Experimental Investigation of a Once-Through Coaxial Steam Generator

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Tests were conducted on a straight-tube coaxial heat exchanger to determine heat-transfer rates for the generation of superheated steam from heat supplied by a solid-propellant, hot-gas generator. Superheated steam at 1600°F and 1500 psia was generated by hot gas at 2200°F that was introduced into a coaxial counterflow tube-in-tube heat exchanger. Repeated runs were made without burnout or noticeable decrease in heat-transfer efficiency. Stable equilibrium conditions were obtained within 60 sec, and sufficient steam was generated from cold water within 10 sec to provide partial power for a steam-turbine powerplant. Over-all heat-transfer rates were determined for superheat, boiling, and preheat regions at high heat flux and large temperature differences. Experimental results were in satisfactory agreement with heat-transfer calculations for forced convection inside tubes. Average heat-transfer rates of over 0.25 × 10⁶ Btu/hr-ft² were obtained for both steam and hot gas, with local rates as high as 0.85 × 10⁶ Btu/hr-ft² at a mass velocity of 1.5 × 10⁶ lb/hr-ft² and a Reynolds number of 1 × 10⁶.

Nomenclature

A = cross-sectional area, surface area, ft²
C = wetted perimeter, ft
c_p = specific heat at constant pressure, Btu/lb-°R
D = diam, ft
D_e = effective diameter, $D_e \equiv 4A/C$, ft
f = friction factor in Fanning equation
G = mass velocity, $G = W/A$, lb/sec-ft²
g = gravitational acceleration, ft/sec²
H = enthalpy, Btu/lb
h = coefficient of heat transfer between surroundings at t_a and surface at t_s
k = thermal conductivity
L = length
Pr = Prandtl number
p = pressure
Q = rate of heat transfer
R = gas constant
Re = Reynolds' number
r = radius
T = temperature, °R, °F as noted
t = time
U = over-all heat-transfer coefficient, $U \equiv Q/A\Delta T$
V = velocity, fps
v = specific volume
w = mass flow rate
μ = viscosity
ρ = density

Introduction

IN 1958, investigation of thermal powerplants for deep, underwater vehicles indicated that a closed-cycle steam system would provide satisfactory performance if a very compact boiler and condenser were available. Preliminary design calculations suggested that a "once-through," coaxial-tube steam generator or boiler would provide the most compact arrangement within a limited space. Calculations showed that satisfactory heat-transfer rates would be possible if the equations generally used for forced convection were valid for the high temperature differences obtained by the combustion of a slow-burning solid propellant with a flame temperature of approximately 2200°F. Limited experimental data were available at that time for highly superheated steam, especially at temperatures up to 1600°F, and also for forced convection boiling at bulk-temperature differences between steam and hot gas exceeding 500°F.

The most critical question appeared to be stability in the evaporation zone with the high temperature differences that are normally considered to be in the "burnout" or film-boiling region. However, since calculated heat-transfer coefficients for steam in the superheat region were sufficiently high, it was thought that the coefficient for film boiling would also be satisfactory if pulsations caused by instability did not occur. Warm-up time was also considered. Although it is difficult to predict for actual dynamic start-up conditions, preliminary estimates indicated that less than 1 min would be necessary to warm up to full-flow steam conditions, starting with the annular steam passages being initially filled with water at room temperature.

Test Setup and Procedure

The once-through boiler designed for the closed-cycle propulsion system for deep submergence, as described in Ref. 1, consisted of a single-pass concentric tube exchanger, shown in Fig. 1. The hot gas from a solid propellant passed through the inner tube, and the water-steam flowed through the annulus in a counterflow direction to the hot gas. The outside tube was $\frac{5}{8}$ in. in diameter and the inside tube was $\frac{7}{16}$ in. in diameter, each with a 0.035-in. wall thickness and with the annular space kept concentric by four wires wound in a 45° helix on the inner tube. The tubes were 35 ft long and wound into a 4-row-long, 4-row-deep coil that was less than 4 in. long and 11 in. in o.d., weighing ~15 lb.

The configuration of this coiled tube was difficult to instrument for temperature profile measurements in the various steam zones. For test purposes, a concentric tube of similar size and length, but mounted within an insulated guard tube, was constructed, as shown in Fig. 2. Thermocouples were mounted every 2 ft along the outer tube for measuring the water or steam temperature in the annulus. Water-inlet and steam-outlet pressures were measured to determine the over-all pressure drop.

The hot gas was provided by the combustion of a slow-burning ammonium nitrate base, end-burning-type grain. Inlet and exhaust temperatures and pressures were measured, but

Presented as Preprint 64-468 at the 1st AIAA Annual Meeting, Washington, D. C., June 29–July 2, 1964; revision received October 13, 1964.

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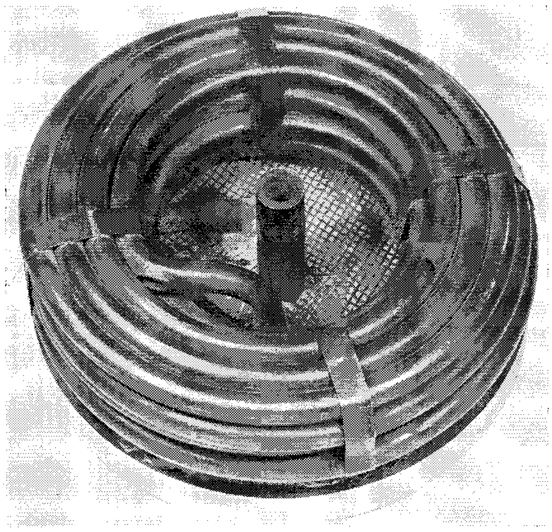

Fig. 1 Heat-exchanger coil and head assembly.

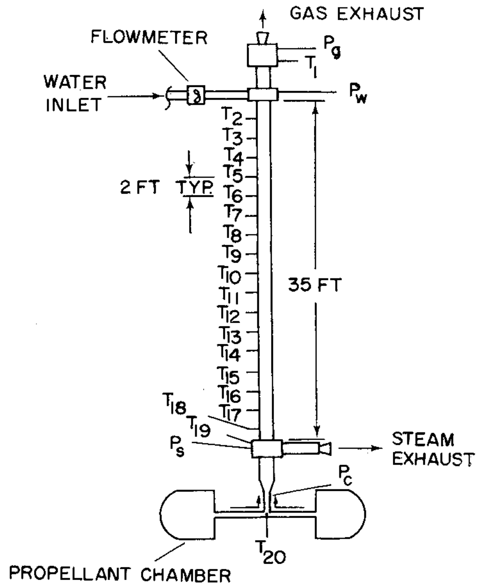

Fig. 3 Experimental heat-exchanger instrumentation.

intermediate measurements along the tube were not made. The burning rate of this grain varies as a function of pressure, but the flame temperature is relatively constant at approximately 2200°F. No attempt was made to vary the flow of the hot gases during a test because stability was considered to be one of the most important parameters.

As shown in Fig. 3, the feed water was supplied to one end of the annular space of the straight, double-tube boiler from a pressurized feed-water tank, and water flow was measured by a Waugh flowmeter. The water was introduced at a predetermined mass flow rate prior to firing the grain, and was maintained at a constant flow rate by either a cavitating venturi or a Waterman flow regulator. A fixed orifice was placed in the steam-discharge line to maintain high pressure at equilibrium conditions.

The hot-gas flow was initiated by ignition of a test-length grain in a combustion chamber, and combustion products were introduced directly into the inner tube of the test boiler. The hot-gas pressure was controlled by throttling the gases after they left the heat exchanger to maintain their pressure at approximately the same as that of the steam. Temperature and pressure readings were taken in the feed line from the grain chamber and in the exhaust line before the gases were throttled. The hot-gas flow was varied by the feeding of either a single grain or two grains simultaneously into the heat exchanger. One-or two-minute test grains were used, and stable test conditions could be attained in less than 1 min. Test data were then taken from oscillograph records during this steady-state condition. The number of tests was limited by the test grains available. No attempt was made to clean the test heat exchanger between runs.

A total of 12 runs was made on the straight-tube boiler, 8 with hot gas and 4 with either water or plant steam for pressure-drop or low-temperature tests, the last two being made with steam from the boiler used to run a turbine. A coiled-tube boiler of similar length and tubing size (Fig. 1) was then used for propulsion system tests and the warm-up time was

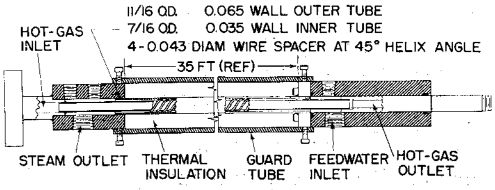

Fig. 2. Coaxial-tube test boiler.

noted. Pressure-drop correlations were also made with the straight-tube boiler.

Heat-Transfer Correlation

Table 1 shows typical data (run 10). Because of the high temperatures encountered in the tests and the resultant reduction in tube strength, thermocouples were soldered to the outside surface of the steam tube, and insulation was used to cover the tube and thermocouples. This arrangement necessitated a slight correction in temperature measurements to compensate for heat losses. The correction was estimated as follows: the location of the saturated liquid point was determined by observing the change in slope of the water temperature. From head-loss data, the approximate head loss at this point was determined. Subtracting the calculated head loss from the water-inlet pressure gave the actual saturation pressure and temperature. The difference between recorded temperature and actual saturation temperature was used as the correction factor. Figure 4 shows corrected temperature vs tube positions for runs 9 and 10.

For the results in this paper, the characteristic area A was the inside surface of the propellant-gas tube. Figure 5 shows a cross section of the heat exchanger with pertinent dimensions and calculated effective diameters.

The gas temperature vs tube position was calculated by balancing the heat absorbed by the steam and working from

Table 1 Typical recorded heat-transfer data (run 10)[a]

P_c	1570	T_5	345
T_{20}	2250	T_6	442
P_g	1360	T_7	390
T_1	398	T_8	420
P_s	1315	T_9	455
T_{19}	1370	T_{10}	Not connected
P_w	1780	T_{11}	540
T_w	90	T_{12}	640
w_g	0.346 (calculated from P_c)	T_{13}	605
w_s	0.1795	T_{14}	590
T_2	173	T_{15}	580
T_3	267	T_{16}	590
T_4	310	T_{17}	825
...	...	T_{18}	1145

[a] P = pressure, psia; T = temperature, °F; and w = flow rate, lb/sec.

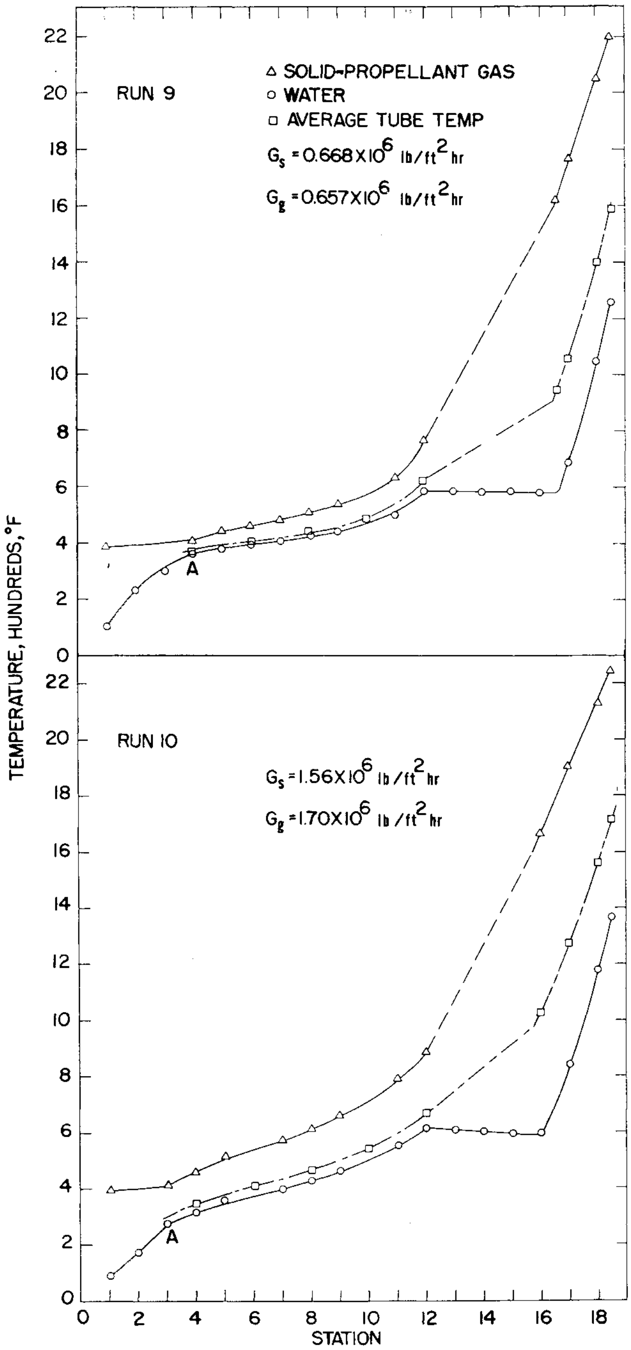

Fig. 4 Typical temperature profiles.

two known propellant-gas temperatures and tube positions. The equations used for these calculations were

$$\begin{aligned} \Delta H_g &= \Delta H_s w_s / w_g \\ \Delta T_g &= \Delta H_g / c_{pg} \end{aligned} \tag{1}$$

Since the propellant exhaust product was a mixture of five gases, the mole fraction of each as a function of temperature was required to calculate physical properties.§ This was accomplished by standard methods utilizing equilibrium constants.[2] Figure 6 shows mole fractions and the calculated specific heat at constant pressure c_p vs temperature. The dew points were determined to be 410°F at pressure of 1360 psia and 406°F at pressure of 1290 psia. The specific en-

§ Calculations were made by J. M. Caraher, Chemical Engineer, Underwater Ordnance Department, U. S. Naval Ordnance Test Station, Pasadena, Calif.

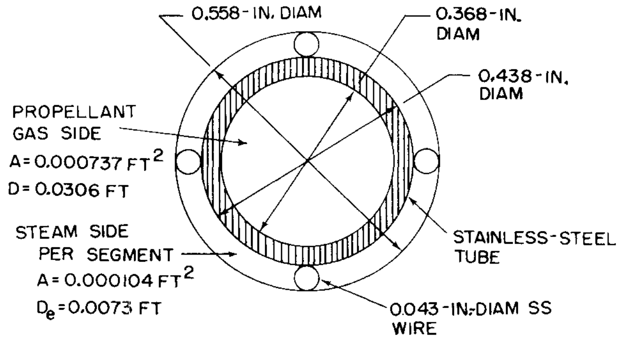

Fig. 5 Detail of tube cross section.

thalpy of steam was obtained[3] by using the corrected temperature and estimated pressure. Pressure drop down the tube was assumed to be linear between specific fixed points. These points were the measured pressure at both ends of the tube and the saturation points.

Figures 7 and 8 show calculated enthalpy of propellant gas vs temperature starting with zero enthalpy at 2230°F working down (heat loss) and 400°F working up (heat gain). The fixed points in the tube used as a base were station 19 (Fig. 3) at the hot end and the dew point at the cold end. The position in the tube where water started condensing from the propellant gas was the point where the water temperature vs tube position experienced an abrupt change in slope (point A, Fig. 4).

Figure 4 shows the calculated propellant-gas temperature vs tube position for runs 9 and 10, and Table 2 shows the tabulated calculations for run 10. Over-all heat-transfer coefficients were calculated as

$$Q = UA\Delta T_m = wdH \tag{2}$$

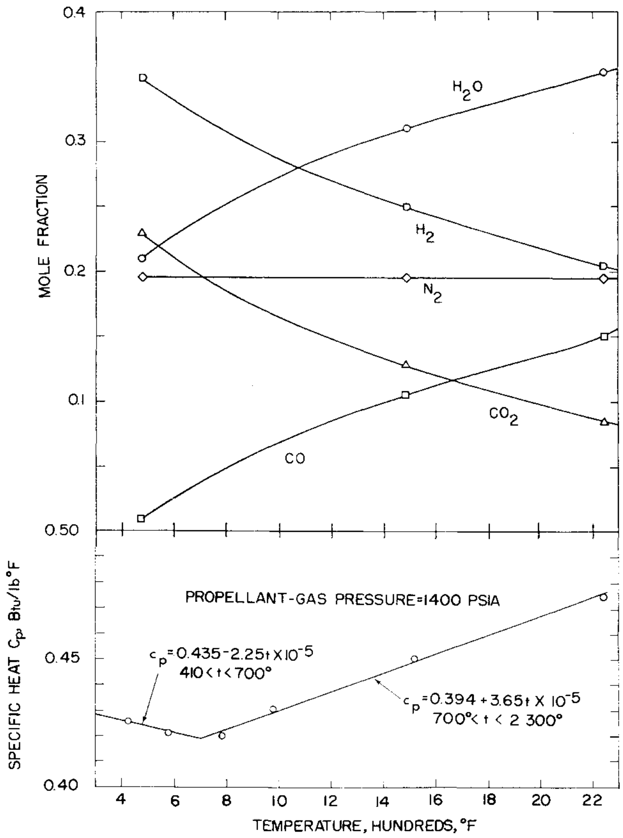

Fig. 6 Specific heat at constant pressure.

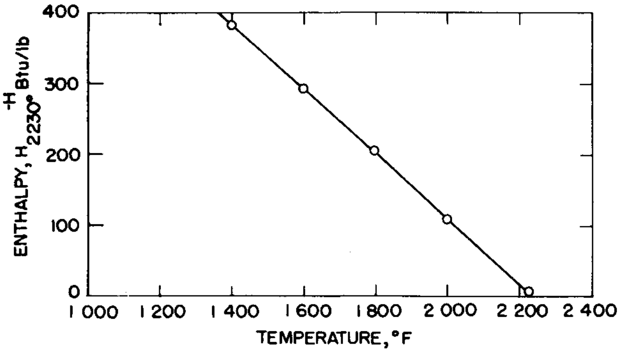

Fig. 7 Enthalpy of propellant gas vs temperature, T_1 = 2300 °F.

where

$$\left.\begin{aligned} \Delta T_m &= \text{log mean temperature difference} \\ A &= C\Delta L \\ U &= w\Delta H/C\Delta L\Delta T \text{ for finite tube lengths} \end{aligned}\right\} \quad (3)$$

$$U = \frac{w}{C}\frac{dH}{dL}\frac{1}{(T_g - T_s)} = \frac{w}{C}c_p\frac{dT}{dL}\frac{1}{(T_g - T_s)} \text{ for point values} \quad (4)$$

Figure 9 shows the experimental value of U vs tube station for runs 9 and 10. These values of over-all heat-transfer coefficient U were used to check the theoretical U, calculated using the forced convection equation. This basic equation is

$$h_L D/k = 0.023\,(Re)^{0.8}Pr^{0.4} \quad (5)$$

This equation was modified for the calculation of superheated-steam and propellant-gas film coefficients. McAdams[4] suggested that, for large temperature differences, a mean or film temperature should be used to evaluate all of the physical properties except fluid velocity.

By rewriting Eq. (5) and evaluating all of the properties at the film temperature except gas velocity, we get

$$h = 0.023\,c_p V_b \rho_f (D V_b \rho_f/\mu_f)^{-0.2} Pr_f^{-0.6} \quad (6)$$

By using the mass velocity of the fluid and the equations of state $p = \rho RT$ and continuity $W = \rho_b V_b A$, we get

$$V_b \rho_f = G(\rho_f/\rho_b) = G(T_b/T_f) \quad (7)$$

$$\left.\begin{aligned} h &= 0.023\,c_{p_f} G(T_b/T_f)(DG/\mu_f)T_b/T_f)^{-0.2}Pr_f^{-0.6} \\ h &= 0.023\,c_{p_f} G^{0.8}\mu_f^{0.2}(T_b/T_f)^{0.8}(D)^{-0.2}Pr_f^{-0.6} \end{aligned}\right\} \quad (8)$$

The equation was written in this form in order to separate the temperature dependent from the independent factors.

The thermal properties of the two fluids are presented by Keenan[3] and Hilsenrath.[5] The propellant-gas properties were averaged on a mole fraction basis, using the table values for each constituent at its temperature and partial pressure. In the cases where temperatures were higher than those the table[3] presented, generalized properties were evaluated and the standard atmosphere data of Kreith[6] were utilized. The generalized properties are presented by Comings[7] as a function of reduced temperature and pressure. For the noncircular cross-sectional area in the annulus, the effective diameter D_e was used. Very little heat-transfer work had been done before under this extreme of temperature and temperature difference, and the data found[8] were largely confined to Reynolds numbers less than 10^5. The data obtained in the present work did not allow independent determination of the film coefficients. However, by calculating the thermal properties of the hot gas[5] and utilizing the available thermal properties of steam,[3] satisfactory correlation of the test data with heat-transfer calculations for forced convection was obtained.

Although the hot-gas temperature could not be accurately determined in the boiling region, calculation of the over-all heat-transfer coefficient from steam to the hot gas indicated that the heat-transfer coefficient for steam during forced convection boiling was comparable to the calculated value for single-phase conditions, as shown in Fig. 9. In Fig. 10, the results of four runs are compared with the forced convection equation. Much better correlation is obtained when film temperatures are used instead of bulk temperatures, as noted by McAdams.[4]

Pressure-Drop Correlation

The over-all pressure drop across the heat exchanger was measured for both steam and hot-gas passages for all of the tests, but no intermediate measurements were made because of the danger of tube failure of pressure taps at the high pressures and temperatures prevailing during the heat-transfer tests. The pressure drop through the annulus used for steam

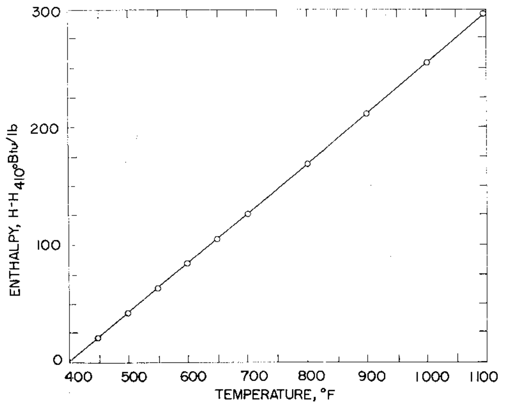

Fig. 8 Enthalpy of propellant gas vs temperature, T_1 = 400 °F.

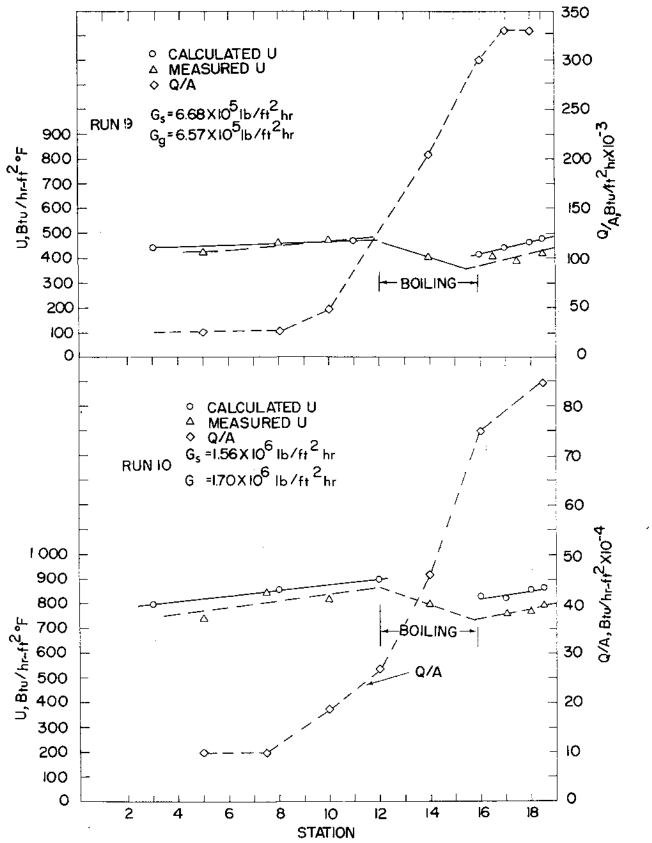

Fig. 9 Over-all heat-transfer coefficient and unit heat flux, run 10.

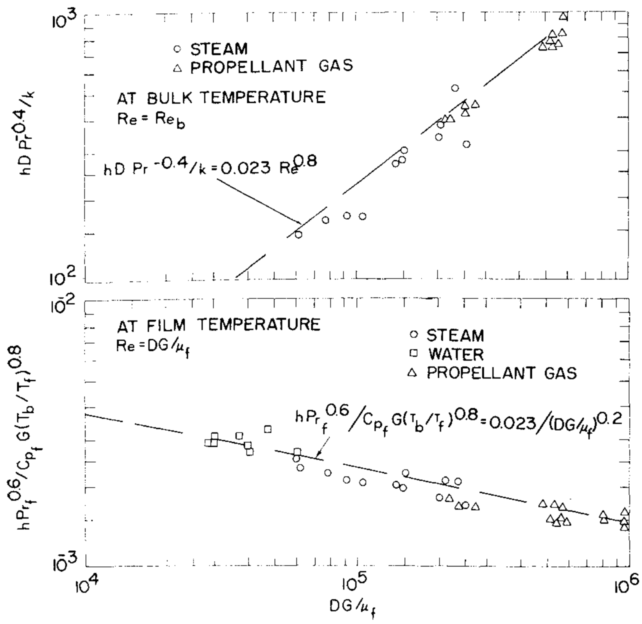

Fig. 10 Correlation of forced-convection data at film temperatures.

passage was initially determined with cold water at various mass velocities as shown in Fig. 11 for both the straight-tube and coiled-tube boilers. The calculated pressure drop based on the Fanning equation

$$-dp = 2fG^2dL/D_e g\rho$$

is also shown plotted in Fig. 11 for water at 60°F. Good agreement with experimental results was obtained for the straight-tube boiler at the higher mass velocities, but the calculated values were as much as 30% less at the lower flow rates corresponding to Reynolds numbers of 1500 to 2000. Similar pressure-drop tests with water also conducted on the coiled-tube boiler indicated a pressure drop ~25% higher than that of the straight tube.

As expected, the pressure drop during heat-transfer tests with two-phase flow was much higher, especially with high superheat where the Reynolds number exceeded 10^5. Correlation for these test conditions has not been made because intermediate pressure-drop measurements in the preheat, boiling, and superheat zones were not available.

The pressure drop across the inner-tube, hot-gas passage was also measured during the heat-transfer test runs. By using the modified Fanning equation for gases[4] where

$$p_1 - p_2 = [G^2(v_2 - v_1)/\alpha g] + [f_m G^2 v_m L/2_g r_h]$$

Table 2 Typical propellant-gas temperature calculation (run 10)[a]

Station	T_s, °F	P_s, psia	h_s, Btu/lb	Δh_s, Btu/lb	Δh_g, Btu/lb	T_g, °F
19	1370	1315	1707	...	...	2250
18	1180	1338	1599	108	56	2130
17	845	1390	1399	308	160	1905
16	600	1430	1186	521	271	1665
12	615	1723	639	390	202	885
11	555	1730	558	309	161	790
9	465	1738	451	202	105	660
8	430	1745	413	164	85	610
7	400	1752	380	131	68	570
5	358	1766	335	86	45	515
4	315	1773	290	41	21	460
3	275	1780	249	...	...	410

[a] W_s = 0.180 lb/sec and W_g = 0.346 lb/sec.

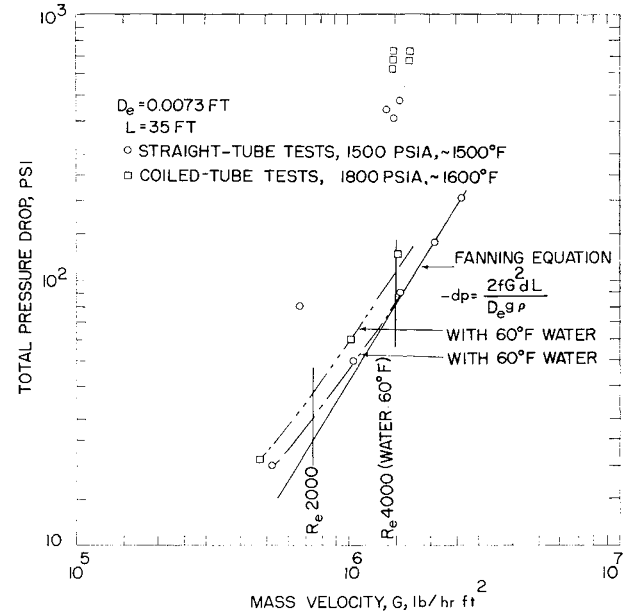

Fig. 11 Pressure drop in annulus of coaxial-tube boiler.

fairly good correlation was obtained with test data. With the large grain producing ~0.345 lb/sec of hot gas at 1500 psia, entering at 2200°F and exiting at 400°F, the average pressure drop from runs 10–12 was 190 psi, as compared with 179 psi from the preceding equation, or 206 psi if the first term for pressure regain caused by gas cooling is neglected. The pressure drop for similar flow conditions but with the coiled tube was higher by a factor of ~1.5.

Good correlation was also obtained with a lower flow rate of ~0.15 lb/sec. In this case, the average pressure drop from three tests was 52 psi as compared with a calculated value of 44 psi, or 52 psi if the pressure regain is neglected.

The warm-up time required for the straight-tube test boiler to produce steady-state conditions was approximately 1 min. The test boiler was surrounded with thermal insulation, which added slightly to the heat capacity of the system. The tests on the coiled-tube boiler mounted inside the grain combustion chamber were considered more significant, and warm-up time was recorded in more detail, as indicated in Fig. 12. These results were obtained from tests with the coiled-tube boiler supplying steam to a turbine. Partial power was available at 10 sec and full power in less than 1 min. No instability or pressure fluctuation occurred during build-up to steady-state conditions.

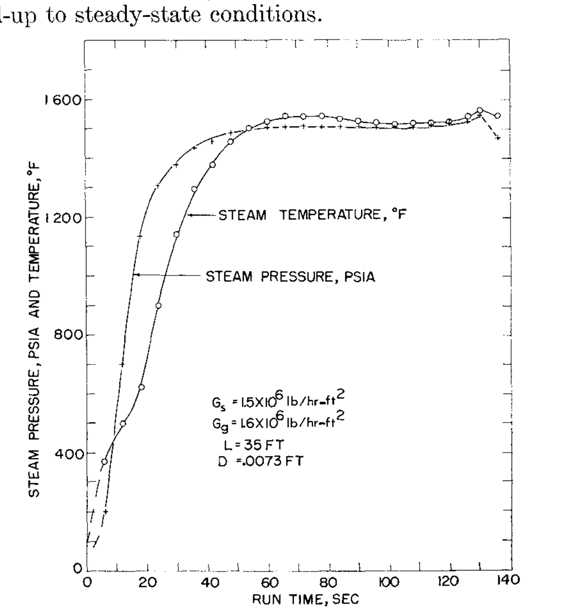

Fig. 12 Warm-up time for steam with coiled-tube boiler, run 7.

Over-all heat-transfer tests on a tightly coiled, coaxial-tube boiler indicated approximately 20% greater heat-transfer coefficients, but with an increase in over-all pressure drop of 50%. Reynolds' analogy[4] indicates that the film coefficient should vary directly with pressure drop for equal mass velocities, and it may be assumed that the variance shown in the tests was caused by the reduction of flow area when the tubes were wound into the tight coil.

Conclusions

The experimental data obtained from the tests on the straight coaxial-tube boiler indicate that the conventional heat-transfer relationship for forced convection can be used for steam at temperatures up to 1600°F and 1500 psia and for propellant hot gas up to 2000°F at mass velocities as high as 1.5×10^6 lb/hr-ft^2 and temperature differences up to 800°F.

Stable conditions were obtained for forced convection boiling between mass velocities of 0.5 and 1.5×10^6 lb/hr-ft^2, and no instability was noted at any time in tests using hot gas from the solid propellant as the heat source.

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MAY-JUNE 1965 J. SPACECRAFT VOL. 2, NO. 3

Dependence of Ground Station Acquisition Effectiveness on Geographic Location and Satellite Orbit

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The "effectiveness" of any geographical location as a ground site for satellite data acquisition depends significantly upon 1) length of time available for acquisition during each pass and 2) elevation angles to the satellite during acquisition. Together, these two criteria determine the percentages of passes from which data may be acquired. The geometric relationships are examined analytically to determine the dependence of station "effectiveness" on satellite altitude and orbital inclination. In general, both elevation angles and acquisition times are larger (and site effectiveness therefore greater) for polar orbits (inclination = 90°) than for sun-synchronous orbits (where the inclination departs from 90° with increasing altitude). For a sun-synchronous orbit, a single station at temperate or subpolar latitudes below 69° can never acquire data from every daily pass, regardless of satellite altitude. For any station, the percentage of passes acquirable from a sun-synchronous orbit is maximum when the satellite is near 1200-naut-mile alt. For a polar orbit, a station at 65° latitude (viz., Fairbanks, Alaska) can acquire data from every pass at all altitudes above 550 naut miles. For any station, the percentage acquisition, average elevation angle, and length of acquisition time increase with increasing altitude.

Effective Acquisition Range

A COMMAND and data acquisition (CDA) station can successfully interrogate a satellite only while the satellite is within radio range for a period of time long enough to acquire data. The satellite rises above the station's effective radio horizon during only a small portion of any one orbital pass, and in general there may be one or more passes each day on which the satellite fails to appear above the station's horizon at all. The effectiveness of any geographic location as a CDA station site depends in part on the percentage of each day's passes on which the satellite comes within acquisition range for an adequate length of time.

The effective acquisition range (EAR) is defined as the distance on the earth's surface from a CDA station within which the subsatellite track must pass if the satellite is to remain for a length of time t or longer above an elevation angle e from the station. The EAR is a function of the e, minimum elevation angle for data acquisition at the station; t, the time period that satellite must remain above e for adequate data acquisition; and h, the satellite's orbital altitude above the earth's surface. The EAR is geometrically independent of the station's geographic location, although local terrain features and radio transmission conditions may affect e. It is conveniently measured in degrees of geocentric arc. Figure 1 shows the EAR for two special cases of $t = 0$: $EAR_{0,0}$ is the "grazing" EAR for $e = 0$, and $EAR_{e,0}$ is the "grazing" EAR for elevation angle e. For a spherical earth of radius R, and a circular orbit,

$$EAR_{0,0} = \cos^{-1}[R/(R + h)] \tag{1}$$

Presented as Preprint 64-319 at the 1st AIAA Annual Meeting, Washington, D. C., June 29–July 2, 1964; revision received November 18, 1964.

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