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Design, Analysis, and Performance of an
Experimental Liquid Hydrogen-Water
Heat Exchanger*

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by

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Design, Analysis, and Performance of an Experimental
Liquid Hydrogen-Water Heat Exchanger*

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ABSTRACT

A 160-ft long liquid hydrogen-water heat exchanger was designed, built, and tested. It consisted of two 80-ft passes, comprising a 4.06-in. I.D. finned aluminum pipe inside a 9½-in. I.D. outer pipe. Hot water (the heat source) flowed in the outer jacket countercurrently to the hydrogen in the inner line. The outlet hydrogen gas was used, in part, to drive the turbine of the liquid hydrogen pump which supplied the heat exchanger with liquid. Tests were made at hydrogen flow rates up to 15 lb/sec with pressure levels up to 1000 psi. The outlet gas temperature at the high flow rates was 260°R.

Tests revealed a problem of large flow and pressure oscillations in the system. A simple dynamic model explaining this behavior and a method of stable operation suggested by the model are described. This method was successfully used.

Hydrogen heat transfer coefficients were obtained. These were in general 60% of the values predicted from the Dittus-Boelter equation. In an effort to improve the hydrogen heat transfer, a twisted tape swirl inducer was included in one of the passes. This gave coefficients that were 90% of those predicted by the Dittus-Boelter equation. Experimental axial temperature gradients are compared to those predicted by means of analog and digital methods. Fin efficiencies and strain data were obtained.

*Work performed under the auspices of the U.S. Atomic Energy Commission.

Introductory Remarks

Large quantities of gaseous hydrogen will be required to power the LH_2 turbopump at the Phoebus reactor test facility (Test Cell C) at the Nuclear Rocket Development Station (NRDS) in Nevada. The quantities required make it economically infeasible to store the hydrogen as a high pressure gas. Therefore hydrogen is to be stored as a liquid and vaporized immediately prior to use. Hot water was chosen as an inexpensive and efficient heat source to do this job.

Liquid hydrogen at flow rates of 50 lb/sec must be vaporized and raised in temperature to room temperature to make it usable as an energy source for the pump turbine. This requires a total heat flux exceeding 77,000 Btu/sec. In order to determine if this were practical in a double-pipe heat exchanger using water, a small-scale model containing a 1-in. I.D. inner pipe 47-ft long was built and tested at Los Alamos. As a result of these tests a full-scale prototype of 4-in. I.D. inner line 160-ft long was constructed and tested at NRDS. The purpose of this paper is to report on the results of this full scale prototype experiment.

In the Nevada tests, liquid hydrogen was supplied to the exchanger at 1000 psi by means of the turbopump which was in turn powered by the exit hydrogen gas from the exchanger (Fig. 6). This type of interlinked operation demands a steady, nonpulsating flow of gas from the unit. After some initial difficulties, this condition was successfully achieved.

As a result of tests on the prototype, the full-scale heat exchanger has been built. It consists of four parallel units each 320 ft long comprising a 4½-in. I.D. hydrogen line inside of a 10-in. I.D. outer water jacket.

Design and Analysis

The feasibility of the design was proved in the small-scale (1-in. diameter) experiments at Los Alamos. Details of these tests have been reported in the literature.¹ In brief, these tests led to the following conclusions:

- 1) Experimental heat transfer coefficients to hydrogen averaged one-half of those predicted by the Dittus-Boelter equation
$$\frac{hD}{k_b} = 0.023 (Re_b)^{0.8} (Pr_b)^{0.4} .$$
- 2) The introduction of a twisted tape swirl inducer in the hydrogen line raised the coefficients to the Dittus-Boelter values.
- 3) Pressure oscillations were potentially troublesome, particularly at pressure levels below 200 psia.
- 4) The formation of ice on the outside of the inner hydrogen line was not excessive and presented no operating difficulties.

Because of the success of the small-scale experiments, a full-scale prototype was designed for testing. It consisted of an aluminum extrusion inner tube 0.75 in. thick with an inside diameter of 4.063 in.

and employing 12 external longitudinal fins 1.5 in. high. The outer jacket was a standard 10 in. schedule 20 pipe. The liquid hydrogen flowed inside the aluminum extrusion with counter flow hot water between the extrusion and the 10 in. pipe.

The prototype exchanger consisted to two 80-ft sections of the full scale exchanger using the aluminum extrusion as described above. The two sections were instrumented with temperature and pressure transducers at 20-ft intervals (see Fig. 1) as well as with strain gauges. A twisted tape swirl inducer was installed in one of the 80-ft sections and the exchanger was plumbed so that both hot water and liquid hydrogen could be introduced into either section.

Prior to testing, careful analysis was made of the expected heat transfer characteristics of the prototype. The differential equations relating the heat fluxes, temperatures and thermal impedances as a function of heat exchanger length were programmed for the MIMIC digital computer code to determine the heat exchanger performance at steady state. The code computes heat transfer coefficients on the water and on the hydrogen side by the Dittus-Boelter equation using variable fluid properties and determines the thermal conductivity of the aluminum tube as a function of a weighted average material temperature. The code then computes the water temperature, water-ice or metal surface temperature, outside wall temperature, inside wall

temperature, hydrogen temperature and the heat transfer coefficients as well as ice thickness, metal thermal conductivity and heat flux. The code takes hydrogen inlet flow rate and enthalpy, water outlet temperature, water flow rate, and the physical dimensions of the heat exchanger and integrates along the length of the exchanger to obtain the exit hydrogen temperature and inlet water temperature as well as the above-listed quantities at all points along the exchanger length.

In Fig. 2, the results from the code are shown for the inlet hydrogen temperature and exit water temperature which existed during a run on the full scale prototype at NRDS. Also shown on Fig. 2 are the temperatures as measured on the prototype during this run. These will be discussed later.

Using the computed hydrogen temperature and heat transfer coefficient and the computed water temperature and heat transfer coefficient at the point of maximum temperature difference across the aluminum tube, a conductive paper analog utilizing a $1/24$ th sector of the aluminum tube was constructed to determine the temperature distribution through the tube. This temperature profile was used to make stress calculations to be compared with measured values. The temperature distribution in the section of the tube is shown in Fig. 3. Stress calculations made from this information agreed well with measured values. From Fig. 3 it is not clear that the fins are very effective in increasing the heat flux. To clarify this point and investigate the possibility of decreasing thermal stresses and reducing fabrication costs, a conductive

paper analog was constructed of a tube section without fins using heat transfer coefficients computed with the modified water side area and wetted perimeter. A comparison of these two conductive paper analog results indicated that the heat flux should be increased in the finned tube by a factor varying from about 4 where ice formation is heavy to about 2 in the no ice section. The factor of 2 is probably the result of the increased heat transfer coefficient in the finned exchanger rather than the result of increased heat transfer area.

In view of these results, the design of the full-scale exchanger extrusion was modified by increasing the inside diameter from 4.063 to 4.463 in. and reducing the fin height from 1.5 in. to 1 in. It was also decided that the full-scale exchanger should be 320 ft in length.

Using this new exchanger geometry, the MIMIC code was again used to calculate the heat exchanger temperatures for the expected operating condition of 8.9 lb/sec per section. Figure 4 shows the results of this calculation which indicates a hydrogen exit temperature of about 550°R or 90°F. Figure 5 shows the computed exit hydrogen and water temperatures as a function of hydrogen flow rate for the 320 ft exchanger with 200°F inlet water at 187 lb/sec for steady state operation.

Field Testing

The full-scale prototype of the heat exchanger was tested at the NRDS in Nevada. Figure 6 is a flow schematic of the facility used for

the testing. The equipment shown is normally used for reactor testing, but with some modification was adapted for use in this rather unusual application. The primary objectives of the experiment were to:

- 1) Determine the steady state heat transfer coefficients at design pressure, flow and heat flux conditions.
- 2) Determine whether pressure oscillations will be present during startup and/or steady state operation.
- 3) Determine the effect of a twisted tape swirl inducer (turbulence promoter) on the H_2 heat transfer coefficient and pressure oscillations.
- 4) Determine if the presence of fins on the outside of the H_2 tube presents any ice formation problems.
- 5) Study turbine startup and switchover from bottle H_2 to heat exchanger H_2 .

Liquid hydrogen was supplied to the test section by means of a Rocketdyne NFS-2 pump-system. This pump was capable of delivering the hydrogen at pressures up to 1000 psi. Since the heat exchanger presented an unusually high impedance for the pump, liquid hydrogen was bypassed through valve ACV-2 on the flow schematic to a flare stack where it was burned. Flow to the heat exchanger was through the throttle valve, ACV-16, and measured by means of a Venturi F30. In order to avoid a hydrogen air mixture in the heat exchanger, it was necessary to supply hydrogen gas through PCV-89. Water through the outer jacket of the test section was supplied from an insulated tank by a commercial

water pump. The water in the tank was heated with a propane heater to the desired temperature before each test. Water at 200°F was used where high effluent gas temperatures were required to drive the turbine; for those runs designed to simulate design conditions in the two test sections, 130°F water was used.

Figure 6 shows the system with the heat exchanger effluent used to power the turbine which in turn drives the liquid hydrogen pump. Other experiments were performed in which hydrogen effluent was flared to the atmosphere after passing through the impedance valve, AVV-23.

The first attempts to operate the heat exchanger involved chilling the liquid hydrogen system to the throttle valve, ACV-16, and maintaining a very small pressure of gaseous hydrogen in the heat exchanger. The experiment was initiated by opening ACV-16 with a fixed setting of AVV-23.

In the few cases where steady state operation was successfully accomplished, the system exhibited a tendency toward instability. When the pump flow rate was slightly perturbed, an undamped oscillatory instability of the heat exchanger flow rate and pressure was observed to occur. A simple theory was evolved to explain this behavior which both gave a reasonable value of the natural frequency of the system and suggested a method for circumventing the unstable operation of the system. It was assumed that the system behaved as a Helmholtz resonator, which is the acoustic analogue of a spring-mass system.² In this system liquid between the pump and the heat exchanger inlet formed the inertial element or mass and the hydrogen gas in the heat exchanger formed the

analogue of a spring. A simple analysis of the system was performed which assumes that the liquid is homogenous and incompressible with the density of liquid hydrogen, and that the gas is homogenous and ideal. These assumptions coupled with the equation of continuity and the Bernoulli equation leads to a natural oscillatory frequency expressed by the following relation:

$$f_n = \frac{A}{2\pi} \sqrt{\frac{\gamma P_g}{m V_g}}$$

where

γ = the ratio of specific heats for hydrogen

P = the pressure in the heat exchanger, lb/ft²

A = cross section area of pipe where filled with liquid, ft²

m = mass of liquid, lb

V_g = volume of gas, ft³

This simple formula applied at a typical operating point yielded an oscillatory frequency of 1.0 cps. This compares with an observed natural frequency of 0.2 cps. This agreement was regarded as adequate, considering the crude assumptions made in the model, to confirm the basic spring-mass model.

Proceeding from the model suggested above, it was obvious that one possible method of avoiding the oscillations observed was to decouple the spring and mass elements of the system. This was easily

done by providing a pressure drop in the system across the throttle valve, ACV-16. It was found that the heat exchanger could be operated smoothly on startup and at steady state by providing, during all phases of the operation, at least a 50 psi pressure drop across the throttle valve. A sequence of operation which was found successful was to establish a pressure of 150 psi in the heat exchanger through the gas control valve, PCV-89 and a pump discharge pressure of about 350 psi with ACV-16 closed and the liquid hydrogen flowing through the pump by-pass valve. Liquid flow into the test assembly was programmed by slowly opening the throttle valve. The gas control valve, PCV-89, controlled the pressure in the heat exchanger assembly. As liquid was added, this valve automatically closed allowing liquid to be gradually introduced into the warm parts of the system. When the pressure due to liquid flow exceeded the demanded gas pressure, the gas control valve completely closed; and flow to the heat exchanger at this point was entirely liquid. From this point it was a simple matter to program the pump speed in order to increase the heat exchanger flow rate to any desired value within its operating range. So long as the throttle valve was maintained at a position which provided at least a 50 psi pressure drop, no oscillatory behavior was observed. It should also be observed that the heat exchanger was operated with no apparent difficulty at pressures below the critical pressure in the heat exchanger.

One section of the experimental device had a twisted aluminum ribbon inserted in it in order to increase the local fluid velocity

at the wall of the inner pipe. The effect of the turbulator at different temperature levels was evaluated by connecting the heat exchanger first with the turbulator in the upstream section and later in the downstream section. The difference in its effects for these two configurations was observed to be small; it seemed to be equally effective in increasing the heat transfer in the cold or the relatively warm sections.

Experimental Results

Detailed heat transfer results for the prototype test have been reported elsewhere.³ In summary, these results showed the following;

- 1) In the leg without a swirl inducer, heat transfer coefficients to hydrogen averaged 60% of the Dittus-Boelter value.
- 2) With a swirl inducer present, coefficients approached those predicted by Dittus-Boelter.
- 3) Fin efficiencies varied from 16% where there was no ice to 100% where the fins were entirely covered with ice. These results compare favorably with those obtained in the analog studies. A thorough discussion of this point can be found in reference 3.

A comparison of the experimentally observed axial temperature gradients with the predicted gradients is shown in Fig. 2 as previously described. Note that agreement is good in the leg containing the swirl inducer, where the experimental coefficients can be closely predicted by the correlating equation employed, i.e. the Dittus-Boelter equation. The decrease in heat transfer in the leg without the swirl inducer led to a decision to include a swirl inducer throughout the entire length

of the full-scale exchanger.

In one test, the water flow was shut off while maintaining full hydrogen flow. After 45 seconds, the temperature halfway across the annular water space was still above freezing. The danger of freezing shut the water passage is not great.

Stresses, predominantly thermal stresses, were measured at several points in the exchanger by means of strain gauges. Measured stresses were between 30,000-40,000 psi at the inner surface of the hydrogen line. These were considered large but not excessive. To decrease them somewhat, the design for the full-scale exchanger was changed to larger I.D. and smaller fins on the inner pipe as mentioned previously.

Summary

The design, analysis, and successful field testing of a double-pipe heat exchanger in a unique application has been described. The exchanger consists of a 4-in. I.D. inner line inside a 10-in. I.D. outer pipe, the unit being 160 ft long. Hot (200°F) water flowing in the annulus provides energy to heat 12½ lb/sec of liquid hydrogen to 450°R.

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1. Williamson, K. D., and Bartlit, J. R., in Advances in Cryogenic Engineering, V. 10, Plenum Press (1965), p. 375.
2. Thurston, R. S., in Advances in Cryogenic Engineering, V. 10, Plenum Press, New York (1965), p. 305.
3. Bartlit, J. R., and Williamson, K. D., Jr., Paper H-6 presented at the 1965 Cryogenic Engineering Conference, August 23-25, Houston, Texas.

SCHEMATIC DIAGRAM OF FULL-SCALE HEAT EXCHANGER SHOWING INSTRUMENTATION

Fig. 1 Schematic Diagram of Full-Scale Prototype Heat Exchanger showing Instrumentation

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Fig. 2 Temperature Profiles for Full-Scale Prototype Heat Exchanger

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Fig. 3 Typical Temperature Profile
through Fin and Tube Wall in
Prototype Heat Exchanger

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Fig. 4 Predicted Temperature Profiles
for Full-Scale Heat Exchanger

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Fig. 5 Water and Hydrogen Outlet Temperature as a Function of Hydrogen Flowrate for the Full-Scale Heat Exchanger (Water Flowrate of 187 lb/sec)

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Fig. 6 Schematic Diagram of the Flow System used in Testing the Full-Scale Prototype Heat Exchanger

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