pumped into the boiler with the pressure control system in operation. The control valve remains closed until the desired operating pressure is reached. Since some water flow is required to supply the system inventory, collection of condensate is not begun until the boiler operating pressure and the condensate flow are steady. Generally, operation is terminated 8 min after the temperature of the boiler exit declines to the saturation temperature. Run times range from 20 to 100 min. In Fig. 4,

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Fig. 3. Steam-side thermocouple placement.

Fig. 4. Temperature-time curves for a typical steam production run.

typical temperature distributions in the salt and boiler tubes show the removal of heat in a stratified manner. It is desirable to extract the heat in this manner since more useful heat can thus be obtained and higher steam temperatures are possible than if the salt were cooled uniformly. This is especially true when the feed water is subcooled.

4. CALCULATIONS

The average overall boiling coefficient is defined by the equation:

$$U_b = \frac{(q/A_b)}{\Delta t_{LM}}, \quad (1)$$

where

U_b = overall heat transfer coefficient for the boiling zone, Btu/hr-ft²-°F;

q = heat transfer rate to the boiling water per tube, Btu/hr;

A_b = area for boiling heat transfer (outer), ft²;

Δt_{LM} = logarithmic mean temperature difference between the freezing salt and the water or steam, °F.

Since part of the boiler length was used for preheating and superheating, it was necessary to determine the interval in which boiling occurs. The rate of movement (velocity) of the lower edge of the boiling zone (v) is determined by observing the time of its passage. As the zone passes a thermocouple, the temperature drops quickly to reflect the arrival of subcooled liquid. By observing the time of arrival at two or more known locations, the rate of advance can be calculated. The leading edge of the boiling zone is located by observing, at a known location, the decline in temperature that indicates movement from a region of superheated steam to

the saturated region. The velocity of the zone is necessary only because the two endpoints cannot be determined simultaneously. The boiling heat transfer area is then calculated as:

$$A_b = \pi D_o L, \quad (2)$$

where

D_o = outside diameter of the boiler tubes, ft;

L = length of the boiling section, ft.

The boiling heat transfer rate is determined from enthalpy data for saturated steam and the boiler rates as:

$$q = \Delta \bar{H}_v \times w, \quad (3)$$

where

$\Delta \bar{H}_v$ = evaporation enthalpy of steam at the operating pressure, Btu/lb;

w = mass flow of H_2O , lb/hr.

The temperature of the steam is constant; therefore,

$$\Delta t_{LM} = \frac{(t_2 - t_{sat}) - (t_1 - t_{sat})}{\ln \frac{t_2 - t_{sat}}{t_1 - t_{sat}}}, \quad (4)$$

where

t_2 = salt temperature at the top of the boiling zone, °F;

t_{sat} = steam saturation temperature, °F;

t_1 = salt temperature at the bottom of the boiling zone, °F.

These values are used to determine the overall heat transfer coefficient as defined by Eq. (1).

To determine the total heat transferred, including that involved in preheating and superheating, the total enthalpy change of water is used, and the heating rate is defined as:

$$q_t = \text{total tube heat flux} = (\bar{H}_s - \bar{H}_w) w/A, \quad (5)$$

where

$\bar{H}_s$ = enthalpy of exit stream, Btu/lb_m;

$\bar{H}_w$ = enthalpy of inlet H₂O, Btu/lb_m;

A = total tube heat transfer area (based on the outside diameter of the tube), 1.718 ft².

In order to determine heat recoveries, the sensible heat released from the vessel must be considered. The total heat recovered from the salt is:

$$Q_r = \text{heat recovered} = (\bar{H}_s - \bar{H}_w) w \Delta\theta - M_v C_{p_v} \Delta t_v, \quad (6)$$

where

$\Delta\theta$ = time of operation, hr;

M_v = mass of the vessel, lb_m;

C_{p_v} = specific heat of the vessel, Btu/lb_m-°F;

Δt_v = av temperature change of the vessel during the run, °F.

The total energy available as heat stored in the salt is a function of the temperature at which the heat is to be removed. If complete removal is defined as reduction of the temperature of all the salt to that of steam production, the total heat available is:

$$Q_A = \text{total available heat} = (\bar{H}_i - \bar{H}_t) m, \quad (7)$$

where

$\bar{H}_i$ = enthalpy of NaNO₃ at the initial molten state, Btu/lb_m;

$\bar{H}_t$ = enthalpy of NaNO_3 at the temperature of steam production,
Btu/lb_m;

m = total mass of NaNO_3 in the system, 32.3 lb_m.

Then, the percent heat recovered can be expressed as:

$$R = \% \text{ recovery} = 100\% \times \frac{Q_r}{Q_A} . \quad (8)$$

It is also desirable to calculate the heat transfer rate based on the area of the top of the vessel to further establish size requirements. This rate is defined as:

$$q_v = \text{heat flux for vessel top} = (\bar{H}_s - \bar{H}_w)w/A_v , \quad (9)$$

where

A_v = area of vessel top, 0.0990 ft².

5. RESULTS

Heat transfer coefficients and boiling zone lengths were determined experimentally in 26 runs at five different pressures. The lengths and associated heat transfer areas (based on tube outside diameter) are shown in Fig. 5 as a function of the steam flow per square foot of cross section. (The data are presented in Appendix C.) In general, higher pressures resulted in smaller boiling areas, as would be expected from the lower enthalpy requirement for evaporation. The tube boiling heat fluxes, calculated as the average heat flux in the boiling zone, are shown in Fig. 6. Lower heat fluxes for the high-pressure runs result from the smaller-temperature driving force available. The measured salt

Fig. 5. Boiling zone lengths and areas.

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Fig. 6. Heat flux in the boiling zone.

temperature is almost constant at 585°F, whereas steam temperatures vary from 300°F at 50 psi to 473°F at 500 psi.

The overall boiling heat transfer coefficients are shown for all runs in Figs. 7-11 as a function of the axial salt freezing rate. This rate is determined by observing the time at which freezing occurs, as indicated by thermocouples in the salt, at the 12 points along the boiler length. These coefficients reflect the total resistance to heat transfer between the salt-bath thermocouples and the steam. Higher coefficients for the high-pressure runs are a result of changing salt-side conditions rather than a steam-side effect. Lower thermal driving forces and higher steam flow rates appear to reduce the thickness of the frozen salt layer at the boiling zone, thus increasing the apparent heat transfer coefficient.

It should be noted that the "apparent" boiling heat transfer coefficient shows an increase at 50, 150, and 250 psi for axial freezing rates above 0.5 in./min. However, this increase is due to the method of temperature measurement and is not a real effect. It indicates that, with the tube spacing in this apparatus, freezing rates above 0.5 in./min do not allow complete freezing of the salt. Some liquid remains at the thermocouple location even after the trailing edge of the boiling zone has passed. Such a situation arises with any system as the heat load is increased. The results presented in Figs. 10 and 11 show that effective production of 375- and 500-psi steam is not possible with this apparatus at the flow rates attempted. Figures 7-9 demonstrate that a vertical-tube freezing-salt boiler can be expected to operate with an overall heat transfer coefficient of 20 to 30 Btu/hr-ft²-°F when designed for high recovery fraction (complete freezing of the salt).

Fig. 7. Boiling heat transfer coefficients at 50 psi.

Fig. 8. Boiling heat transfer coefficients at 150 psi.

Fig. 9. Boiling heat transfer coefficients at 250 psi.

Fig. 10. Boiling heat transfer coefficients at 375 psi.

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Fig. 11. Boiling heat transfer coefficients at 500 psi.

Figure 12 shows the heat recoveries achieved with this system. These data result from those experiments in which boiling was continued until the boiling zone reached the top of the salt vessel. More heat could have been extracted as wet steam. The heat fluxes are average values for the entire tube length, including the preheat and superheat sections. Statistical analysis of these data indicates that an infinite slope (no dependence of recovery on heat flux) cannot be rejected. Apparently, heat load has no effect on the quantity of heat recovered in the region of the data. This is true even for the high-pressure runs with subcooled feed water. Obviously, this would not be the case with saturated water feed.

6. CONCLUSIONS

A thermal energy storage system utilizing NaNO_3 or a similar salt in a passive role can achieve acceptable energy recoveries. It is likely that the number and spacing of tubes would be changed to suit specific needs. For instance, high flow rates would require fewer but longer tubes for maximum recovery. It is desirable to remove heat in a stratified manner, thus producing large gradients between the salt temperatures at the water inlet and steam outlet. Wider spacing of the boiler tubes is possible if the system is to be limited to low steam flow rates. To maintain stratified heat removal, one-pass exchangers appear to be preferred.

The salt-phase thermocouples are located approximately 0.5 in. from the tube walls in these experiments. At this distance, the salt adjacent to the boiling zone remains molten (at 585°F) for the higher flows. It is for this reason that higher coefficients and heat fluxes are obtained.

Fig. 12. Heat recovery from the NaNO_3 thermal energy storage system.

The frozen salt layer is thinner and thus offers less resistance to heat transfer. Apparently, the boiling zone moves past the thermocouple before the freezing front. Further experiments will be necessary to establish a firm understanding of natural convection and heat transfer processes in realistic natural convection systems. These experiments should be designed to measure both axial and radial temperature gradients in the salt. A method of directly determining the location of the freezing front should be used. Experiments with varying tube spacings and lengths would provide the data necessary for the design of a variety of such devices. Investigations should be directed toward the determination of maximum heat fluxes (at a prescribed recovery efficiency) as a function of the ratio of tube area to heat content of the salt. This would require a number of experiments with several boilers. However, such data are necessary to provide a sound basis for a process design.

7. ACKNOWLEDGMENTS

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APPENDIX A:

CHEMICAL ANALYSES AND PHYSICAL PROPERTIES OF NaNO_3

Table A-1. Analyses of NaNO_3 before and during the heat transfer experiments

Impurity	Concentration (wt %)		
	Total system time above mp		
	Before ^a	710 hr	1000 hr
NO_2^-	<0.0005	<0.0015	0.0053
OH^-	<0.0005	<0.0075	0.0005
Fe	<0.0002	0.0020	0.0030
Cr	<0.0002	0.0008	0.0009
Ni		<0.0002	<0.0010

^aThe NaNO_3 , which was obtained from Mallinckrodt, was analytical reagent grade and granular in form.

Table A-2. Properties of NaNO_3^a

Melting point, °F	585
ρ , lb _m /ft ³	118.7 at 585°F (liq)
$\Delta V_{\text{melting}}/V_{\text{solid}}$	0.107
ΔH_f , Btu/lb _m	74.5
C_p , Btu/lb-mol-°F	37.0 (liquid at 585°F)
k_{liq} , Btu/lb-ft-°F	0.319

^aData obtained from George J. Janz, Molten Salts Handbook, Academic, N.Y., 1967.

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APPENDIX B:

ADDITIONAL DETAILS OF THE BOILER CONSTRUCTION

Fig. B-1. Energy storage vessel/boiler configuration used in the heat transfer experiments.

Fig. B-2. Boiler tube configuration at vessel exit.

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APPENDIX C:

EXPERIMENTAL DATA

Table C-1. Experimental data for thermal energy storage study using NaNO_3

Run number	Steam pressure (psi)	Sat. temp. ($^{\circ}\text{F}$)	q/A_b (Btu/hr-ft 2)	q_t (Btu/hr-ft 2)	R (%)	U (Btu/hr-ft 2 - $^{\circ}\text{F}$)	W (lb/hr)	G (lb/hr-ft 2)	L (ft)	v (fpm x 10)
2	50	295	6350	6700	32.2	27.6	9.94	2800	2.08	4.67
3	50	295	6950	5860	89.2	25.6	8.71	2450	1.89	3.25
4	50	295	7950	6080	90.2	22.5	9.04	2550	2.23	3.75
5	50	295	3580			25.9	3.16	890	1.17	1.75
6	50	295	3450			26.6	2.97	840	1.14	1.33
7	50	295	4050			22.2	5.08	1430	1.67	2.17
12	500	475	4910	4580	82.9	44.4	6.89	1940	1.53	2.75
13	500	475	5230	5530	88.6	47.0	8.33	2350	1.72	2.33
14	500	475	3770	3360	75.9	34.1	5.06	1420	1.46	1.92
15	500	475	2450			21.9	3.01	850	1.33	1.33
16	250	408	3190	2160	78.3	28.3	3.21	900	1.20	1.58
17	250	408	3960			25.3	4.83	1380	1.48	1.92
18	250	408	5240			29.5	6.93	1970	1.58	2.75
19	250	408	5970			33.7	8.77	2470	1.75	4.17
20	150	367	3780			31.3	2.98	840	0.983	1.50
21	150	357	4670			27.9	5.19	1460	1.38	2.00
22	50	295	3520	2210	69.3	27.1	3.21	900	1.21	1.50
23	50	295	4720	4420	70.4	20.9	6.80	1910	1.91	2.42
24	50	295	5720			23.9	8.83	2490	2.05	4.00
25	150	367	5070			25.9	6.89	1940	1.69	2.58
26	150	367	6410			30.3	8.81	2480	1.71	3.92
27	375	442	6480			43.2	8.58	2420	1.51	3.50
28	375	442	4380			35.0	6.37	1790	1.48	2.58
29	375	442	3300			27.8	4.91	1380	1.48	1.75
30	50	295	4630	1530	81.9	33.4	2.38	640	0.653	1.17
31	150	367	4580	3390	82.9	26.9	5.35	1650	1.59	2.08

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