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**PERFORMANCE CHARACTERISTICS OF A SHORT REENTRY TUBE
STEAM GENERATOR AT LOW STEAM OUTPUT**

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PERFORMANCE CHARACTERISTICS OF A SHORT REENTRY TUBE STEAM GENERATOR AT LOW STEAM OUTPUT

R. S. Holcomb M. E. Lackey

ABSTRACT

The performance of a short reentry tube steam generator was investigated for steam flow rates of 1-7 lb/hr and pressures from atmospheric to 200 psia for possible application in a radioisotope power plant. The outer boiler tube had an OD of 3/4 in. and a length of 50 in. Six combinations of internal geometry were tested to determine the configuration that would yield the highest exit steam temperature at 200 psia with an outer tube wall temperature of 1000°F.

An exit steam temperature of 720°F at a flow rate of 5.2 lb/hr was achieved with Model 2, which had a 12-in.-long insulating sleeve in the lower end of the central tube. A boiler tube having this configuration should operate in a stable manner for flow rates up to 6 lb/hr if a pressure drop of about 100 psi is provided between the feed pump discharge and the boiler.

INTRODUCTION

The reentry tube boiler is a once-through boiler that employs two concentric tubes through which the fluid passes to become vaporized and superheated. The liquid enters and flows upward inside a central tube where it is vaporized and the vapor is superheated somewhat. The vapor leaves the inner tube and flows downward through the annulus between the inner and outer tube where it receives additional superheat. The heat is supplied to the outside of the outer tube and the fluid in the central tube is heated by the vapor in the annulus. A schematic drawing of the reentry tube boiler is shown in Fig. 1.

The reentry tube boiler has been proposed for use in molten-salt reactor steam power plants¹ and in radioisotope steam power plants.² This type of boiler has the advantage for molten-salt reactor plants that the steam annulus between the inner and outer tubes would act as a buffer between the high-temperature molten salt and the lower temperature feedwater and thus help alleviate such difficult problems as salt freezing, thermal

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Fig. 1. Diagram of a Reentry Tube Boiler.

shock, and boiling flow instabilities. The reentry tube boiler is particularly well suited for use in a radioisotope power plant because it provides a convenient method of coupling the power conversion system with the heat block-shield since the boiler tubes can bayonet into the heat block with all of the header connections at one end. This feature makes it possible for the heat block-shield and the power conversion system to be shipped separately to the deployment site, thus facilitating adherence to the shipping and safety regulations.

The reentry tube boiler concept was tested initially by Konishi and Koch.³ They operated an electrically heated single tube boiler using water with flow rates up to 6 lb/hr and pressures up to 214 psia. Their test section consisted of a 1/4-in.-OD inner tube inside a 3/8-in. schedule 40 pipe 68 in. long. For the plain tubes and with the inlet water subcooled, the observed exit steam temperature corresponded to the saturation temperature for each value of boiler pressure. By preheating the water to near the saturation temperature and adding a 1-ft-long insulating sleeve on the outside of the bottom of the inner tube, an exit steam temperature of 425°F at atmospheric pressure was achieved. They recommended the use of turbulence promoters on the inside surface of the outer tube to increase the heat transfer to the steam in the annulus.

The present work was performed as a part of the Isotope Kilowatt Program in connection with the possible application of the reentry tube boiler to a radioisotope steam power plant for undersea use.² The power unit has a design power output of 3 kw(e) with a heat input of about 30 kw(t) to the boiler. The design steam conditions at the turbine inlet were initially 800°F at 200 psia with a flow rate of about 75 lb/hr. The boiler would employ from 12 to 18 tubes, giving a steam flow of 4 to 6 lb/hr per tube. In order to limit the temperature of the isotope fuel can in the steel heat-block shield, the outside diameter of the boiler tubes needs to be at least 3/4 in. with the maximum allowable temperature on the outside of the boiler tube about 1000°F at the midplane of the heat block. The temperature will be somewhat lower than 1000°F at the ends of the heat block because the requirement for end shielding prevents the fuel element from extending the full length of the heat block. The length of the boiler tube was limited to about 4-5 ft corresponding to the range of length for a minimum weight shield.

The objectives of the present work were to find a design that would produce superheated steam at near the design conditions of the isotope power plant, to investigate the performance over a range of flow and pressure, and to investigate the effect of the outer tube wall axial temperature distribution on the exit steam temperature. Preliminary calculations indicated that a geometry with spiral fins on the inside surface of the outer tube that would extend to near the surface of the central tube would improve the heat transfer in the annulus. This configuration was selected on the basis that the fins would provide increased heat transfer area from the outer tube to the steam in the annulus, and that a narrow gap between the fin tips and the central tube would provide a low-resistance path for heat transfer from the outer tube to the central tube. A stainless steel tube with a 3/4 in. OD and 8 spiral fins machined on its inside surface that had been fabricated as a spare for another program was available. Since this configuration appeared to be promising for this application and because considerable expense would have been involved in tooling to produce a tube with a new geometry, this tube was selected for testing. An existing test facility was modified to meet the requirements of the reentry tube boiler test. The test conditions included water flow rates from 1 to 7 lb/hr, pressures from atmospheric to 200 psia, and outer tube wall temperatures up to 1000°F.

APPARATUS

The major components of the reentry tube boiler test facility include a pressurized water tank, the boiler, preheaters, condenser, and rotameter. A schematic diagram of the test facility is shown in Fig. 2. Distilled water flows from the water supply tank, through two valves, through the rotameter, through two preheater sections, and into the boiler. Steam discharges from the boiler, flows through a needle valve and into the condenser. Heat is supplied to the preheaters and boiler by electric heaters.

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Fig. 2. Schematic Diagram of Test Facility.

Boiler

The boiler consists of two concentric tubes surrounded by clamshell electric heaters. The inner and outer tubes are both connected to a 3/8-in.-pipe tee, which provides an outlet connection for the steam flow. The outer tube is welded to one end of the run on the tee. The inner tube passes through the tee and is attached to the other end of the run on the tee by means of a threaded compression fitting. The tee and the compression fitting were machined to enlarge the ID to allow for insertion of the inner tube. The steam outlet line is welded to the side outlet of the tee. A cross section of the boiler assembly is shown in Fig. 3. The water flows inside the inner tube where it is vaporized. The steam flows in the annulus between the inner and outer tube and discharges through the outlet line. The outer tube is a 3/4 in. OD by 50-in.-long stainless steel tube with eight spiral fins machined on the inside of the tube. The fins are 0.050 in. high by 0.050 in. wide arranged on a pitch of 0.3 in. The bore of the tube at the fin tips is 0.530 in. An end plug is welded into the top end of the outer tube.

The inner tube is a 1/2-in.-OD stainless steel tube with a wall thickness of 0.035 in. It is inserted through the tee into the outer tube and extends to a point 1/2 in. from the top end of the outer tube. Two special features of the inner tube are provided to enhance the performance of the boiler. The first of these is a section with a reduced diameter tube and a concentric sleeve located at the lower end of the boiler. The sleeve is open at one end to the steam in the annulus, which allows the space inside the sleeve to be filled with a stagnant layer of steam that provides the proper degree of thermal insulation between the inner tube and the annulus. The smaller tube has an ID of 0.245 in. and an OD of 0.375 in. The gap between the smaller tube and the sleeve is 0.0275 in. The length of the insulated section of the inner tube was varied from 0 to 24 in. in different models.

The second special feature of the inner tube is a solid plug that is inserted inside the inner tube at the top end of the boiler. The plug has a diameter of 0.375 in. and is tapered to a diameter of 1/8 in. with a rounded leading end. The purpose of the plug is to form a flow passage

Fig. 3. Boiler Assembly.

with a reduced width and consequently increase the heat transfer coefficient in the region of the inner tube where the steam is of high quality or superheated. The length of this plug was varied from 0 to 24 in. in different models of the inner tube.

The characteristics of the six models of the inner tube that were tested are given in Table 1.

Table 1. Characteristics of Reentry Tube
Boiler Test Models

Model No.	Inner Tube Plug Length (in.)	Insulating Sleeve Length (in.)
1	3	0
2	3	12
3	0	12
4	24	12
5	3	24
6	3	16

The heat is supplied to the boiler by four pairs of clamshell electric heaters. Each of the four pairs of heaters has an independently controlled power supply. The heaters have an ID of 1 1/4 in. and are 12 1/8 in. long. Lava spacers are used to provide a 1/4 in. radial gap between the boiler tube and the heater. The presence of the spacers creates a 1/2-in.-long axial gap between heaters. The boiler assembly is covered with 3 in. of ceramic fibrous blanket insulation (B&W Kaowool).

Water Supply System

The water supply system consists of a stainless steel pressure vessel and a bottle of high-pressure nitrogen gas. The vessel is filled with distilled water and then pressurized with nitrogen. The two valves in series

in the water supply line are then opened and set to provide a steady flow of water at the desired rate. When the vessel is nearly empty, it is refilled.

Condenser

A 55-gallon stainless steel drum about half full of water serves as a direct-contact condenser for the test facility. The boiler exit steam line is reduced in size to a 1/4-in.-OD stainless steel tube from the needle valve used to control the boiler pressure over to the condenser. The tube runs down inside the 55-gallon drum so that the steam is discharged at the bottom of the drum and well below the surface of the water. The condensate is used as a source of water to refill the water supply vessel.

Electrical and Instrumentation System

Electrical power is supplied to the preheaters and the four boiler heaters from six variable transformers, which have a range of output voltage of 0 to 270 volts. The voltage scale on each transformer was calibrated with a voltmeter and the scale setting was recorded for each run. The current was measured with a multirange ammeter that had full-scale ranges of 3, 7.5, and 30 amperes.

The temperatures are measured using chromel-alumel thermocouples. The thermocouples on the boiler outer tube are bare junction thermocouples, which are insulated with ceramic beads on each wire and with the junctions spotwelded to the tube outside surface. The same type is used to measure the temperatures on the inner and outer surfaces of the insulation surrounding the clamshell heaters. The entering water temperature, the inner tube outlet temperature, and the steam exit temperature are measured with chromel-alumel thermocouples enclosed in a 1/8-in.-diam stainless steel sheath. The thermocouple emf was measured and recorded on a 24-point recording potentiometer. A diagram showing the location and identification number of the thermocouples is shown in Fig. 4.

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Fig. 4. Thermocouple Location Diagram.

The water inlet pressure and the steam outlet pressure are measured with 0 to 600 psi Bourdon pressure gages. The water flow rate is measured with a rotameter that was calibrated by collecting the water in a graduated beaker for a measured time period.

PROCEDURE

The independent variables for the test conditions include flow rate, pressure, and outer tube wall temperature. The flow rate was varied from 1-7 lb/hr. The range of pressure was from atmospheric up to 200 psia, with most runs made at either atmospheric pressure or 200 psia. The outer tube wall temperature was varied from 800 up to 1000°F, with most of the tests run at a wall temperature of 1000°F.

The procedure used for each of the experiments was as follows:

1. The water supply vessel was filled with distilled water.
2. The water supply vessel was pressurized with nitrogen gas from a nitrogen bottle. A pressure of 100 psig was used for operating the boiler at atmospheric pressure, and a pressure of 300 psig was used for a boiler pressure of 200 psia.
3. The water inlet valves were opened and a water flow rate of about 2 lb/hr was started through the boiler.
4. Power was turned on to the preheaters and boiler heaters, and the boiler tube wall and insulation were allowed to heat up.
5. The flow rate was set to the value desired and the boiler outlet throttle valve was set to give the desired boiler pressure.
6. The power level to the preheaters was adjusted until the boiler inlet temperature was just slightly less than the saturation temperature at the boiler pressure. The power input to each of the boiler heaters was adjusted until the desired outer tube wall temperature was achieved in each of the four sections of the boiler.
7. When the boiler appeared to be at thermal equilibrium, as indicated by stabilization of the temperatures, the system data were recorded. The voltage and current for each heater, temperatures, pressure, and flow rate were read and recorded.

METHOD OF CALCULATION

The net electrical heat input was calculated from the measured current and voltage and compared with the heat absorbed by the steam as indicated by the measured flow and temperatures.

Net Heat Input

The gross electrical power input was calculated for each of the four boiler heaters by the relation

$$P = EI .$$

Each of the variacs was calibrated by measuring the voltage with a precision voltmeter at each 10% point on the variac scale. The current was read with a multiscale precision ammeter.

The heat losses from the boiler tube were measured for a range of temperatures with the boiler dry. The conductance of the insulation was calculated from the measured heat loss and correlated with the mean temperature and temperature difference across the insulation. The conductance values were used to calculate the heat loss for a given run from the measured temperatures at the inner and outer surfaces of the insulation.

The net electrical heat input for each of the four boiler heaters was calculated by subtracting the heat loss from the gross power input. The average heat flux was calculated by dividing the net heat input by the area of the outside surface of the outer tube per heater. The total net heat input was found by taking the sum of the heat inputs for the four heaters.

Heat Absorbed by Steam

The heat that was absorbed by the steam was calculated from the enthalpy rise per pound of steam and the steam flow rate through the boiler. The enthalpy at the boiler inlet was taken as that for liquid at the measured temperature. The enthalpy at the boiler outlet was taken as that for superheated steam at the measured outlet temperature.

The flow rate was found by taking the value corresponding to the rotameter reading from the rotameter calibration curve shown in Fig. 5.

The heat absorbed by the steam was calculated as the product of the enthalpy rise per pound of steam times the flow rate.

The ratio of the heat absorbed by the steam to the net electrical heat input was calculated for each run, and where the ratio varied from unity more than about 20%, the run was discounted. The variation was less than 10% for most of the runs.

RESULTS AND DISCUSSION

The test results are presented for the most part in terms of the steam temperature achieved at the exit of the boiler, which is designated as the superheater exit steam temperature. The steam temperature at the exit of the central tube is also given, since it is indicative of the nature of the heat transfer process between the central tube and the annulus. The power distribution at a given wall temperature is also compared for three of the six models.

Exit Steam Temperature

The results obtained on Model 1 are presented in Fig. 6. At atmospheric pressure, the central tube exit temperature is very high at a low flow rate, but decreases rapidly to the saturation temperature at a flow of about 3 lb/hr, and then remains at the saturation temperature. The resistance of the exit steam line causes a slight increase in boiler pressure with flow rate and consequently a small increase in the saturation temperature. The superheater exit steam temperature exhibits a pronounced minimum value at a flow rate of 2.75 lb/hr and has a maximum value of 480°F at about 4 lb/hr. At 200 psia, the central tube exit steam temperature decreases gradually with flow rate, tends to level off, and then decreases rapidly toward the saturation temperature. The superheater exit steam temperature gradually increases with flow rate and appears to level off at 7 lb/hr. The superheater exit steam temperatures achieved with Model 1 were considerably lower than the desired goal of 800°F. The heat

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Fig. 5. Rotameter Calibration Curve.

Fig. 6. Central Tube Exit and Superheater Exit Steam Temperature vs Flow Rate for Model 1 at Atmospheric Pressure and 200 psia.

flux that can be transferred from the heat block at the design temperature from the radioisotope system would limit the allowable flow per tube to about 5 lb/hr. The superheater exit steam temperature measured at 200 psia and 5.2 lb/hr was only 450°F.

Since the superheater exit steam temperature was lower than the central tube exit steam temperature for flows up to 6 lb/hr, it appeared that the steam in the lower portion of the superheater annulus was losing too much of its heat to the boiling water in the central tube. This led to the selection of a design utilizing a central tube fitted with a loose sleeve to create a steam gap at the lower end of the boiler and thus provide thermal insulation between the annulus and the central tube. The length of this sleeve was chosen as 12 in. for Model 2.

The steam temperatures found with Model 2 are presented in Fig. 7. The central tube exit steam temperature at atmospheric pressure dropped more quickly with an increase in flow rate, but had the same general trend as for Model 1. The superheater exit steam temperature increases rapidly with flow rate, with the rate of increase diminishing up to 6 lb/hr. No minimum similar to that for Model 1 was observed at atmospheric pressure.

The results at 200 psia showed a trend in the central tube exit steam temperature similar to that seen with Model 1, except that the temperature decreased more quickly, had a broader level region, and reached the saturation temperature at 7 lb/hr. The superheater exit steam temperature had much higher values than those for Model 1. The temperature measured at 5.2 lb/hr flow was 720°F, which would be quite acceptable for a steam Rankine cycle since it would give a cycle efficiency of only slightly less than the original goal of 800°F.

The results above were obtained for a boiler inlet temperature just below the saturation temperature corresponding to the boiler pressure. When the inlet temperature was reduced to 120°F for a boiler pressure of 200 psia and a flow rate of 5.2 lb/hr, the resulting superheater exit temperature was only 390°F, which is just slightly higher than the saturation temperature. Thus it seems to be imperative that the feedwater be preheated to near the saturation temperature in order to obtain high superheater exit temperatures.

Fig. 7. Central Tube Exit and Superheater Exit Steam Temperature vs Flow Rate for Model 2 at Atmospheric Pressure and 200 psia.

In Models 3 and 4, the length of the plug in the top of the central tube was varied to determine its effect on the performance. These results are shown in Figs. 8 and 9, respectively. With Model 3, where the plug was removed, the central tube exit steam temperature was reduced, but the superheater exit steam temperature was nearly the same as for Model 2. In the case of Model 4, where the plug was made 24 in. long, the central tube exit steam temperature was increased, but again the superheater exit steam temperature was practically unchanged from Model 2.

In an attempt to see if an even higher superheater exit steam temperature could be achieved, Model 5 was made with an insulated section 24 in. long. These results are given in Fig. 10. At atmospheric pressure, the central tube exit temperature was slightly lower than for Model 2. The superheater exit steam temperature was also lower than for Model 2. The central tube exit steam temperature at 200 psia fell more quickly to the saturation temperature than for Model 2. Again, the superheater exit steam temperature was lower than that for Model 2, i.e., the temperature was 655°F at a flow rate of 5.2 lb/hr as compared to 720°F for Model 2.

The reduction in superheater exit temperature experienced when the insulating sleeve length was increased from 12 to 24 in. confirms the analysis indicating that there is a optimum length of the insulating sleeve for a particular design flow rate. This is brought about by the fact that, as the insulating sleeve length is increased, less heat is put into the water in the central tube in the lower part of the boiler. This produces two effects:

1. It changes the role of the upper part of the central tube from a superheater to part evaporator, thereby reducing the wall temperature of the central tube.
2. It reduces the central tube exit temperature.

These two effects combine to reduce the maximum steam temperature reached in the upper part of the annulus and this in turn causes a lower superheater exit steam temperature.

The results from Models 2 and 5 indicated that an exit steam temperature of about 740°F at 200 psia and a flow rate of 5.2 lb/hr might be achieved with a sleeve length of 16 in. Model 6 was fabricated with this sleeve length and was tested. These results are shown in Fig. 11. At

Fig. 8. Central Tube Exit and Superheater Exit Steam Temperature vs Flow Rate for Model 3 at Atmospheric Pressure and 200 psia.

Fig. 9. Central Tube Exit and Superheater Exit Steam Temperature vs Flow Rate for Model 4 at Atmospheric Pressure and 200 psia.

Fig. 10. Central Tube Exit and Superheater Exit Steam Temperature vs Flow Rate for Model 5 at Atmospheric Pressure and 200 psia.

Fig. 11. Central Tube Exit and Superheater Exit Steam Temperature vs Flow Rate for Model 6 at Atmospheric Pressure and 200 psia.

atmospheric pressure, the observed exit steam temperatures were higher than those for either Models 2 or 5. However, at 200 psia, the exit steam temperatures were essentially the same as those for Model 2. Thus, there is apparently a fairly broad range of length of the insulating sleeve over which the exit steam temperature is constant. This is illustrated in Fig. 12, where the superheater exit steam temperature is shown as a function of the insulating sleeve length for 200 psia and flow rate of 5.2 lb/hr. It appears that the maximum exit temperature that can be achieved is essentially the same as that observed for the 12-in. sleeve length, which was 720°F.

The data discussed above were all taken with an outer tube wall temperature of about 1000°F over the length of the boiler. The radioisotope heat block-shield will have unfueled shield regions at the upper and lower ends, and therefore the heat block temperature might be somewhat lower at the ends than at the center. In order to determine the effect of the tube wall temperature on the superheater exit steam temperature, data were taken with reduced tube wall temperatures at the upper and lower boiler heaters. The superheater exit steam temperature is given vs the tube wall temperature at 6 in. from the steam exit in Fig. 13. The exit steam temperature is reduced by only about 25°F as the tube wall temperature is decreased from 1000 to 950°F, but below 950°F the rate of reduction is more pronounced.

Heat Distribution

The heat input requirement for each of the four boiler sections is of interest as it relates to the wall temperature distribution that would result in the radioisotope heat block-shield application. The actual conditions in the heat block would lie between that of uniform wall temperature and that of uniform heat input. Both of these cases were approximated on Model 2 at a flow rate of 5.2 lb/hr and 200 psia. The heat input and the tube wall temperature for each of the four boiler sections is shown for both cases in Table 2.

Fig. 12. Superheater Exit Steam Temperature vs Insulating Sleeve Length at 5.2 lb/hr Flow and 200 psia.

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Fig. 13. Superheater Exit Steam Temperature vs Tube Wall Temperature for Model 2 at 5.2 lb/hr Flow and 200 psia.

Table 2. Tube Wall Temperature and Heat Input
Distribution for Model 2

Case	Boiler Section	Tube Wall Temperature (°F)	Heat Input (Btu/hr)
Uniform wall temperature	1 (top)	1012	1257
	2	1040	453
	3	968	2038
	4 (bottom)	1010	872
Uniform heat input	1	905	1356
	2	908	1323
	3	821	1391
	4	990	1090

In both these cases the superheater exit steam temperature was about 720°F. These results, along with the information from the tests at lower uniform wall temperatures, indicate that the exit steam temperature is essentially independent of both the wall temperature and heat input to sections 1, 2, and 3 within wide limits, and depends only on the tube wall temperature in boiler section 4. Thus, it would appear that satisfactory exit steam temperatures can be obtained in a radioisotope heat block without creating excessive hot spots along the length of the block because of the variable heat-removal capability of the boiler tube. Where the heat input demand of the boiler tube is greater, the tube wall temperature will tend to be reduced. This will in turn reduce the heat block temperature at that point. Axial conduction in the heat block will cause heat to flow to the lower temperature, and reduce the temperature at the point where the heat-removal capability of the boiler tube is smaller.

A primary objective in testing Models 3 and 4 was to determine if the distribution of heat input to each of the four boiler sections could be improved over that observed for Model 2 for a uniform wall temperature of 1000°F. The net heat input to each of the boiler sections is shown for Models 2, 3, and 4 in Table 3. Considerable improvement is shown with Model 3 in that the heat input to section 2 is increased to 812 Btu/hr from 453 Btu/hr for Model 2. An even further increase to 944 Btu/hr was

Table 3. Boiler Tube Heat Distribution
for a 1000°F Uniform Wall Temperature

Model No.	Boiler Section			
	1	2	3	4
	Heat Input, Btu/hr			
2	1257	453	2038	872
3	1294	812	2089	903
4	1246	944	2054	861

found for Model 4. Unfortunately, these improvements were offset by poorer flow stability characteristics of Models 3 and 4 and the overall performance of Model 2 is considered to be superior to that of both Models 3 and 4.

Stability of Operation

An oscillation in the boiler pressure and flow rate was observed to some extent for all of the models tested. It was found that the magnitude of these oscillations, and particularly the variation in flow rate, could be reduced to quite acceptable levels by increasing the pressure in the water supply vessel and accepting a large pressure drop across the water inlet valves. With a pressure drop of about 100 psi across the inlet valves, the boiler pressure fluctuated about 3 psi at atmospheric pressure and about 5 psi at 200 psia for Models 2 and 6. The flow rate varied approximately $\pm 5\%$ at both pressure levels. Above a flow rate of about 6 lb/hr at 200 psia, these oscillations increased to about 10 psi and $\pm 10\%$ of the flow rate. The magnitude of the oscillations began increasing at a lower flow rate for Model 3, with a 10 psi pressure oscillation observed at a flow rate of 5.2 lb/hr.

In addition to pressure and flow oscillations similar to that of Models 2 and 6, a temperature instability was found for Models 4 and 5 at atmospheric pressure. The tube wall temperatures in boiler sections 1 and

3 were unstable with respect to heat input in that only a slight change in heat input to section 3 caused a large change in the wall temperature in section 1, and when an attempt was made to adjust the wall temperature by changing the heat input to section 1, this caused a large change in the wall temperature in section 3. No combination of heat inputs was found for which the wall temperature would attain a stable value.

Calculated Performance

An attempt was made to calculate the performance of the Model 1 re-entry tube boiler so that an analytical model might be found by which the performance of future designs might be predicted. To do this, a number of assumptions were made about the heat transfer processes in both the central tube and the annulus.

The boiling heat transfer coefficient inside the central tube was assumed to be constant up to a quality of 70%, and the decrease exponentially to the convective heat transfer coefficient for steam at 100% quality. A boiling heat transfer coefficient of 1500 Btu/hr·ft²·°F was assumed for a pressure of 200 psia. The convective heat transfer coefficient to the superheated steam was calculated from the following equation for steam Reynolds numbers in the range from 2000 to 10,000:

$$\frac{hDe}{k_f} = \frac{0.005}{(L/De)^{0.1}} Re_f Pr_f^{0.4}$$

where the steam properties are evaluated at an average value between the bulk temperature and the wall temperature.

The steam flow in the annulus was in the laminar flow range. The heat transfer coefficient in the annulus was calculated by two methods: from the resistance to heat conduction across the gap using the static gas thermal conductivity, and from the following laminar flow heat transfer equation:

$$\frac{hDe}{k} = 1.86 \left(\frac{Re Pr}{L/De} \right)^{1/3}$$

The larger of the two values was used, since the heat transfer coefficient should never be less than that for conduction across a static gas layer. The calculated value of the average heat transfer coefficient in the annulus was much too low to transfer the heat to the central tube at the steam temperatures measured. It appears that some effect is present to cause the heat transfer coefficient in the annulus to be considerably higher for the steam to the central tube than that for the outer tube to the steam. One possibility is that the flow distribution in the annulus is such that the bulk of the steam flows at a higher velocity in a narrow region surrounding the colder central tube. This assumption was made, and the heat transfer coefficient was calculated separately for the central tube and outer tube surfaces. The thickness of the gas layer used for calculating the inner coefficient was taken as 0.0075 in., which was one-half of the radial clearance between the central tube and the tip of the fins on the outer tube. The thickness of the gas layer used for the outer coefficient was about four times the inner layer, thus the heat transfer coefficient for the central tube was about four times that for the outer tube.

The heat transferred by radiation from the outer tube to the central tube was calculated from the relation:

$$q = \frac{A_o \sigma (T_o^4 - T_c^4)}{1/\epsilon_o + (1/\epsilon_c - 1) A_o/A_c}$$

where

A_c = Surface area of the central tube,

A_o = Surface area of the outer tube,

T_c = Central tube temperature, °F,

T_o = Outer tube temperature, °F,

ϵ_c = Emissivity of central tube,

ϵ_o = Emissivity of outer tube.

The surface area used for the outer tube was the area between fins and the fin tips, but neglecting the sides of the fins. This was compensated for by using a view factor of one between the two surfaces. A value of 0.8 was assumed for the emissivity of both surfaces.

The central tube wall temperature and the temperature distribution of the steam in the annulus, which was calculated for a flow rate of 5.2 lb/hr at 200 psia for Model 1, is shown in Fig. 14. To reduce the length of the calculations required, the central tube exit steam temperature, which was observed experimentally at these conditions, was assumed for the calculated performance, which leaves only the superheater exit steam temperature to be compared to the measured value. An exit temperature of 516°F was calculated and a temperature of 450°F was measured. This is not considered to be very good agreement. It is therefore recommended that the calculated performance of any new reentry boiler design be used only as an approximation, and that the boiler be tested to determine more exactly its performance.

CONCLUSIONS

The following conclusions are drawn from the results:

1. The Model 2 tube boiler will produce an exit steam temperature of 720°F at a flow rate of 5.2 lb/hr at 200 psia with an outer tube wall temperature of 1000°F.
2. The optimum length of the insulating sleeve is between 12 and 16 in. for this boiler length.
3. The superheater exit steam temperature is a strong function of the outer tube wall temperature near the exit end of the boiler, but is influenced very little by the temperature distribution over the rest of the length within rather wide limits.
4. The axial variation of heat removal capability of the boiler tube should produce no excessive hot spots in a radioisotope heat block-shield.
5. The Model 2 boiler should operate in a stable manner with flow rates up to 6 lb/hr if a pressure drop of about 100 psi is provided between the feed pump discharge and the boiler.
6. The heat transfer rates in the annulus cannot be predicted by calculations very well, and any new boiler design should be tested to determine its performance.

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Fig. 14. Calculated Temperature Distribution for Model 1 at 5.2 lb/hr Flow and 200 psia.

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Appendix A

EXPERIMENTAL DATA AND RESULTS

Data from the experiments and the calculated results are presented in the following table. Included in the tabulation are the boiler outlet pressure, water flow rate, water inlet temperature, central tube exit temperature, superheater exit steam temperature, outer tube wall temperatures, calculated net heat input to each boiler section, total net heat input, and the calculated amount of heat absorbed by the steam.

Table A-1. Experimental Data and Results

Model No.	Boiler Outlet Pressure (psia)	Water Flow Rate (lb/hr)	Water Inlet Temperature (°F)	Central Tube Exit Temperature (°F)	Superheater Exit Steam Temperature (°F)	Outer Tube Wall Temperature (°F)				Net Heat Input (Btu/hr)					Heat Absorbed by Steam (Btu/hr)
						Top 1	2	3	Bottom 4	1	2	3	4	Total	
1	14.4	0.975	212	950	228	1000	1010	995	998	95	25	33	831	984	954
	14.4	1.36	200	930	248	998	990	990	1010	122	60	127	1009	1318	1360
	14.4	2.0	210	830	345	1010	1005	980	1010	506	131	241	1297	2175	2071
	14.4	2.0	200	880	338	1000	1015	990	1010	210	-18	10	1274	1476	2085
	14.4	2.53	200	810	310	1015	1020	1008	1000	501	110	329	1341	2281	2604
	14.4	2.76	208	750	250	985	988	990	992	513	104	300	1415	2332	2739
	14.4	2.76	200	770	240	1005	1005	1000	1000	649	75	318	1596	2638	2748
	14.4	3.12	198	218	445	965	1040	1035	1018	1642	-38	335	1130	3069	3417
	14.4	4.23	200	230	480	990	1010	950	1010	2286	-63	448	1660	4331	4694
	14.4	5.2	205	230	450	970	1040	978	980	2523	581	230	1626	4960	5670
	14.4	0.975	83	960	215	1000	1010	1010	1060	27	-51	20	656	652	1073
	14.4	2.0	83	830	240	980	980	1005	1000	344	194	360	1437	2335	2226
	14.4	3.12	83	230	380	995	1020	1050	1000	1695	-5	425	1582	3697	3680
	14.4	4.23	83	225	280	955	1055	1045	995	2262	598	681	1689	5230	4788
	54	3.12	280	815	325	1000	1020	1050	1000	442	74	548	1465	2529	2955
	54	3.69	275	750	480	1010	1035	1020	1005	655	82	917	1532	3186	3800
	54	4.23	270	300	530	1000	980	1020	1005	1421	36	1064	1699	4220	4481
	54	5.2	278	295	550	995	1025	1040	1005	2329	128	897	1682	5036	5516
	100	5.2	323	555	552	1010	970	990	990	1408	168	1915	1537	5028	6263
	200	0.975	370	950	390	990	980	1000	1000	33	40	0	864	937	839
	215	2.0	375	940	395	990	1000	1005	990	179	180	406	1289	2054	1711
	205	3.12	380	860	395	1005	1005	1005	990	414	250	494	1333	2491	2659
	205	4.23	370	785	425	1025	1020	1010	990	829	182	1356	1460	3827	3731
	210	5.2	380	670	460	990	1010	980	1000	1210	422	1907	1624	5163	4636
	200	5.2	375	660	450	995	980	1010	1010	1186	234	1878	1558	4856	4640
	202	6.25	375	650	560	1015	1030	990	1010	1114	310	2077	1641	5142	5957
	198	7.23	365	455	590	950	1010	960	990	1770	534	2183	1684	6171	7088
2	14.4	0.975	198	715	235	1016	1014	995	995	128	44	89	513	774	971
	14.4	2.0	200	820	515	1010	1035	990	998	252	-10	335	866	1443	2253
	14.4	2.0	205	805	515	1020	1025	1005	1000	404	158	701	872	2135	2243
	14.4	2.53	205	555	575	990	1010	1020	995	1123	-118	512	881	2398	2898
	14.4	3.12	195	225	600	1040	1040	980	1000	1457	165	944	883	3119	3657
	14.4	4.23	190	224	640	921	992	1014	1003	2300	20	920	1086	4326	5061
	14.4	5.2	202	230	655	890	966	1030	1008	2538	925	1128	1116	5707	6197
	20	6.25	205	240	680	850	1010	1025	1000	2434	1568	1474	1018	6494	7507
	210	2.0	355	920	415	990	980	990	1000	116	387	1081	812	2396	1782
	200	3.12	360	840	560	1000	1000	990	1005	128	-55	1265	741	2079	3020
	215	4.23	360	690	685	1010	1010	980	1000	1042	281	2610	730	4663	4371
	215	5.2	384	645	720	1012	1040	968	1010	1257	453	2038	872	4620	5334
	215	6.25	380	620	735	990	1015	970	1000	1437	1035	2694	928	6094	6487
	205	7.23	382	390	742	1000	1010	940	1000	2144	1627	2700	1076	7547	7550

Table A-1. Continued

Model No.	Boiler Outlet Pressure (psia)	Water Flow Rate (lb/hr)	Water Inlet Temperature (°F)	Central Tube Exit Temperature (°F)	Superheater Exit Steam Temperature (°F)	Outer Tube Wall Temperature (°F)				Net Heat Input (Btu/hr)					Heat Absorbed by Steam (Btu/hr)
						Top 1	2	3	Bottom 4	1	2	3	4	Total	
3	210	5.2	120	610	390	985	990	960	1000	1521	1397	2480	811	6209	5791
	210	5.2	353	515	715	905	908	821	990	1356	1323	1391	1090	5160	5494
	210	5.2	375	382	740	530	620	800	1010	1564	1520	1534	1244	5862	6538
	210	5.2	372	470	488	803	905	897	805	1052	1488	1733	418	4691	4764
	210	5.2	365	510	667	890	888	910	908	1338	1246	1856	716	5156	5300
	215	5.2	365	575	660	898	995	995	900	1034	1094	2266	595	4989	5279
	210	5.2	360	475	645	660	1000	1005	900	528	1531	2247	628	4934	5268
	205	5.2	355	505	590	860	950	955	840	1420	1358	2156	526	5460	5148
	210	5.2	365	565	600	865	1030	1000	860	1183	1221	2330	418	5152	5118
	14.4	0.975	193	925	370	1000	1020	985	990	24	86	254	805	1169	1038
	14.4	2.0	188	720	480	1000	985	1015	1000	425	216	849	847	2337	2244
	14.4	2.53	188	220	470	993	1000	1000	1010	779	167	1127	981	3084	2815
	14.4	3.12	180	220	600	1000	1060	980	1000	1736	10	960	975	3681	3704
	14.4	4.23	190	233	640	928	990	1030	1000	2228	386	1221	1097	4932	5085
	14.4	5.2	188	238	670	900	990	1015	1005	2440	1468	1137	1142	6187	6308
	20	6.25	192	242	680	860	985	1025	995	2435	1766	1509	865	6575	7587
	210	3.12	360	740	490	1015	1025	1010	1000	1818	1370	2358	906	6452	6572
	195	4.23	360	550	700	950	1030	995	1000	1032	424	1792	777	4025	4430
	215	5.2	368	540	715	1020	1045	1000	1005	1294	812	2089	902	5097	5410
	215	6.25	360	500	720	990	1010	1005	995	1818	1370	2358	906	6452	6572
4	14.4	0.975	183	720	412	985	1005	998	1002	234	108	92	391	825	1067
	14.4	0.975	192	910	412	1000	1010	995	990	73	82	264	622	1042	1058
	14.4	2.0	190	970	565	1025	1015	1019	1005	55	587	600	903	2145	2292
	14.4	3.12	190	810	625	1010	1015	1000	985	482	896	1095	832	3305	3710
	14.4	3.69	190	650	652	980	998	1002	990	1039	967	994	978	3978	4425
	14.4	4.73	198	240	690	910	1000	980	1010	1964	1128	1630	986	5708	5737
	20	6.25	195	242	695	920	1010	985	1005	2452	1279	2253	1206	7190	7614
	22	7.23	190	250	696	800	915	930	1002	2395	2259	2307	1347	8308	8833
	210	3.12	365	985	495	1025	1015	1003	1005	116	295	1425	749	2585	2894
	210	4.23	379	710	683	1000	990	1005	995	928	854	1820	914	4516	4303
	205	5.2	358	692	715	1025	998	1007	1000	1246	944	2054	861	5105	5468
	205	6.25	365	580	740	1005	1010	998	1003	1570	1023	2150	879	5622	6606
	210	7.23	373	395	742	955	990	975	1000	2500	1935	2482	1012	7929	7577
	14.4	0.975	180	790	240	980	970	990	990	274	204	398	-46	830	990
	14.4	2.0	205	660	500	960	980	1030	990	389	75	509	641	1614	2200
	14.4	3.12	210	230	575	990	1000	980	1000	1517	83	1333	523	3456	3550
5	14.4	4.23	210	230	590	950	1010	990	995	2306	960	1393	551	5210	4862
	200	1.88	370	910	425	980	990	990	990	-17	-47	1057	570	1563	1660
	195	3.23	365	740	485	1005	1000	995	1000	716	678	1131	506	3031	2983
	200	4.23	355	685	575	1015	995	1005	1000	710	854	1281	515	3360	4146
	215	5.2	370	540	655	955	990	1005	1005	1357	1920	1308	436	5021	5439
	205	6.25	373	390	705	975	980	1010	1005	2496	2258	1371	421	6546	6440
	215	7.23	370	390	718	600	880	1000	1005	2650	2466	1667	332	7115	7508

Table A-1. Continued

Model No.	Boiler Outlet Pressure (psia)	Water Flow Rate (lb/hr)	Water Inlet Temperature (°F)	Central Tube Exit Temperature (°F)	Superheater Exit Steam Temperature (°F)	Outer Tube Wall Temperature (°F)				Net Heat Input (Btu/hr)					Heat Absorbed by Steam (Btu/hr)
						Top 1	2	3	Bottom 4	1	2	3	4	Total	
6	14.4	0.975	203	218	353	975	1025	988	999	276	17	51	230	574	1023
	14.4	2.0	213	994	555	1028	992	1003	1008	258	579	712	840	2389	2242
	14.4	3.12	205	883	620	1025	1020	1000	1005	394	1068	1158	855	3475	3640
	14.4	4.23	200	625	658	1015	1005	995	1005	1295	1112	1486	865	4758	5055
	14.4	5.2	203	245	692	945	1000	981	1015	1569	1018	2403	858	5847	6541
	20	6.25	217	248	705	908	951	943	1005	2410	1514	2465	933	7322	7523
	210	2.0	378	993	445	1007	1011	1000	1008	-101	110	1037	740	1786	1752
	200	3.12	364	973	583	1017	1007	1002	1005	96	444	1754	761	3055	3037
	205	4.23	385	823	685	993	994	1005	1003	707	954	1912	631	4204	4263
	210	5.2	378	718	720	1028	1015	1013	1015	1383	1152	2008	631	5174	5591
	210	5.2	367	688	707	1015	1012	1003	1005	1380	1141	2020	636	5177	5618
	210	6.25	378	543	732	983	1033	1003	1005	1739	1556	2375	657	6327	6509
	195	7.23	385	480	748	983	1038	994	1010	2211	1754	2411	736	7112	7530

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