NAA-SR-4382
REACTORS-POWER
35 PAGES

HNPF COLD TRAP EVALUATION

By

R. CYGAN

ATOMICS INTERNATIONAL

A DIVISION OF NORTH AMERICAN AVIATION, INC.
P.O. BOX 309 CANOGA PARK, CALIFORNIA

CONTRACT: AT(11-1)-GEN-8
ISSUED: DECEMBER 15, 1959

DISTRIBUTION

This report has been distributed according to the category "Reactors-Power" as given in "Standard Distribution Lists for Unclassified Scientific and Technical Reports" TID-4500 (15th Ed.), August 1, 1959. A total of 620 copies was printed.

CONTENTS

		Page
	Abstract	v
I.	Introduction	1
II.	Background	2
III.	Summary	3
IV.	Description of Test Units	3
	A. No. 1 Design	3
	B. No. 2 Design	4
V.	Apparatus	6
VI.	Procedure	8
VII.	Results	10
	A. Cold Trapping	10
	B. Pressure Drop	10
	C. Thermal Performance	11
VIII.	Recommendations and Conclusions	13
	Appendix - Analysis of Heat Transfer Data	14
	A. Analysis of No. 1 Cold Trap Design	14
	B. Analysis of No. 2 Cold Trap Design	21
	C. Summary	26
	References	28

FIGURES

1.	No. 1 Design Cold Trap	3
2.	No. 2 Design Cold Trap	4
3.	Schematic of Cold Trap Test Loop	6
4.	Pressure Drop Variation With Oxide Quantity No. 1 Design	10
5.	Pressure Drop Variation With Oxide Quantity No. 2 Design	11
6.	No. 1 Design Predicted Temperature Profile for Cleanup Operation	14
7.	No. 1 Design Measured Temperature Profile During Cleanup Operation	15

FIGURES

		Page
8.	No. 1 Design Predicted Temperature Profile for Normal Operation	17
9.	No. 1 Design Measured Temperature Profile During Normal Operation	19
10.	No. 2 Design Predicted Temperature Profile for Cleanup Operation	21
11.	No. 2 Design Measured Temperature Profile During Cleanup Operation	22
12.	No. 2 Design Predicted Temperature Profile for Normal Operation	23
13.	No. 2 Design Measured Temperature Profile During Normal Operation	25

ABSTRACT

Two designs of sodium cold traps for the HNPF have been subjected to full scale tests. Performance features that were investigated included: oxide removal efficiency, oxide capacity, pressure drop characteristics, economizer effectiveness, and temperature profiles. Results indicate that both designs should perform satisfactorily in the Hallam plant.

I. INTRODUCTION

The Hallam Nuclear Power Facility (HNPF) has been authorized by the United States Atomic Energy Commission as part of the program to demonstrate the technical and economic feasibility of using nuclear reactors for central station power. The HNPF consists of a 240-Mwt Sodium Graphite Reactor (SGR) and the associated equipment for generating electrical power. The use of the SGR concept for this facility is intended to utilize and extend the knowledge gained from the operation of the Sodium Reactor Experiment, which Atomics International built and has operated for the AEC since July 12, 1957.

The HNPF will be located in southeastern Nebraska at the Sheldon Station of the Consumers' Public Power District (CPPD).

The use of sodium as a reactor coolant requires that attention be given to the effects of impurities in the sodium and to means for their removal. The presence of impurities in sodium may accelerate corrosion or impair thermal and/or mechanical performance. Of the several impurities present in sodium, our concern in cold trapping is chiefly with the oxygen, usually present as Na_2O. The removal of this oxide can be effected by utilizing the difference in solubility of the oxide in sodium at different temperatures. Since the solubility decreases with lower temperature, the presence of a lower temperature region in a sodium system automatically performs cold trapping. A specific area is set aside or added to a system to control this cold-trapping action. The cold trap that was investigated during this program was the circulating type. In this configuration, a portion of the total sodium stream is cooled while passing through the cold trap and then returned to the main stream. The sodium is usually cooled by an external coolant, Tetralin in this case. If any heat recovery system is utilized, the cold trap is said to have an economizer.

The operation of a circulating trap depends on the feed stream temperature being higher than the lowest temperature in the cold trap. As the temperature of the feed is lowered passing through the unit, a supersaturated solution of oxide results (momentarily) which in turn causes precipitation of the oxide in the cold trap. Depending on the unit efficiency, the effluent stream will have an oxide content approaching the saturation concentration at the minimum temperature encountered in the cold trap.

Although the cold trapping ability of a given unit is quite predictable, its oxide capacity (together with the pressure drop at capacity) has not been predictable. The prime purpose of this testing was to determine whether the specified trap designs were capable of cold trapping out 200 lb of Na_2O before the pressure drop across the unit reached 20 psig.

II. BACKGROUND

The experience with the circulating cold traps installed in the SRE[1] indicated several areas of potential difficulties if the same design was utilized for HNPF. One area which resulted in operating problems was the cold trap economizer. This heat exchanger is placed in the sodium stream ahead of the cold trap to precool the stream into the unit and reheat the effluent. If the feed is saturated with sodium oxide (Na_2O) at the inlet temperature to the economizer, precipitation of oxide will occur immediately after any cooling takes place. This results in deposition of oxide in the narrow annular spaces of the economizer with the resultant rapid plugging of these passages.

The other area of major difficulty was the nonuniform deposition of oxide within the cold trap unit itself. This caused the pressure drop across the cold trap to increase so rapidly that the unit was not useful long before the design quantity of oxide had been deposited inside the unit.

To minimize plugging problems encountered in the economizer, the HNPF designs eliminated narrow passages in this section. As an added simplification, it was found most convenient to incorporate the economizer inside the cold trap shell. Where a relatively narrow section is required, only sodium that has been purified by cold trapping passes through the section.

To increase the oxide capacity of the cold trap unit, two modifications in design were made. The first was an extension of the approach used in the economizer section, i.e., avoidance of small sodium passages or spaces in all areas where oxide saturated sodium flowed. The second was to substitute a forced convection, cooled heat removal system for the boiling liquid interface in a vertical jacket that formerly had been used.

III. SUMMARY

Each of the designs tested demonstrated adequate cold-trapping performance for the required operating conditions. The latter were sodium flow of 40 gpm at 350°F, and 10 gpm at a temperature of 607°F. The final pressure drop across the unit after adding approximately 200 lb of oxide (Na_2O) was less than 7 psi in each case. The indicated cold-trapping efficiency was 90% or better in all cases. The thermal economizer performance exceeded predictions in all tests. Methods for calculating heat transfer coefficients for the designs under consideration are given in the Appendix.

IV. DESCRIPTION OF TEST UNITS

A. No. 1 DESIGN

This arrangement (Figure 1) consists of a 20 in. diam trap section 9 ft long surrounded by a cooling jacket over the lower 8 ft. The lower 6-ft section of the trap portion is packed with woven stainless steel wire mesh (24 lb/ft^3 density).

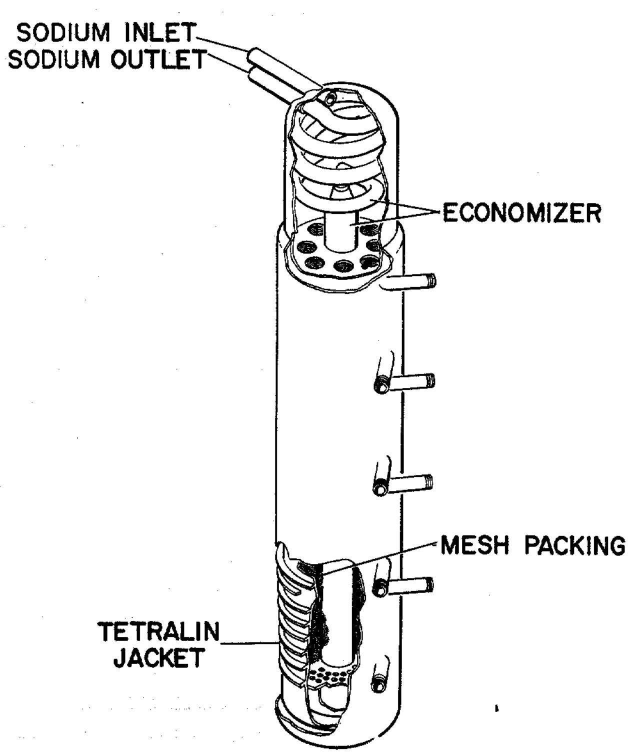

Figure 1. No. 1 Design Cold Trap

The internal economizer consists of a 6 in. diam tube 6-1/2 ft long with an 18-ft coil (3-1/2 turns) of 2-3/8 in. tubing at the upper end.

Sodium flow through the cold trap is as follows: The feed enters the unpacked dome section and gets its initial cooling from the effluent in the economizer coil. After passing the coil, it enters the mesh-packed region which is surrounded by the cooling jacket. Disk and doughnut baffles placed in the mesh force the sodium to flow in a longer path and prevent straight line channeling, which would permit bypassing portions of the packing. After passing downward through the entire packed column, the flow is reversed and enters the lower end of the thermal economizer pipe. After some warming of the sodium, the fluid enters the coiled portion of the thermal economizer where the major portion of the reheating is accomplished. The sodium leaves the cold trap from the exit of the economizer section.

Heat rejection from the cold trap is through coolant (Tetralin) circulated in the cooling jacket. Although the jacket in this unit was designed to permit Tetralin circulation in four separate circuits, the seal between channels was not tight enough to permit such a system of coolant control. As a result, all four circuits were operated in series throughout the testing. Spiral flow dividers spaced through the entire jacket prevented short circuiting of flow from the entrance to the outlet pipe. Tetralin coolant enters the jacket at the bottom (the coldest sodium region) and flows upward.

B. No. 2 DESIGN

This arrangement (Figure 2) differs from the No. 1 type chiefly in the design of the heat rejection system and the sodium economizer. The shell is a closed 22-in. diam cylinder about 8 ft long. An inner 20-in. diam shell (open at one end) serves as the sodium economizer (thermal). This inner shell also contains the mesh packing and the Tetralin cooling coil. The latter is an 18-in. diam helix formed using 2-3/8-in. OD tubing, 94 ft long.

Sodium flows first into the inner shell at the top of the trap. It passes downward through the mesh packing as it is cooled by the Tetralin coil. After leaving the mesh, the direction is reversed and the liquid metal flows upward through the annulus between the inner and outer shells.

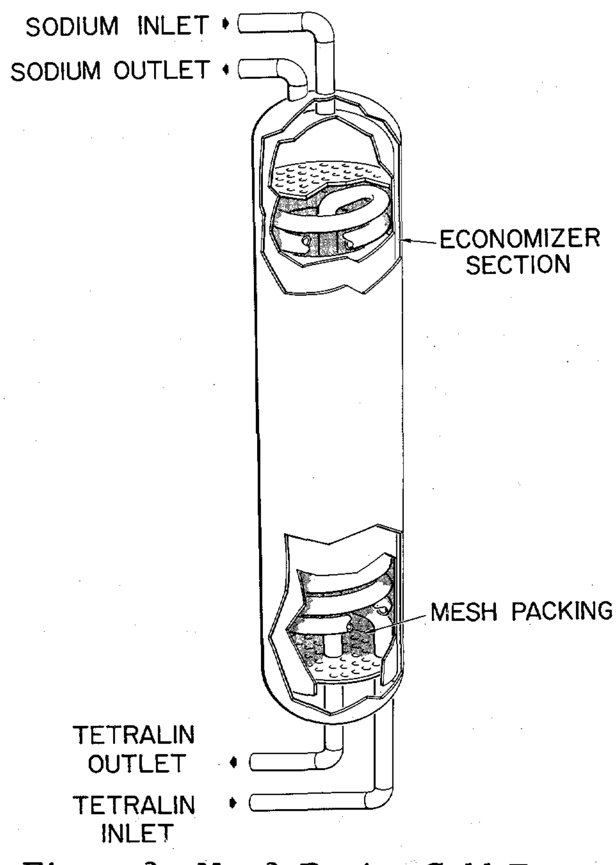

Figure 2. No. 2 Design Cold Trap

The economizer action (thermal) takes place in this section of the cold trap. The Tetralin enters the bottom of the helix and leaves through the central return tube.

Both designs were required to perform under the following two operating conditions: cleanup operation occurs when the sodium flowrate is 40 gpm with the inlet temperature 350°F. Normal operation occurs when the sodium flowrate is 10 gpm with the inlet temperature 607°F. The minimum sodium temperature for both operations is 250°F. The overall size of these units was determined primarily by the heat transfer requirements. The minimum size in any case must contain at least four times the volume of the oxide to be trapped; 200 lb in this design. The final requirement was that the pressure drop across the unit must not exceed 20 psig before 200 lb of oxide have been deposited in the unit.

V. APPARATUS

The loop used in conducting the cold trap tests is shown schematically in Figure 3. The primary source of heat was a 600,000 Btu/hr gas fired sodium furnace. Hot sodium leaving the furnace passed through the supply tank where

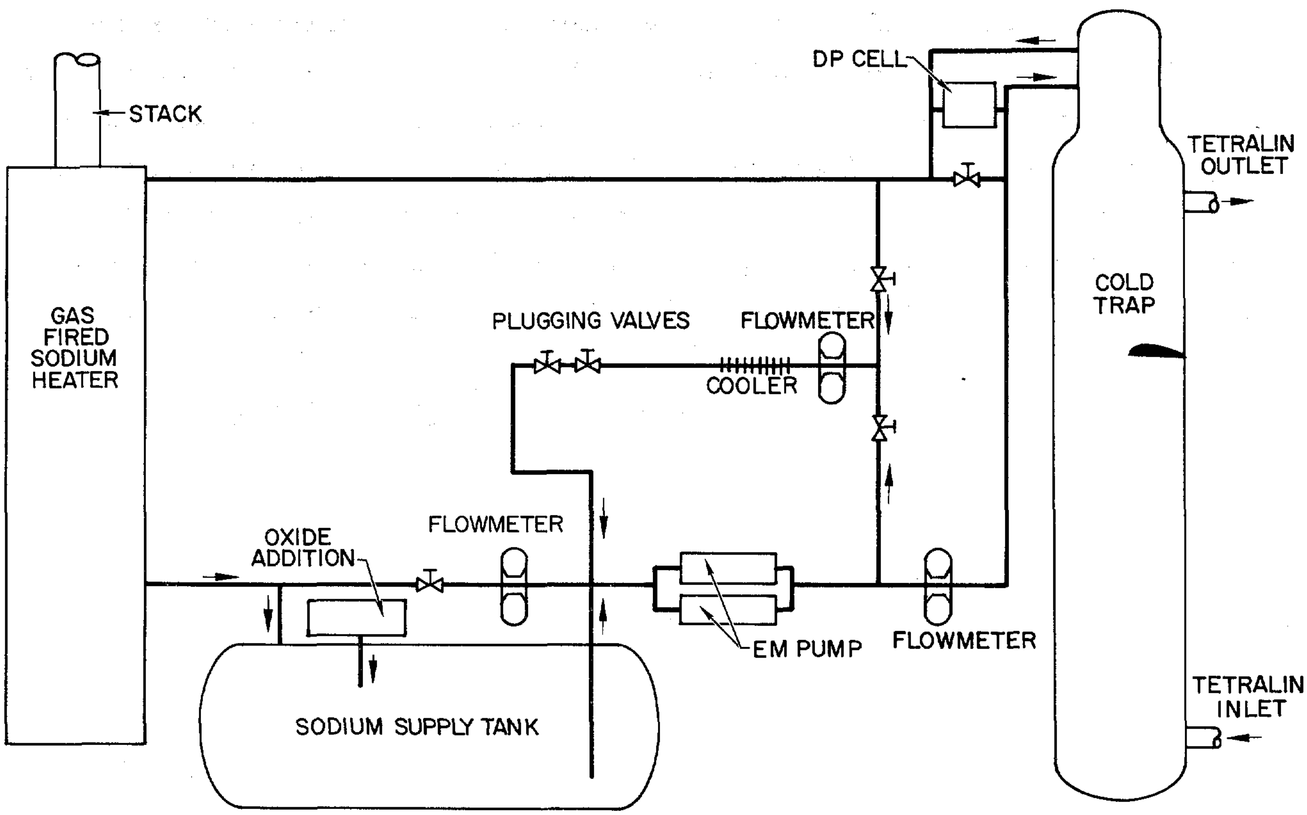

Figure 3. Schematic of Cold Trap Test Loop

sodium oxide was added to the liquid sodium. The oxide containing sodium stream was then pumped by a pair of electromagnetic (ac conduction) pumps to the inlet of the cold trap. The cold trap effluent was then returned to the furnace for reheating.

Oxide determinations in the system were made using the plugging indicator. Valving in this auxiliary system was so arranged that checks of either the feed or effluent cold trap stream could be made. Forced convection air cooling in a finned pipe section was used for heat rejection in the plugging indicator circuit. Valves with different plugging orifice sizes were used to cover the full range of plugging temperatures to be measured in the system.

Two types of oxide addition systems were used during the course of the test program. The first system employed a sealed mechanical screw feed which metered granular commercial sodium monoxide (Na_2O) from a hopper, maintained under an inert gas atmosphere, into the sodium supply tank. The second more successful apparatus consisted of a small gas pipe situated close to a turbulent sodium surface through which gaseous oxygen was admitted. A modification of this same system was subsequently added. The latter consisted of a small premix tank where oxygen gas addition to sodium took place before entering the supply tank.

Tetralin coolant was utilized to reject heat from the cold trap unit. Heat was removed from the Tetralin through Tetralin to water heat exchangers. The water was cooled in a forced draft cooling tower. To assure proper temperature control of Tetralin coolant at all outdoor temperatures, an immersion heater was installed in the Tetralin inlet to the cold trap jacket.

The sodium supply tank was constructed with several special features for the oxide addition process. One was a swirl chamber where the entering sodium stream enters a dish to create an area of relative turbulence in order to obtain good mixing. The bottom half of the tank contained a series of five baffles backed up with woven wire mesh filters. These were utilized to insure that only dissolved Na_2O was being removed by the sodium stream leaving the tank.

The entire test loop and the cold traps were fabricated from low carbon steel. The only austenitic stainless components were the mesh filters. Ordinary packed stem valves were used throughout with the exception of the oxide indicator system which contained bellows-sealed stem needle valves. All sodium piping was welded or ring-joint flanged.

VI. PROCEDURE

After leak checking the completed system, the supply tank was filled with about 370 gal of sodium. During initial operation, the cold trap was bypassed in order to establish the quantity of sodium oxide present in the system before any had been added intentionally. After a number of reproducible plugging runs were obtained, the oxide saturation temperature was established at 370°F. When the cold trap unit was placed on stream, the additional surface contamination raised the system oxide saturation temperature to 475°F.

Complete sets of temperature and flow data were taken before oxide addition was started in order to establish cold trap performance in the "clean" condition. The runs were made at the two expected operating conditions; namely, (a) sodium flow 39 gpm, 350°F sodium inlet temperature, 250°F minimum cold trap temperature, and 125°F Tetralin inlet temperature, and, (b) sodium flow 10 gpm, 607°F sodium inlet temperature, 250°F minimum cold trap temperature, and 125°F Tetralin inlet temperature. The thermal performance at these conditions is given in Figures 6 through 13. Similar data were taken as the test progressed to observe any effects as the quantity of oxide in the cold trap increased.

The first method of oxide addition was mechanical. The addition apparatus consisted of a variable speed screw feeder fed from a closed hopper. The feed material was commercial (95% Na_2O, balance Na) sodium oxide (granular, 40 mesh). The calibrated screw feed rate was adjusted to add the theoretical amount of oxide required to saturate the sodium passing through the supply tank. From all plugging meter tests during this phase of operation, it was clear that the rate of solution into the sodium was considerably slower than the oxide feed rate. This resulted in the cold trap feed stream being unsaturated at the testing temperature. With all available operating variations, it was not possible to obtain the desired oxide saturation temperature with this method of oxide addition. It appears that the fairly large particle size combined with the rather low temperature prevented the desired rate of solution.

To improve the solution rate, addition of gaseous oxygen close to a turbulent surface of sodium was used. The oxygen inlet nozzle was located about 1/2 in. above the sodium surface. With oxygen feed rates as high as 3 cfh, the rate of sodium oxide formation and its subsequent solution in the sodium was observed (from plugging indicator tests) to be practically instantaneous.

To assure saturation at the desired temperature, a slight excess of the theoretical amount of oxygen was added. Over an extended period, this excess of oxygen showed up as precipitated sodium oxide in two areas of the sodium supply tank. One of these was the unwet portion of the supply tank above the sodium. The oxide in this area was in the form of fluffy white snow varying from 1/4 to 3/4 in. in thickness. The remainder of the excess oxide had settled to the bottom of the tank as a sludge. The major portion of the latter had collected in the area ahead of the first mesh filter section with a small amount ahead of the second filter section. The quantity of oxide in each area was estimated in order to get a better value for the amount of oxide deposited in the test cold trap.

VII. RESULTS

A. COLD TRAPPING

Both designs have demonstrated the ability to continuously remove oxide from the stream at the rates of 0.26 and 0.72 lb of Na_2O per hour, the former during normal operations and the latter during cleanup operations. The approximate cold trapping efficiency as measured by a plugging meter was very nearly 100% at both trapping rates. The cold-trapping efficiency as used herein is defined as the following ratio:

$$\frac{\text{inlet stream oxide concentration} - \text{outlet stream oxide concentration}}{\text{inlet stream oxide concentration} - \text{saturation concentration at minimum cold trap temperature}}$$

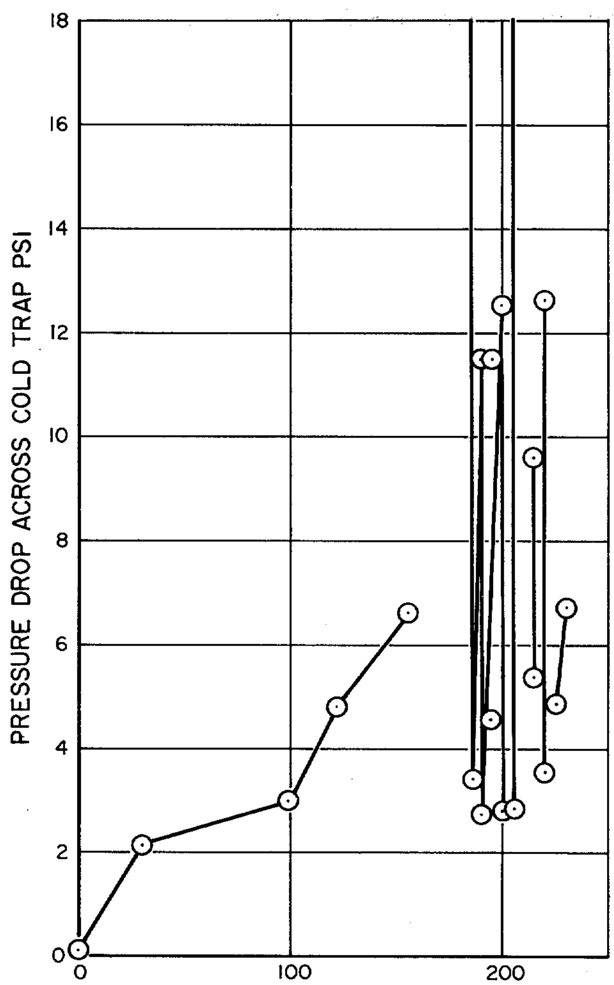

Figure 4. Pressure Drop Variation With Oxide Quantity No. 1 Design

Although repeatable plugging runs were quite easy to obtain on the feed stream, the effluent plugging checks were considerably more difficult to obtain. The latter can be attributed to the small total oxide content of the purified sodium stream. The cold-trapping efficiency was not significantly affected by the quantity of oxide present in the cold trap (up to 200 lb which was the test limit). The sodium residence times for each trap were: 3 and 12 min for No. 1 design, and 2 and 8 min for the No. 2 design.

B. PRESSURE DROP

The variation of pressure drop across the cold trap, as a function of oxide quantity, is shown for each trap in Figures 4 and 5. Until about 100 lb of oxide were added to the trap, the increase in pressure drop was moderate and orderly. With more than this quantity of oxide in the trap, the pressure drop across the

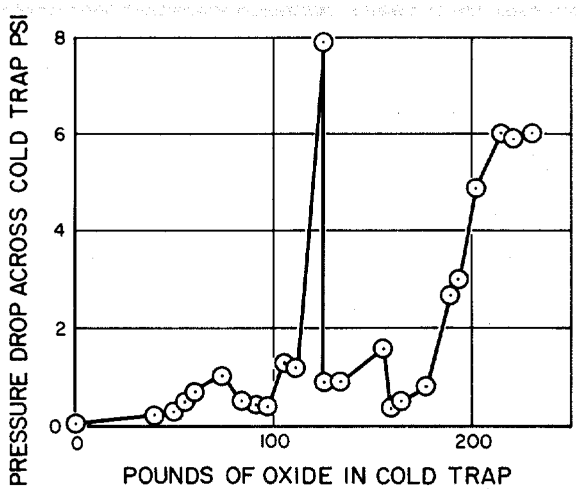

Figure 5. Pressure Drop Variation With Oxide Quantity No. 2 Design

cold trap increased rapidly. However, at the same time, the pressure drop across the external system also increased markedly. As a result, even though the pressure drop across the cold trap did not exceed pump capacity, the external system pressure drop increase combined with the former exceeded pump capacity and rated flow could not be maintained. Shutting off the oxygen feed (redistribution operation) resulted in a reduction in pressure drop throughout the system. This mode of operation is represented by the vertical lines at constant oxide quantity. The presence of a single such line in Figure 5 does not mean that other such operating points, as these in Figure 4, did not occur. Instead it is the result of an accelerated testing schedule during which complete data was recorded on a fixed schedule during each operating shift. As a result, the pressure drop reading recorded might be in any portion of the oxygen addition cycle. The final pressure drops are remarkably similar considering the design variations between the two types: 6.7 psi for design No. 1, and 6.0 for design No. 2.

C. THERMAL PERFORMANCE

1. No. 1 Design

The measured economizer performance of the sodium circuit exceeded the theoretical predictions. For cleanup phase operation, the economizer effectiveness was 50% compared to a predicted 27%. For normal operation, the comparison was 62% predicted and 79% actual. Economizer effectiveness is defined as follows:

$$\text{Effectiveness} = \frac{\text{sodium outlet temperature minus minimum sodium temperature}}{\text{sodium inlet temperature minus minimum sodium temperature}}$$

The change in economizer performance due to the presence of 200 lb of oxide in the cold trap is small; a decrease of 4% for the cleanup phase of operation.

The coolant circuit was somewhat oversized since considerable throttling of flow was necessary to obtain the required operating conditions. With the specified Tetralin inlet temperature of 125°F, the flow had to be reduced from the calculated 49 gpm to 10 gpm during cleanup phase. As a result, the Tetralin outlet temperature was 258° instead of 166°F. Similarly for normal operation, flow had to be reduced from 7.5 gpm to 1.8 gpm with the resultant outlet temperature of 350°F instead of 241°F.

2. No. 2 Design

The actual performance of the economizer exceeded that predicted, for both modes of operation. For cleanup phase operation, the economizer effectiveness was predicted to be 22% compared to the actual value of 37%. For normal operation, the comparison was 47% predicted and 74% actual. As contrasted to design No. 1, there was a more significant change in thermal performance caused by the addition of the 200 lb of sodium oxide. There was a decrease of 70% in the economizer effectiveness for conditions approaching the cleanup phase operation.

As was true in design No. 1, the Tetralin flow rate had to be reduced from the predicted 22 gpm to 15 gpm for cleanup operation and from 5.5 gpm to 2.5 gpm for normal operation.

VIII. RECOMMENDATIONS AND CONCLUSIONS

Both designs will remove sodium oxide from a sodium system at both required operating conditions (cleanup and normal), with efficiencies of 90% or better. The oxide capacity of both designs is over 100 lb. No operating problems are indicated for additions of approximately 100 lb. An additional 100 to 150 lb may be added to the cold trap with a moderate final pressure drop across the unit (less than 7 psi), if the oxide, in excess of 100 lb, is not added continuously. The economizer effectiveness of the No. 1 design is better than that of the No. 2 design. In addition, there is considerably less deterioration in the economizer performance of the No. 1 design compared to the No. 2 design as the oxide content of the trap approaches the capacity of the unit. The somewhat simpler design features of the No. 2 type result in a lower fabrication cost. The choice for the Hallam plant probably will be decided by whether initial cost or thermal economy is the more important factor.

The analysis in the Appendix has been carried out in order to suggest methods of calculating the heat transfer coefficients in these units so that the performance of similar designs might be predicted somewhat better. From these results, the troublesome areas for analysis seem to be indicated. For the No. 1 type, the flow dividers acting as fins in the Tetralin cooling jacket combined with the somewhat low value of the shell side sodium heat transfer coefficient originally predicted, led to lower Tetralin flowrates than were expected. With the actual Tetralin flow rather low, problems of controlling the minimum cold trap temperature were encountered. In the No. 2 type, the low prediction for the shell-side sodium heat transfer coefficient probably led to the higher Tetralin flowrates predicted. The control problem resulting was of somewhat lesser magnitude. It is felt that the methods of analysis suggested in the Appendix might prove helpful in predicting the performance of cold trap units of the type that were successfully tested during this program.

APPENDIX
ANALYSIS OF HEAT TRANSFER DATA

A. ANALYSIS OF No. 1 COLD TRAP DESIGN

A comparison of the predicted and actual performance can be made from the graphs of Figures 6, 7, 8, and 9. In all cases, the performance exceeded predictions. The following analysis is an attempt to obtain better agreement between calculated and measured heat transfer coefficients. Two modes of operation (cleanup and normal) will be investigated. For each mode of operation, the heat transfer coefficients for each of three sections (cooling section, pre-economizer, and economizer) will be evaluated theoretically and compared to the measured values.

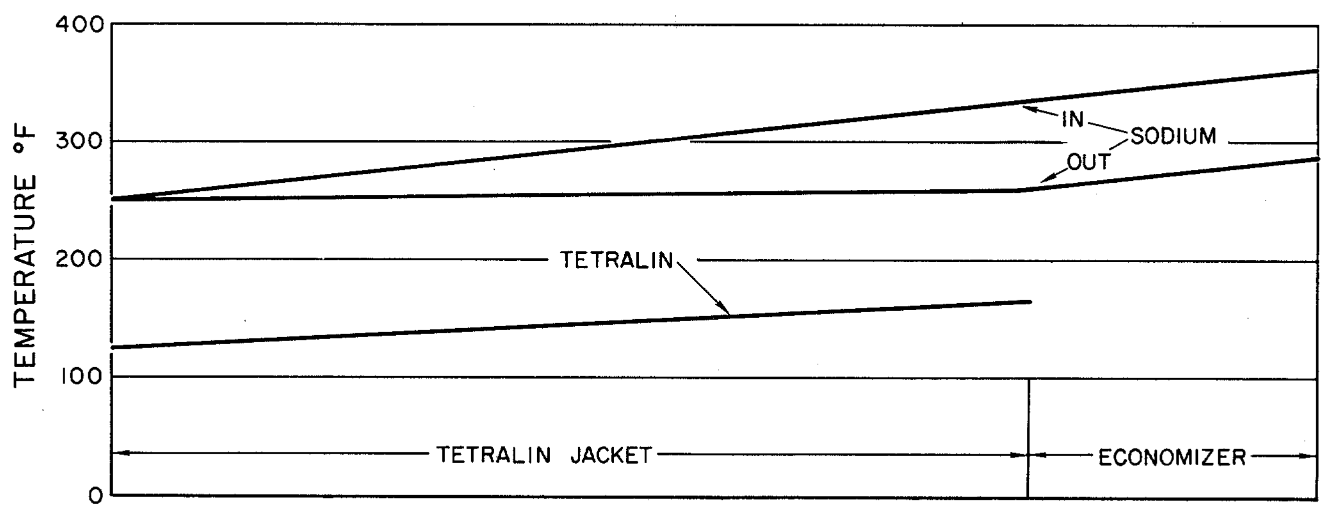

Figure 6. No. 1 Design Predicted Temperature Profile for Cleanup Operation

1. Cleanup Operation

The cleanup mode of operation will be analyzed first. During this operating phase, the following conditions are present:

Sodium flow: 39.9 gpm entering at 350°F,

Tetralin flow: 10.0 gpm entering at 125°F, and

Minimum sodium temperature: 250°F.

Temperatures for other sections of the cold trap, for the case under consideration, are given in Figure 7. For purposes of analysis, the cold trap will be

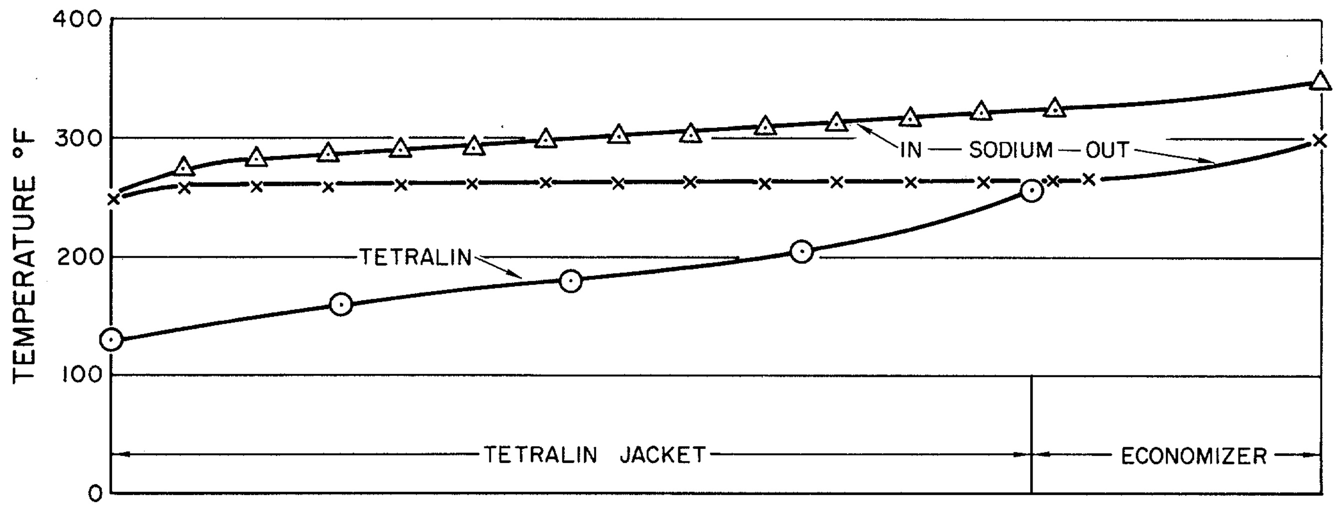

Figure 7. No. 1 Design Measured Temperature Profile During Cleanup Operation

divided into three sections. These are, the cooling section, the pre-economizer, and the economizer. The pre-economizer is the large diameter portion of the economizer shown in Figure 1.

a. Cooling Section

The cooling section is the region of maximum cold-trapping action. After being pre-cooled by the sodium-to-sodium economizer, the sodium temperature is further lowered by the surrounding Tetralin jacket with an assist from the pre-economizer.

The heat transfer coefficient for the sodium side of the cooling jacket may be calculated using the following modified Donohue equation[2] for baffled shells:

$$N_{Nu} = 0.19\left(D_e'\right)^{0.6}\left(\frac{D_t G_e c_p}{k}\right)^{0.6} \qquad \ldots(1)$$

Because the unit utilizes disk and doughnut baffles, a characteristic mass flow which combines the mass flows occurring in the various sections is used. This is calculated from the following equation developed by Short:[3]

$$G_e = G_h\left(\frac{L}{S}\right)^{0.5}\left(\frac{A_h}{A_s}\right)^{0.6}\left(\frac{D_t}{D_s}\right)^{0.86} + G_a\left(\frac{L}{S}\right)^{0.5}\left(\frac{A_a}{A_s}\right)^{0.5}\left(\frac{D_t}{D_s}\right)^{0.86}$$

...(2)

$$+ G_r\left(\frac{S}{L}\right)^{0.5}\left(\frac{A_h}{A_s}\right)^{0.1} 50\left(\frac{D_t}{D_s}\right)^{1.17}$$

Substituting and evaluating, a value of 51,300 is obtained for G_e. The stainless mesh packing which fills the entire region under consideration reduces the net flow area to 80%. Because of this, the characteristic mass flow must be increased. The resulting value for substitution in Equation (1) is 64,000. After substitution of proper values in Equation (1), the resulting Nusselt number is 19.3. The heat transfer coefficient for the sodium side of the cooling jacket = 1883. The conductance of the 1/2-in. thick carbon steel shell at the average temperature is 658. With these two values, the average film temperature on the Tetralin side can be calculated for use in the following equation:[4]

$$N_{Nu} = 0.027\left(N_{Re}\right)^{0.8}\left(N_{Pr}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$$ ...(3)

Evaluating after proper substitution gives a value of N_{Nu} = 156.8, and h = 96.2.

Because of the configuration of the Tetralin flow channel, the flow dividers and the outer shell will act as a fin. The latter will increase the area over which the Tetralin film coefficient is effective. The amount of the increase contributed by the fin is called the fin effectiveness and may be computed in the usual manner. Using (by extrapolation) Figure 1 of Reference 5, the fin effectiveness is 24%. Since the total fin area is 52.6 ft^2, the effective fin area is 12.62 ft^2. As a result, the effective heat transfer area for the Tetralin is 30.6 + 12.6 = 43.2 ft^2. As an aid in computing the overall U, the previously calculated coefficient for the Tetralin side can be converted to an effective h, which will take into account the new area. The value for this effective h becomes 135.7. Computing in the usual manner, the overall coefficient is 106.

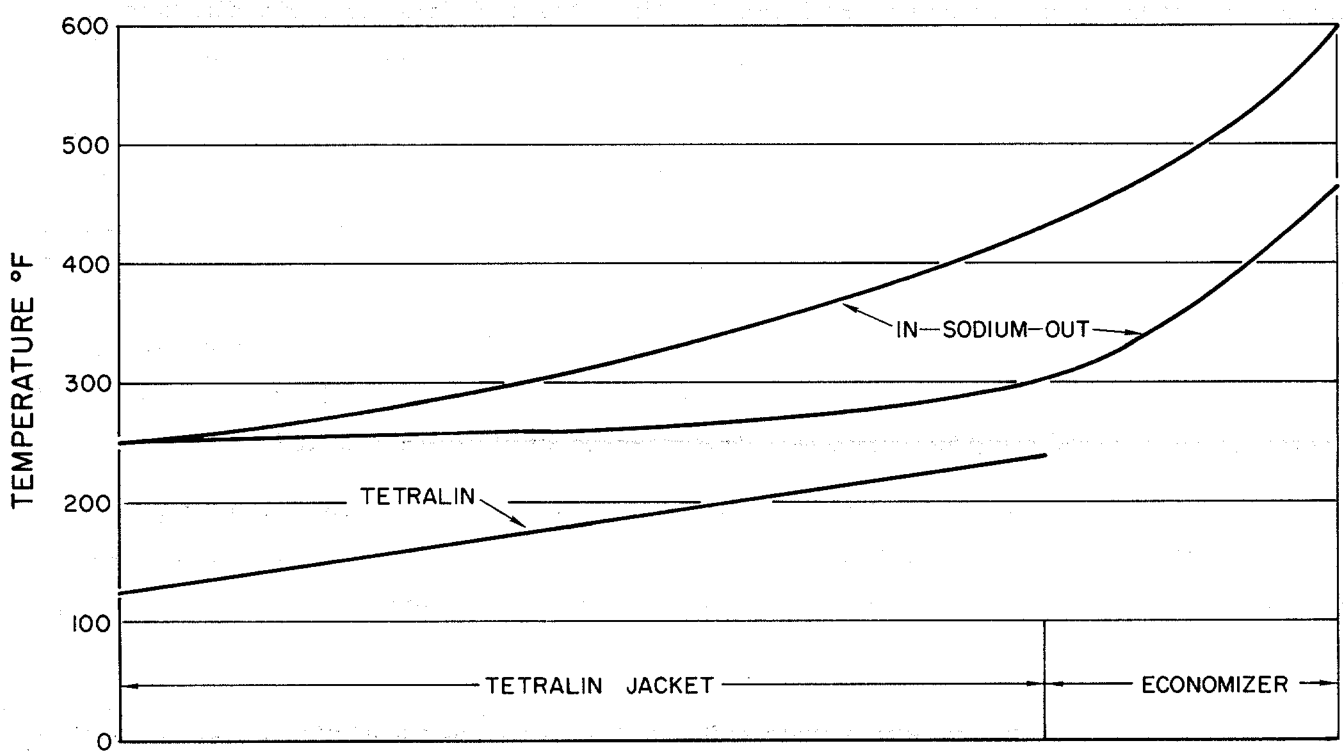

Figure 8. No. 1 Design Predicted Temperature Profile for Normal Operation

The overall coefficient based on the experimental data is calculated in the conventional manner. The net heat transferred is 289,000 Btu/hr through an area of 32.4 ft^2. The log mean temperature difference based on temperatures given in Figure 7 is 93°F. The resulting overall coefficient is 95.8.

It is fortunate that the theoretical U exceeds that obtained from experiment. Since it is known from observation that the contact between the Tetralin flow dividers and the sodium shell was imperfect (indicated by the flow bypassing observed), the theoretical U should be higher than the measured value. We now can introduce another factor called bond efficiency which will correct for the imperfect bond. This new factor will permit an additional lowering of fin effectiveness to bring the theoretical U in agreement with the measured U. The bond efficiency that results from this approach is 60%.

b. Pre-economizer Section

This is the region where the first reheating of the effluent takes place by the sodium in the cooling section. The outside heat transfer coefficient for this section has been calculated above as 1883. The inside film coefficient may be calculated using the following equation:[6]

$$N_{Nu} = 0.625\left(N_{Pe}\right)^{0.4} \qquad \text{...(4)}$$

Substituting and evaluating, the resulting value of h is 633. The value for the conductance of the steel tube wall is 2720. These separate values are combined in the conventional manner to obtain the overall coefficient as 402.

To compute the experimental overall coefficient, the following data is used: area = 9.84 ft^2, heat transferred = 102,000 Btu/hr, and log mean temperature difference = 28°F. The measured U resulting is 368.

c. Economizer Section

This is the region of major heat recovery which does the first cooling and concomitantly the last heating of the sodium passing through the cold trap. The economizer consists of a spiral coil of 2-3/8 in. tubing.

The heat transfer coefficient for the inside sodium film can be calculated using Equation (4). This operation gives a value of 2780. The thermal conductance of the tube wall is 1734. The outside coefficient may be computed using the following modified Donohue equation[2] for unbaffled shells:

$$N_{Nu} = 0.128\left(D_e'\right)^{0.6}\left(\frac{D_t G_e c_p}{k}\right)^{0.6} \qquad \text{...(5)}$$

Evaluating this equation after the proper substitutions gives an outside coefficient equal to 487. When these coefficients are combined in the conventional manner, the overall coefficient is 353.

For a heat transfer rate of 197,800 Btu/hr with a log mean temperature difference of 54°F through an area of 9.18 ft^2, the experimental U is 399.

2. Normal Operation

During this operating phase, the following conditions apply:

Sodium flow: 10.0 gpm entering at 600°F,

Tetralin flow: 1.86 gpm entering at 125°F, and

Minimum sodium temperature : 250°F.

Temperatures for other sections of the cold trap for the case under study are given in Figure 9. The same areas will be investigated as in the previous mode of operation.

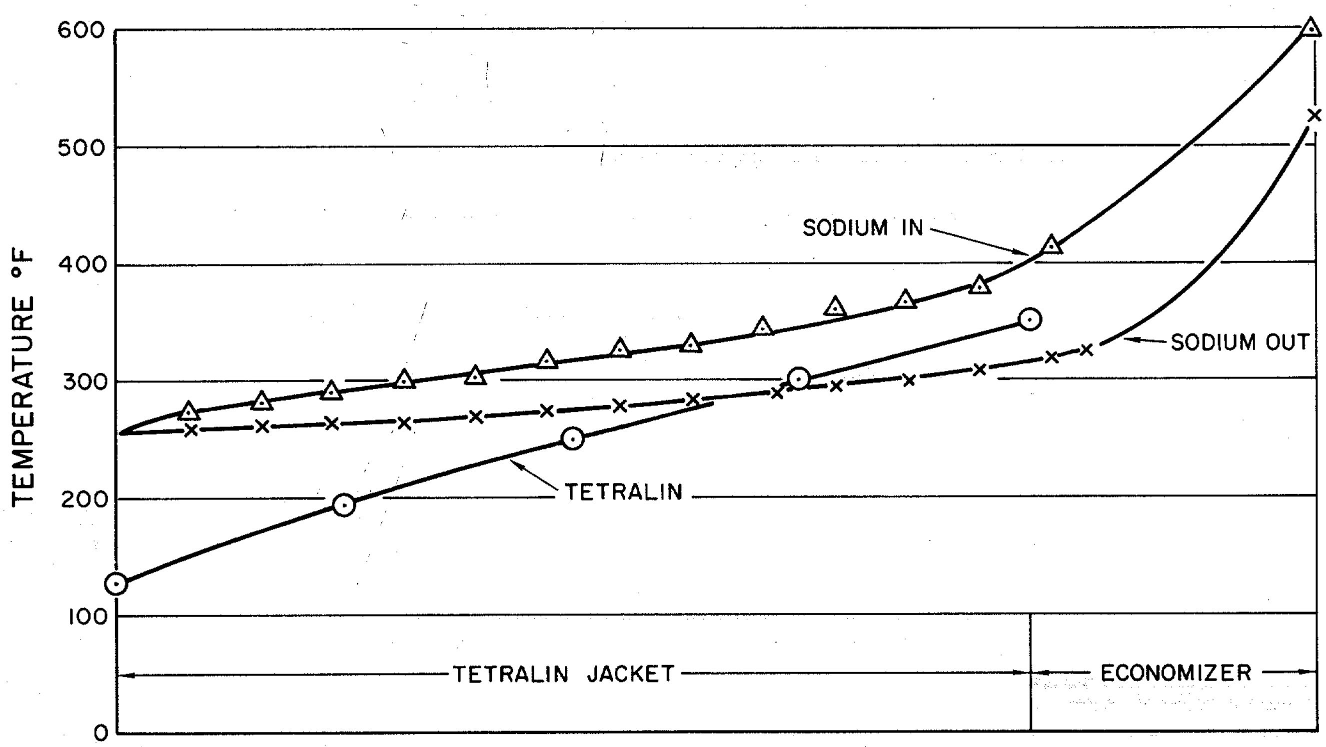

Figure 9. No. 1 Design Measured Temperature Profile During Normal Operation

a. Cooling Section

The film coefficient on the sodium side may be calculated using Equation (1). This gives a value of 792. The conductance of the steel shell is 648. The Tetralin film coefficient may be computed from the following equation:[7]

$$N_{St} = 0.0067 \left(N_{Pr}\right)^{-0.8} \left(\frac{\mu}{\mu_s}\right)^{0.14} \qquad \ldots(6)$$

This equation is chosen in preference to Equation (3) because of the relatively low Reynolds number for the Tetralin (3700). Substituting the appropriate values gives a film coefficient of 25.4. Adjusting this as before for the additional effective area of the fins gives an effective coefficient of 48.2. Combining these individual coefficients gives an overall coefficient of 42.7.

With a heat transfer rate of 100,200 Btu/hr at a log mean temperature difference of 84°F for an area of 32.4 ft^2, the overall experimental coefficient is 37.0.

As before, a bond efficiency can be computed. For this case, the value is 70% compared to 60% for cleanup operation. The higher value for normal operation might be accounted for by the higher temperature difference between the shell and the Tetralin jacket. This condition would tend to expand the shell more tightly against the flow dividers, thereby improving the thermal bond.

b. Pre-economizer Section

The tube outside coefficient has been calculated above as 792. The conductance of the steel tube wall is 2720. The inside sodium coefficient computed using Equation (4) has a value of 572. Combining these factors for the overall coefficient gives the result 226.

For an area of 9.84 ft^2 with a net heat transfer of 85,600 Btu/hr at a temperature difference of 44°F, the experimental overall heat transfer coefficient is 198.

c. Economizer Section

Computation of the tube inside film coefficient using Equation (4) yields a value of 1550. Conductance of the tubing wall is 1682. For the low flow rate in the area outside the tube, the Reynolds number is approximately 2000. For this low value, the better correlation is obtained using the forced convection in laminar flow approach. This can be done graphically (in part) by calculation of the factor ϕ by the following equation:[8]

$$\phi = \left(\frac{4 c_p G_e}{\pi k L} \right) \qquad \text{...(7)}$$

For the conditions of the experiment ϕ has a value of 13.7. Using the proper curve from Figure 22-7 of Reference 4, the Nusselt number given is 4.4. Solving this for the heat transfer coefficient gives a value of 1000. Combining this with the other factors results in an overall coefficient of 445.

With a log mean temperature difference of 79°F across an area of 9.18 ft^2 with a net heat flow of 296,000 Btu/hr, the overall heat transfer coefficient is 408.

B. ANALYSIS OF No. 2 COLD TRAP DESIGN

A comparison of the predicted and actual thermal performance can be made from the graphs of Figures 10, 11, 12, and 13. Once again two modes of operation (cleanup and normal) will be investigated. For each mode of operation, the overall heat transfer coefficients for each of two sections (cooling section and economizer) will be evaluated theoretically and compared to the experimental values.

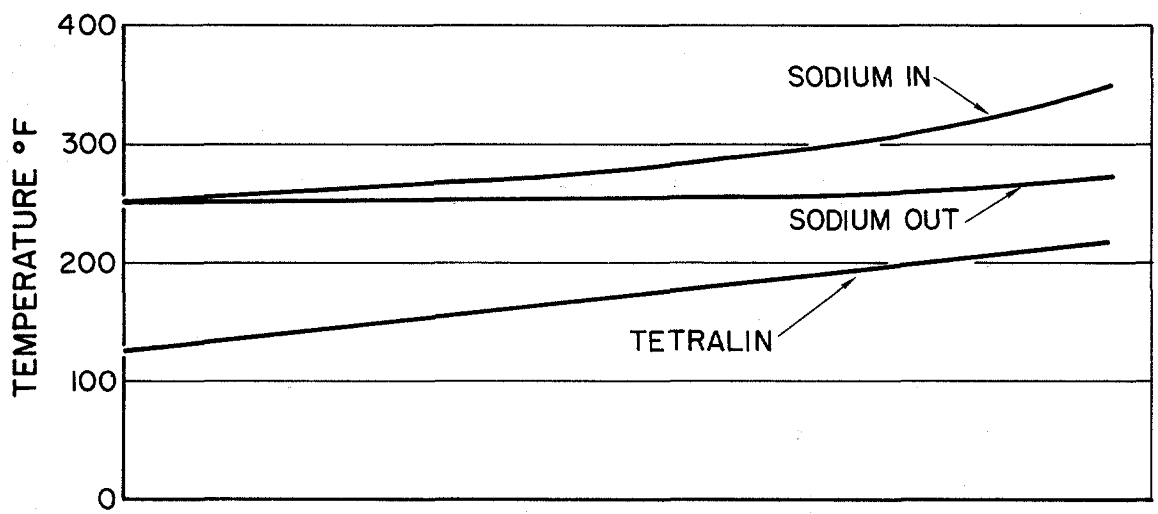

Figure 10. No. 2 Design Predicted Temperature Profile for Cleanup Operation

1. Cleanup Operation

During this operating phase, the following conditions exist:

Sodium flow: 17,900 lb/hr entering at 352°F,

Tetralin flow: 6,220 lb/hr entering at 115°F, and

Minimum sodium temperature: 246°F.

Temperatures for other sections of the cold trap for the case under consideration are given in Figure 11. For purposes of analysis, the cold trap will be divided into two sections. These are, the cooling section and the economizer section.

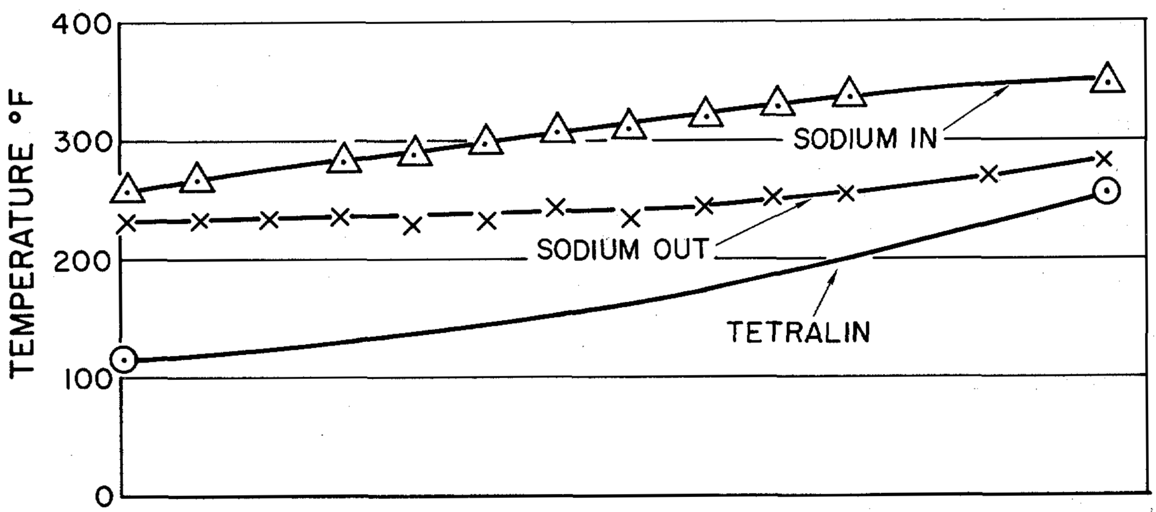

Figure 11. No. 2 Design Measured Temperature Profile During Cleanup Operation

a. Cooling Section

This consists of the mesh packed region which contains the Tetralin cooling coil of 2-3/8-in. tubing about 94 ft long. Although the greater part of the cooling in the region is done by the Tetralin coil, some heat is removed by the economizer that surrounds this mesh packed area.

The outside sodium coefficient may be calculated using Equation (4) with the equivalent diameter and flow area evaluated as described below. Since the Tetralin coil is an open spiral, the true flow area is taken as the geometric mean of shell area and the net shell area after subtracting the face area of the coil.

This results in a value of 1.58 ft^2. Further, since this volume is filled with mesh packing which has an effective free volume of 80%, the corrected mean flow area becomes 1.26 ft^2. This is the flow area that will be used in the flow calculations. The equivalent diameter for this configuration is taken as four times the flow area divided by the wetted perimeter. The wetted perimeter used in this calculation includes the inside diameter of the economizer shell, the outside diameter of the coil return pipe, and the mean diameter of the coil spiral. Using this approach, the calculated equivalent diameter becomes 0.498 ft. Substituting the various values in Equation (4) gives a value of 2.90 for the Nusselt number. Evaluating the latter for heat transfer coefficient results in an h = 280. The conductance of the tube wall at the average temperature is 1755. The Tetralin coefficient is calculated using Equation (3). This gives 127 for the value of the Tetralin film coefficient. When these factors are combined, the resulting overall heat transfer coefficient is 84.

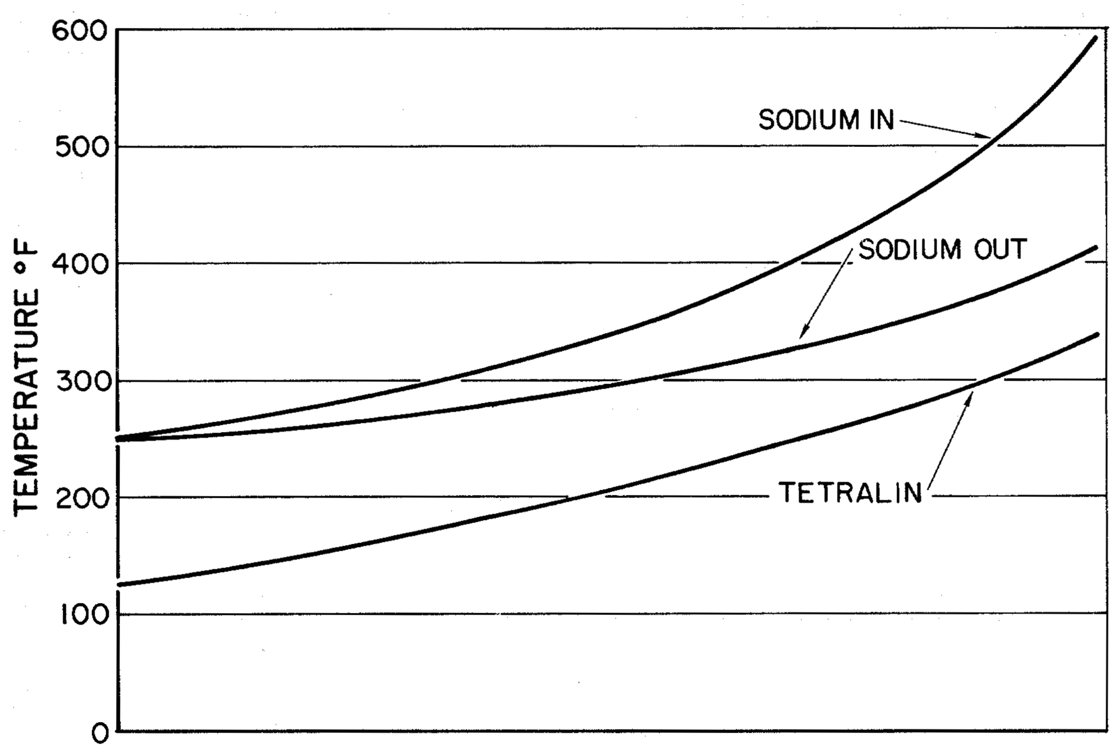

Figure 12. No. 2 Design Predicted Temperature Profile for Normal Operation

For a heat transfer rate of 439,000 Btu/hr across an area of 52.1 ft^2 at a log mean temperature difference of 121°F, the overall heat transfer coefficient is 70.

b. Economizer

This region consists of a single shell dividing the same sodium stream which flows in opposite directions on each side of the dividing shell. The sodium being reheated flows through the relatively thin annulus formed by the economizer shell and the containment vessel.

The shellside coefficient has been calculated above and has a value of 280. The conductance of the economizer shell is 1350. Since the annular space is small, the film coefficient may be computed using the following equation:[9]

$$N_{Nu} = 5.8 + 0.02\left(D_e V \frac{c_p}{k}\right)^{0.8} \qquad \text{...(8)}$$

After substitution and evaluation, this equation gives a Nusselt number of 5.816. The latter can be solved for the film coefficient to give 2270. When all the factors are combined, the resulting overall heat transfer coefficient is 210.

For a heat flow of 293,000 Btu/hr across a 34.9 ft^2 area at a log mean temperature difference of 42°F, the overall heat transfer coefficient is 200.

2. Normal Operation

During this operating phase, the following conditions are present:

Sodium flow: 4390 lb/hr entering at 607°F,

Tetralin flow: 930 lb/hr entering at 127°F, and

Minimum sodium temperature: 245°F.

Temperatures for sections of the cold trap for the case under consideration are given in Figure 13. The same areas will be investigated as in the previous mode of operation.

a. Cooling Section

The Reynolds number for the sodium flowing outside the Tetralin coil (using the same flow area and equivalent diameter as in section 1a) is 1240. For such a low value of Reynolds number, the heat transfer can better be evaluated

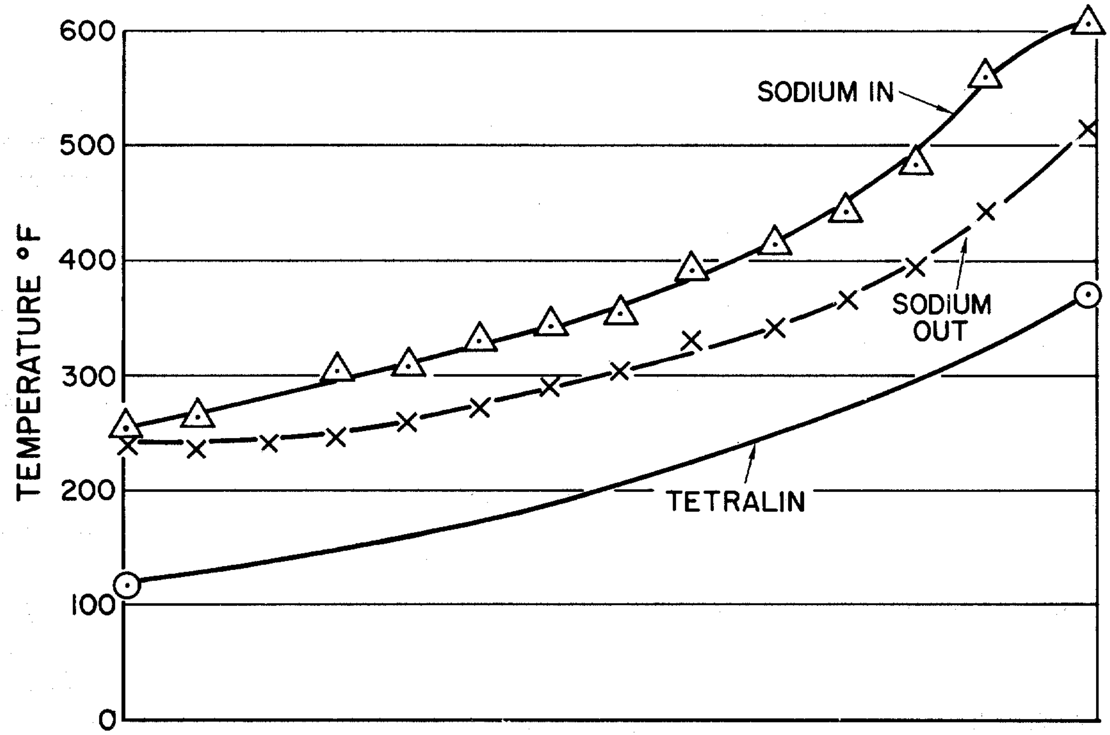

Figure 13. No. 2 Design Measured Temperature Profile During Normal Operation

if considered as forced convection in laminar flow. ϕ calculated from Equation (7) yields a value of 4.3. Because of the indeterminate geometry in this region, the Nusselt number chosen from Figure 22-7 of Reference 4 is that for a shape intermediate between a round and flat duct. The resulting Nusselt number is 5.9. Solving for h obtains a value of 557. The conductance of the tubing wall is 1700. The Reynolds number for the Tetralin flowing inside the coil is 2700. The Tetralin film coefficient for this condition may be evaluated using Figure 26-1 of Reference 4. By so doing, a value of 0.0025 for the following expression is obtained:

$$N_{St} \left(N_{Pr} \right)^{2/3} \left(\frac{\mu_s}{\mu} \right)^{0.14} \qquad \ldots(9)$$

Solving this for the Stanton number gives a value of 5.39×10^{-4}. This in turn can be solved for the film coefficient to give the result 12.2. Combining the various factors gives an overall heat transfer coefficient of 11.85.

For a log mean temperature difference of 180°F across an area of 52.1 ft^2 at a heat flow of 126,600 Btu/hr, the overall heat transfer coefficient is 13.37.

b. Economizer

The shellside coefficient calculated above is 557. The conductance of the steel economizer wall is 1280. The film coefficient for the annular space computed using Equation (8) is 2200. The resulting overall coefficient is 330.

For a heat flow of 394,000 Btu/hr across a 34.9 ft^2 area at a log mean temperature difference of 37°F, the measured overall heat transfer coefficient is 306.

C SUMMARY

It can be seen that the calculations of heat transfer coefficients in the complex flow pattern region of the shell offer considerable variation of analysis. It is agreed that although the methods described seem to give reasonable agreement between the experimental and theoretical values, other approaches could also be reasonably used. Although other theoretical methods (not cited here) were used and gave results in lesser agreement with measured values, still others are certainly available. The analysis herein gives agreement of ±20% for all cases. This was considered adequate and pursuit by more abstruse methods was not further considered.

Since the primary objective of the test was the evaluation of cold trapping performance and not heat transfer information, standard uncalibrated thermocouples were used. The standard error for the grade of thermocouples employed in the temperature range of this experiment is ±4°F. Using three measurements to obtain the average for a point gives a probable error of the mean of ±2 °F.

Since the temperature difference used in computing the overall heat transfer coefficient involves the difference between two temperature differences, the cumulative error at this point is ±8°F. For the temperature differences encountered in these measurements, the percent error from this source alone ranges from 10 to 27%. Therefore, without rigorous analysis, it will be assumed that the thermocouple error is the principal one in the results.

REFERENCES

1. R. B. Hinze, "Control of Oxygen Concentration in a Large Sodium System," NAA-SR-3638 (1959)

2. R. A. Tidball et al., "Final Report on the 1000 Kw Air Cooled Liquid Metal Heat Transfer Loop," MSA-TR-39

3. B. E. Short, "A Review of Heat-Transfer Coefficients and Friction Factors for Tubular Heat Exchangers," Trans. Am. Soc. Mech. Engrs. 64 (1942), p 779

4. M. Jakob, Heat Transfer, Vol I, (New York, J. Wiley, 1949) p 547

5. W. M. Kays and A. L. London, Compact Heat Exchangers, (Palo Alto, The National Press, 1955)

6. B. Lubarsky and S. J. Kaufman, "Review of Experimental Investigation of Liquid-Metal Heat Transfer," NACA-TN-3336

7. M. Jakob, Heat Transfer, Vol I, (New York, J. Wiley, 1949) p 550

8. M. Jakob, Heat Transfer, Vol I, (New York, J. Wiley, 1949) p 462

9. C. B. Jackson (ed.), "Liquid Metals Handbook-Sodium (NaK) Supplement," Atomic Energy Commission (July 1955)

NOMENCLATURE

A = area in ft^2

c_p = specific heat Btu/lb °F

D_e = equivalent diameter, ft

D_e = equivalent diameter, in.

D_t = diameter of tube, ft

D_s = diameter of shell, ft

G = mass flow lb/hr-ft^2

G_e = equivalent mass flow lb/hr-ft^2

k = thermal conductivity Btu/hr-°F-ft^2-ft

L = length of heat exchanger, ft

N_{Nu} = Nusselt number

N_{Pe} = Peclet number

N_{Pr} = Prandtl number

N_{Re} = Reynolds number

N_{St} = Stanton number

S = spacing between subsequent baffles

μ = viscosity lb/hr-ft at bulk temperature

μ_s = viscosity lb/hr-ft at wall film temperature

V = velocity of flow ft/hr

SUBSCRIPTS

a = referred to annular space of disk in disk and doughnut HX

h = referred to hole of doughnut of disk and doughnut HX

r = referred to area perpendicular to tubes, radially of disk and doughnut HX

s = referred to shell