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

R. S. Holcomb
M. E. Lackey

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REACTOR DIVISION

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Oak Ridge, Tennessee
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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 plants\(^1\) 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
Fig. 1. Diagram of a Reentry Tube Boiler.
shock, and boiling flow instabilities. The reentry tube boiler is particu-
larly 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. 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 Pro-
gram in connection with the possible application of the reentry tube boiler to a radioisotope steam power plant for undersea use. 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.
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

<table>
<thead>
<tr>
<th>Model No.</th>
<th>Inner Tube Plug Length (in.)</th>
<th>Insulating Sleeve Length (in.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>12</td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>12</td>
</tr>
<tr>
<td>4</td>
<td>24</td>
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</tr>
<tr>
<td>5</td>
<td>3</td>
<td>24</td>
</tr>
<tr>
<td>6</td>
<td>3</td>
<td>16</td>
</tr>
</tbody>
</table>

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.
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
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.

- $T_{wall} = 1000^\circ F$
- Flow = 5.2 lb/hr
- $P = 200 \text{ psia}$
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

<table>
<thead>
<tr>
<th>Case</th>
<th>Boiler Section</th>
<th>Tube Wall Temperature (°F)</th>
<th>Heat Input (Btu/hr)</th>
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<tbody>
<tr>
<td>Uniform wall temperature</td>
<td>1 (top)</td>
<td>1012</td>
<td>1257</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1040</td>
<td>453</td>
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<tr>
<td></td>
<td>3</td>
<td>968</td>
<td>2038</td>
</tr>
<tr>
<td></td>
<td>4 (bottom)</td>
<td>1010</td>
<td>872</td>
</tr>
<tr>
<td>Uniform heat input</td>
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<td>905</td>
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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

<table>
<thead>
<tr>
<th>Model No.</th>
<th>Boiler Section 1</th>
<th>2</th>
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<td>Heat Input, Btu/hr</td>
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<td>3</td>
<td>1294</td>
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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 ±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 ±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:

\[
\frac{hD_e}{k_f} = \frac{0.005}{(L/D_e)^{0.1}} \frac{Re_f Pr_f^{0.4}}{Pr_f^{0.4}}
\]

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:

\[
\frac{hD_e}{k} = 1.86 \left( \frac{Re \ Pr}{L/D_e} \right)^{1/3}
\]
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:

\[
q = \frac{A_o \sigma (T_o^4 - T_c^4)}{1/e_o + (1/e_c -1) A_o/A_c}
\]

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,
- \(e_c\) = Emissivity of central tube,
- \(e_o\) = 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.
Fig. 14. Calculated Temperature Distribution for Model 1 at 5.2 lb/hr Flow and 200 psia.
REFERENCES


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

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7-16. A. P. Fraas  
17. J. H. Frye, Jr.  
18. J. H. Gillette  
19. A. G. Grindell  
20. K. W. Haff  
21. H. W. Hoffman  
22. R. S. Holcomb  
23. P. R. Kasten  
24. M. E. Lackey  
25. E. Lamb  
26. M. E. LaVerne  
27. D. B. Lloyd  
28. M. I. Lundin  
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30. R. E. MacPherson  
31. H. C. McCurdy  
32. A. J. Miller  
33. A. M. Perry  
34. R. A. Robinson  
35. M. W. Rosenthal  
36. A. F. Rupp  
37-46. G. Samuels  
47. A. W. Savolainen  
48. A. C. Schaffhauser  
49. M. J. Skinner  
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51. I. Spiewak  
52. D. A. Sundberg  
53. D. B. Trauger  
54. J. J. Tudor  
55. A. M. Weinberg  
56. G. D. Whitman  
57. J. V. Wilson  
58. H. C. Young  
59. Biology Library  
60-62. Central Research Library  
63-64. Y-12 Technical Library  
65-114. Laboratory Records Department  
115. Laboratory Records Department (RC)

## External Distribution

116. F. P. Baranowski, Division of Production AEC, Washington, D.C. 20545
117. D. R. Bartz, Jet Propulsion Lab., NASA, 4800 Oak Grove Drive, Pasadena, California 91103
119. Les Chadbourne, AiResearch Manufacturing Company, 402 S. 36th Street, Phoenix, Arizona 85034
120. D. F. Cope, RDT Site Office, ORNL
121. W. M. Crim, Jr., Research and Technology Department, Building 322, Ft. Belvoir, Virginia 22060
122. R. R. Dahlen, Advanced Engineering, 3M Company, Building 551, 2501 Walnut Street, Roseville, Minnesota 55113
123. R. V. Degner, Rocketdyne, 6633 Canoga Avenue, Canoga Park, California 91303
124. R. N. Endebrock, Isotopic Auxiliary Power Branch, AEC, Washington, D.C. 20545
125. R. E. English, NASA, Lewis Research Center, Cleveland, Ohio 44135
126. E. E. Fowler, Division of Isotopes Development, AEC, Washington, D.C. 20545
127. N. C. Gibbon, Cryogenic Products Department, Linde Division, P.O. Box 44, Tonawanda, New York 14150
128. J. C. Graf, General Electric, Valley Forge, Pennsylvania 19481
129-133. W. D. Holloman, Division of Reactor Development and Technology, AEC, Washington, D.C. 20545
134. A. E. King, Westinghouse, Lima, Ohio 45801
135. M. Klein, Division of Reactor Development and Technology, AEC, Washington, D.C. 20545
136. G. S. Leighton, Sundstrand Aviation, 1100 Connecticut Avenue, N.W., Washington, D.C. 20036
138. S. V. Manson, NASA Headquarters, Washington, D.C. 20546
139. R. F. Mather, NASA, Lewis Research Center, Cleveland, Ohio 44135
140. B. T. McCauley, Aerojet-General Corp., Azusa, California 91702
141. R. E. Miggeman, Sundstrand Aviation, 4747 Harrison Avenue, Rockford, Illinois 61101
152. J. Pidkowicz, RDT Site Office, ORNL
153. W. D. Pouchot, Westinghouse Astronuclear Laboratory, P.O. Box 10864, Pittsburgh, Pennsylvania 15236
154. C. R. Ross, Savannah River Laboratory, Aiken, S.C. 29801
155. George Shepherd, Fairchild Technology Corp., One Goddard Drive, Rockaway, New Jersey 07866
156. G. W. Sherman, Aeronautical Space Power Division, Air Force Aeronautical Propulsion Laboratory, Wright-Patterson Air Force Base, Ohio 45433
157. Anthony Stathoplos, Nuclear Technology Corporation, 116 Main Street, White Plains, New York 10600
158. B. Sternlicht, Mechanical Technology, Inc., 968 Albany-Shaker Road, Latham, New York 12110
159. D. Stewart, Battelle Northwest Laboratory, Richland, Washington 99352
160. P. Swenson, Jr., Swenson Research Inc., 5135 Richmond Road, Bedford Heights, Ohio 44146
161. J. E. Taylor, Thompson-Ramo-Wooldridge, 23555 Euclid Avenue, Cleveland, Ohio 44117
163. Tom Widmer, Thermo Electron Corporation, 85 First Avenue, Waltham, Massachusetts 02173
164. E. S. Wilson, Space Nuclear Systems, AEC, Germantown, Md. 20767
165-166. Division of Technical Information Extension (DTIE)
167. Laboratory and University Division, ORO
168-170. Director, Division of Reactor Licensing, AEC, Wash.
171-172. Director, Division of Reactor Standards, AEC, Wash.