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POTASSIUM CONDENSING TESTS OF HORIZONTAL MULTITUBE CONVECTIVE AND RADIATIVE CONDENSERS OPERATING AT VAPOR TEMPERATURES OF 1250° TO 1500° F

by O. A. Gutierrez, N.J. Sekas, L. W. Acker, and D. B. Fenn
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TECHNICAL PAPER proposed for presentation at Rankine Cycle Space Power System Specialists Conference sponsored by the American Institute of Aeronautics and Astronautics Cleveland, Ohio, October 26-28, 1965

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ABSTRACT

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Two types of horizontally oriented multitube vapor condensers, dissipating heat loads up to 150 KW thermal, were tested over a range of vapor temperatures from 1250° to 1500°F. One condenser was a 7-tube counterflow shell-and-tube type convective heat exchanger, condensing potassium vapor inside the tubes and using NaK 78 as the shell side coolant. The other unit was a radiative type condenser, consisting of 9 coplanar, parallel-finned tubes manifolded to common inlet and outlet headers. This unit was tested inside an oil-cooled chamber in a vacuum environment of 3×10^{-3} torr. The objective of the tests was to study the performance and operating problems associated with current state-of-the-art units at near realistic operating conditions. The tests were conducted in the Space Radiator and Condenser Facility at the NASA Lewis Research Center. This paper presents some of the results obtained from these tests as well as a description of the units tested. A brief description of the facility and its capabilities is also included. It appears from these tests that it is possible to operate either type of multitube condenser in a stable manner over a large range of operating conditions. The advisability of using cocurrent coolant flow at the vapor inlet end of convective condensers is indicated, and the importance of properly evaluating inlet pressure drops is also pointed out by the results.

Author

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INTRODUCTION

Rankine cycle power generation systems for use in space must depend on radiation to reject the waste heat. As shown in figure 1, the waste heat can be dissipated in a radiative condenser, wherein the working fluid is condensed in a radiator. An alternate method is to use an intermediate liquid-cooled convective condenser, with the heat being lost to space from an all-liquid radiator. Each of these schemes has relative advantages and disadvantages, so that both should be studied and evaluated.

Also of interest in the application of Rankine power cycles to space missions is the use of alkali liquid metals as working fluids. The advantages of such fluids are obvious: their high saturation temperatures and low vapor pressures allow large reductions in system size and weight when compared with more conventional working fluids.

As part of the Lewis Research Center overall investigation in this field, a program was instituted to determine the steady-state operating characteristics of liquid metal condensers operating at near-realistic conditions and to define problem areas in their design and performance. Both a convective and a radiative condenser of multitube design were tested in a horizontal attitude to minimize the effort of gravity on their performance. The convective condenser was of the shell-and-tube type, and the radiative condenser was a coplanar centrally finned unit. Potassium was used as the test fluid.

This paper presents some experimental results from both condensers, as well as a description of the facility used.

FACILITY DESCRIPTION

The facility used for the condensing tests is shown in figure 2. It consists of three heat-transfer systems: an all-liquid heating loop, a two-phase loop, and a coolant system, which is either an oil-cooled vacuum chamber whenever heat transfer by radiation is under study or an all-liquid loop whenever the condenser is convectively cooled.

Liquid NaK 78 in the heater loop is pumped at rates up to 35,000 lb./hr. through an I^2R electric heater, with a capacity of 200 KW and using the flowing liquid metal itself as the resistance load. The heated liquid then flows through the shell side of the shell-and-tube boiler and returns to the pump for recirculation. The pump is of the electromagnetic type, as are the pumps in the other two liquid metal loops, but it has the added characteristic that its controls are set to allow reversal of the flow direction, permitting operation of the boiler as a countercurrent or cocurrent flow unit.

Liquid potassium, the test fluid, is pumped through the tube side of the boiler, which consists of 43 parallel 0.375-in.-o.d. by 0.035-in.-wall tubes 14 ft. long, mounted inside a 4-in.-i.d. shell. Each tube has a 0.040-in. orifice at the liquid potassium inlet. The boiler is mounted in a horizontal plane, as is the rest of the two-phase loop. The boiler outlet pipe is wrapped with radiant heaters having a capacity of 5 KW to serve as a superheater. After leaving the superheating section the potassium vapor is directed through either a radiative or convective condenser by means of manually operated shutoff valves. In either arrangement the condensate is returned to the pump, which is capable of handling a maximum flow of 600 lb./hr. of potassium during two-phase operation. Flow control is accomplished by a combination of throttling and pump speed regulation.

When a convective condenser is being studied, subcooled NaK is pumped through the shell side of the condenser at rates up to 35,000 lb./hr. The coolant is directed through shutoff valves to either an all-liquid radiator located inside the vacuum chamber, as would be the case in a flight system, or to an air cooler whose capacity can be controlled by varying the air flow rate. In both arrangements, the cooled NaK is returned to the pump for recirculation.

The vacuum chamber used as a heat sink on radiative studies is an 8-ft.-diam. by 18-ft.-long cylindrical tank equipped with mechanical vacuum pumps capable of maintaining a pressure of 10^{-3} torr. All surfaces of the tank exposed to the vacuum environment are made of stainless steel. The shell of the tank is oil cooled, and the temperature of the cooling oil is controlled by a system that includes both water-cooled and steam-heated exchangers, as well as means of varying the oil flow rate. The oil system is capable of handling the full-rated heat load of the facility. Every loop is instrumented to measure flow, temperatures, and pressures to supplement research instrumentation on the test condensers.

Operating temperatures in the systems are limited by the materials used. All the heating loop and the portion of the test loop from the boiler inlet to the superheater outlet are manufactured from a cobalt-base alloy, which allows operation up to 1850° F. The rest of the test loop and the cooling loop are fabricated from AISI 316 stainless steel, which limits operation on this section of the facility to 1600° F.

CONVECTIVE CONDENSER TESTS

Test Section

The shell-and-tube NaK cooled convective condenser is shown in figure 3. This condenser consisted of seven 1/2-in.-o.d. by 0.030-in.-wall tubes 6 ft. long, built of AISI 316 stainless steel. The tubes were set on an 11/16 in. triangular pitch inside a 2-in.-i.d. shell. Potassium vapor flowed inside the tubes and was completely condensed before leaving the unit by the NaK stream flowing on the shell side in the opposite direction. The tube entrances were tapered and rounded as much as the proximity of the tubes allowed. The shell side was unbaffled to maintain longitudinal flow in the coolant stream. The condenser was positioned with the tubes in a horizontal attitude. The unit was well insulated and heat losses in the order of 1000 Btu./hr. were measured for the operating conditions of the tests.

Test Program

The test program consisted of changing each of four independent variables over as wide a range as possible while holding the others constant and taking steady-state data at each operating condition. These independent variables were the coolant inlet temperature, the vapor inlet temperature, and the flow rates of both the NaK coolant and potassium streams.

The range of variables covered during the tests were as follows:

(1) Coolant inlet temperatures from 1160° to 1400° F

(2) Potassium saturation temperatures from 1250° to 1500° F

(3) Coolant flow rates from 6000 to 25,000 lb./hr.

(4) Potassium flow rates from 200 to 500 lb./hr.

Instrumentation

Potassium vapor pressure at the condenser inlet was measured by a Taylor gage as well as by an unbonded strain gage pressure transducer. Header-to-header pressure drop in the potassium stream was measured directly by an unbonded strain gage differential pressure transducer. Both strain gage transducers were maintained at 180° F in an oven to prevent freezing of the potassium. The transducers were mounted at a lower level than the condenser, and liquid potassium was trapped in the lines connecting the transducers to the loop. Provisions were made on the piping to allow in-service calibration of the transducers against a pressure standard during operation.

Since heat losses were very small as indicated previously, inlet and outlet temperatures of test fluid and coolant were measured by Chromel-Alumel thermocouples welded to the outer skin of the pipes. In addition, a well couple was located in the vapor inlet pipe, and a skin differential thermopile was used to read the coolant temperature change directly.

Besides inlet and outlet instrumentation, the condenser shell was equipped with skin couples located about 2 in. apart along its entire length to obtain a temperature profile of the coolant. This profile was used to determine condensing lengths for the different test conditions.

Temperatures were read on a multipoint self-balancing potentiometer capable of recording one point per second. The thermopile output was read on a manually balanced precision potentiometer. Output of strain gage transducers were recorded on a multichannel oscillograph.

Test Results

Satisfactory operation of this multitube horizontal convective condenser was obtained over a wide range of operating conditions. The results discussed in this paper will be limited to the effect of coolant inlet temperature on condensing 1400° F potassium entering the condenser at a quality of about 88 percent, with a coolant flow rate of 15,000 lb./hr.

Changing the temperature difference across the condensing surface when dissipating a constant heat load changes the amount of surface, or length, required for condensation. This fact is shown in figure 4. The coolant shell skin temperature is plotted against distance from the vapor inlet for four different values of coolant inlet temperatures. From these plots the point in the condenser where condensation ends can be located (small arrows). It should be pointed out that the curve for 1194° F coolant inlet temperature represents a mean condensing heat flux of approximately 450,000 BTU/(hr.)-(ft.^{2}), while the curve for 1315° F has a heat flux of about 47,000 BTU/(hr.)-(ft.^{2}), a decade smaller. This results from the fact that as the coolant inlet temperature becomes higher, the coolant outlet temperature approaches the vapor inlet temperature, and a longer length of very low heat flux occurs.

The vapor inlet temperature of 1400° F shown in the figure represents the temperature at the inlet header before the vapor accelerates into the tube entrances. It is to be expected that after the acceleration the tube vapor temperature will be lower than the header vapor temperature.

The condensing lengths obtained from profiles like those in figure 4 were crossplotted against their corresponding coolant inlet temperatures to obtain the performance map shown in figure 5. As the coolant inlet temperature increases at constant potassium flow rate, the vapor length becomes more sensitive to the coolant temperature as indicated by the increasing slope of the curves in figure 5. For instance, at the lower potassium flow rate curve of figure 5 and a coolant inlet temperature of 1220° F, an increase of 10° F in the inlet temperature changes the condensing length about 2 in. However, the same temperature change at a coolant inlet temperature of 1310° F changes the condensing length about 25 in. Such a change as the last one could be detrimental to system performance by an incomplete condensation of the vapor within the condenser or by introducing large oscillations in the system.

In a countercurrent condenser design with a liquid metal coolant, the condensate temperature generally is equal essentially to the coolant inlet temperature. Thus, the coolant inlet temperature is determined by the desired degree of condensate subcooling. The preceding results point to a condensing length sensitivity consideration when designing for small amounts of subcooling.

The condenser pressure drop data are presented in figure 6. The overall pressure drop from header to header is shown, including entrance losses to the tubes. Pressure drop increased with condensing length for a given inlet vapor flow rate. It is significant that satisfactory operation of the multitube condenser was obtained even when the pressure drop was essentially zero

Preliminary analysis of the data indicated that pressure drop from header to tube inlet was higher than the 10 percent of the velocity head loss anticipated for rounded entrances from gas flow considerations.

Discussion

Proper evaluation of the entrance effects can be very significant in the design of convective condensers because of the change in the vapor saturation temperature corresponding to the pressure drop. This significance is greater with liquid metal condensers for space applications than for earthbound condensers using conventional fluids because of the larger saturation temperature to vapor pressure gradients characteristic of the liquid metals and the close approach of the coolant temperature to the vapor temperature dictated by the weight optimization inherent in the design of any space power system.

The radiator is one of the heavier components of a Rankine cycle space power system, and its size depends almost exclusively on its operating temperature to the fourth power. Therefore, increasing the coolant outlet temperature (radiator inlet temperature) for a given coolant inlet temperature as dictated by subcooling requirements would reduce the size of the radiator and decrease the total weight of the heat rejection system, even though the convective condenser size may increase. When coolant flow is countercurrent to the vapor flow, the saturation temperature of the vapor at the tube inlet throat represents the upper limit for the coolant outlet temperature. If, however, the coolant flow is cocurrent, it can be possible for the coolant outlet temperature to be higher than the value of the vapor saturation temperature at the vapor inlet throat. This can only be possible if the vapor saturation temperature increases along the condensing length. This saturation temperature rise can be obtained if the heat flux rate is sufficiently high to allow the momentum pressure recovery to be greater than the frictional pressure drop, thus creating a static pressure rise as condensation progresses.

Of course, optimization of the complete heat rejection system, not the condenser and radiator alone, will determine the final operating characteristics of any one application.

RADIATIVE CONDENSER TESTS

Test Section

The other multitube condenser tested was the radiative unit shown partially fabricated in figure 7. This unit consisted of nine centrally finned tubes 14 ft. long manifolded into common inlet and outlet headers, with all tubes set in the same plane. Material of construction was AISI 316 stainless steel. The tubes had a 0.500-in. i.d. and a 0.420-in. thick wall. The centrally located fins were 0.90 in. long and 0.080 in. thick. The tubes were welded tangentially to the bottom of the inlet header on a 3.25 in. pitch, with the tube entrances being rounded and smoothed. A gap of 1/8 in. was left between the edges of adjacent fins.

This tube cross section was selected by a weight optimization design procedure (ref. 1) developed by Lewis Research Center engineers for two-phase radiators. This procedure took into account the power system size, as well as mission requirements and environmental hazards. This tube configuration was selected for a 300 KW (electric) Rankine cycle power system with a 90 percent probability of successfully completing a 500-day mission on the basis of meteoroid population information available in 1961. The nine-tube panel used as a test unit constitutes one tenth of the 90-tube radiator required for such an application. Knowledge about meteoroid hazards has recently increased greatly, and the same design program today would dictate much thinner tubes.

Surface Preparation

After fabrication, the outside surface of the radiative condenser was sandblasted and oxidized in order to provide a stable high emissivity surface. Oxidation was selected over other coatings because of its relative simplicity and durability. Oxidation was accomplished by inserting the whole panel in an open-air furnace at 1200° F for a period of 24 hrs. A resultant total hemispherical emissivity of 0.72 to 0.75 was measured on test coupons that had been attached to the radiator panel throughout the surface preparation procedure.

Test Unit Mounting

The radiative condenser was mounted inside the vacuum tank shown in figure 8. The inside surfaces of the tank were sandblasted and coated with black paint which gave the wall an emissivity of 0.91 to 0.93. The pressure in the vacuum tank was maintained near 3×10^{-3} torr. In this rarified atmosphere the amount of heat transferred by convection from the radiator to the walls was less than 1/2 percent of the total heat load.

Instrumentation

Figure 9 shows the radiative condenser mounted inside the vacuum tank in a horizontal attitude and completely instrumented prior to operation. This unit lent itself more readily than the convective condenser to the application of extensive instrumentation. Each of the nine tubes had thermocouples spot-welded to its outside surface every 8 in. along its entire length. These thermocouples provided temperature profiles for every tube. In addition to these longitudinal profiles, thermocouples were applied to obtain cross-sectional profiles of tube and fin temperatures at three locations along the radiator.

Immersion couples were applied to the inlet header, and wall couples at inlet and outlet pipes outside the vacuum chamber were included as well.

Cooling-oil temperature rise across the vacuum chamber was measured as well as vacuum tank surface temperatures, both on the vacuum side and on the room side of the chamber. The former were used for monitoring the inside temperature of the tank, the latter to calibrate and meter the heat losses from the vacuum tank to the surrounding atmosphere. A total of over 250 thermocouples were employed to instrument the radiator and vacuum chamber and were recorded at the rate of 40/sec. with the aid of an automatic digital data recorder system.

Pressure instrumentation was applied to the inlet and outlet headers of the radiator as well as to the tube entrance of three different tubes. This pressure instrumentation was of the unbonded strain gage type and the application was similar to the one used on the convective condenser tests.

Test Program

The only independent variables affecting the performance of the radiative condenser were the potassium flow rate and temperature level. The vacuum tank walls were maintained at temperatures below 300° F, and any temperature variations at this temperature level had negligible effect on the radiator performance. The test program consisted of taking steady-state data at potassium flow rates ranging from 200 to 600 lb./hr. at saturation temperatures of 1235°, 1350°, and 1420° F.

In addition to the temperature, pressure, and flow data obtained for each run, visual and photographic observation of the radiative condenser during operation was carried out by utilizing two viewing windows located in the end of the vacuum tank. Over the range of vapor temperatures studied, enough light was emitted by the radiator as the only source of illumination to allow black-and-white as well as color photographs to be taken.

Results

The radiator was operated successfully as a condenser for a continuous span of 250 hr., during which time more than 80 steady-state data conditions were obtained. Figure 10 shows a typical temperature profile obtained for one of the tubes. It can be seen that the outside temperature of the tube remained fairly constant along the condensing length. Once condensation was completed, the wall temperature fell steeply, indicating the subcooled temperatures of the condensate. The intersection of the two dissimilar temperature patterns clearly indicated the position of the interface in the tube. Similar profiles were obtained for all tubes, and the addition of all the condensing lengths for each run gave the measure of the radiator area used for condensation.

The high sensitivity of condensate outlet temperature to length available for subcooling should be noted. In contrast to the convective condenser, where the coolant inlet temperature fixes the amount of subcooling, the radiator is dissipating its energy to a very low temperature sink and any miscalculating on the length available for subcooling will alter the condensate outlet temperature drastically.

Mean condensing heat fluxes varied from 23,000 to 49,000 BTU/(hr.)-(ft.^{2}) depending on vapor temperature. A photograph of the radiator taken during the same run as figure 10 is shown in figure 11. The photograph was taken as a time exposure by using the emitted light from the radiator. The subcooling length of the radiator dropped in temperature so abruptly that it appears dark in the photograph. It can be noticed that the condensing length in all of the tubes was not the same. For this particular run there was a difference of approximately 12 in. between the shortest and longest condensing lengths. This difference varied from 0 to 24 in. over the range of conditions tested.

The results from the different runs are presented graphically in figure 12 as average condensing tube length against average inlet vapor flow rate per tube. The vapor phase flow rate was obtained from the total flow rate and the inlet quality as calculated from heat balance considerations. Results are grouped according to inlet vapor temperatures. The solid lines shown in the figure are performance lines obtained from heat fluxes calculated from an analytical study (ref. 2) of the cross section of the same fin-tube configuration. This analysis assumed the inside surface of the tube to be at a constant temperature equal to the inlet header vapor temperature; that is, it ignores any pressure drops or film coefficients on the vapor side. It can be seen that there is good agreement between experimental and simplified analytical results at low values of flow rate and high temperature levels. However, the deviation between actual and ideal values increased at higher flow rates, especially at low inlet vapor temperatures.

A reason for this increase in actual over ideal condensing length with increase in flow and decrease in vapor temperature is the effect of pressure drop. As the pressure drop increases the vapor temperature in the tubes becomes lower than the header temperature if it is to remain in equilibrium. Therefore, the header vapor temperature no longer represents the vapor temperature within the tubes. This effect becomes more pronounced at the lower values of vapor temperature because of the increasing vapor temperature-to-vapor pressure gradient of potassium.

The pressure drop characteristics of the radiative condenser appear in figure 13 as header-to-header pressure drop against average inlet vapor flow rate per tube. Comparing figure 12 and 13 shows that the higher deviations of condensing length from the ideal values correspond to the higher pressure drops.

Discussion

Examination of the results from the radiative condenser tests indicated that pressure drop characteristics were again the most significant factor in the operation of the condenser from the designer's point of view. In the case of the radiative condenser, errors in overestimating as well as underestimating pressure drops could cause significant design deficiencies. If actual pressure drop is less than predicted the radiator would be too large because of the higher surface temperatures encountered during condensation. The excess surface will then provide condensate subcooling below design temperature which may degrade the boiler performance. If actual pressure drop is greater than the predicted pressure drop, the radiative condenser would be too small to handle the design heat load. This condition is similar to that experienced on the convective condensers, but it is aggravated by the fact that the radiative unit cannot be designed with excess surface because of the effects of excessive subcooling.

CONCLUDING REMARKS

The results of the work performed during this program can be summarized as follows:

1. Successful operation of multitube potassium horizontal condensers, both convectively and radiantly cooled, was obtained over a vapor temperature range of 1250° to 1500° F. No appreciable instabilities were encountered over the range of operating conditions even when the pressure drop across the unit was essentially zero.
2. Pressure drops during condensing and entrance effects had a marked bearing on liquid metal condenser performance because of the large saturation temperature-to-vapor pressure gradient of the liquid

metal. Close evaluation of the effect of pressure variations on condensing length is of great importance to the designer.

3. The importance of properly predicting the condensing length is much greater in the design of a radiative condenser than in the design of a convective condenser. Improper evaluation of the condensing length on a radiative condenser will affect the subcooling obtained because of the very low value of the sink temperature. The same discrepancy on a convective condenser can be compensated for by adding extra surface which will not affect the condensate subcooling which is fixed by the coolant temperature.

REFERENCES

1. Krebs, R. P.; Haller, H. C.; and Auer, B. M.: Analysis and Design Procedures for a Flat, Direct-Condensing, Central Finned-Tube Radiator. NASA TN D-2474, 1964.

2. Stockman, N.; Bittner, E. C.; and Sprague, E. L.: Comparison of the One Dimensional and Two Dimensional Heat Transfer in Central Finned Tube Radiators. Proposed NASA Technical Note.

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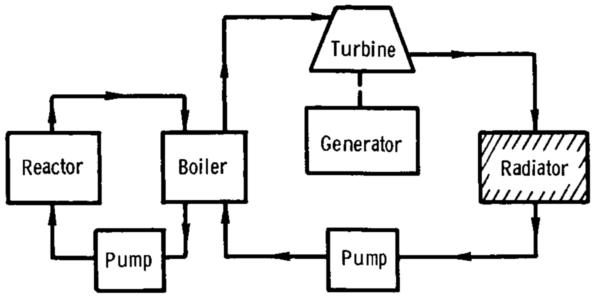
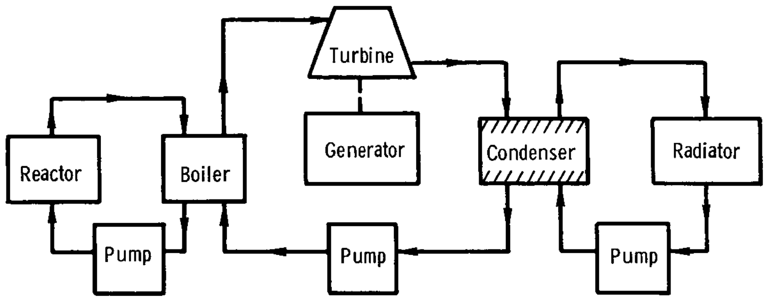


(a) Radiative condenser.



(b) Convective condenser.

Figure 1. - Rankine power systems.

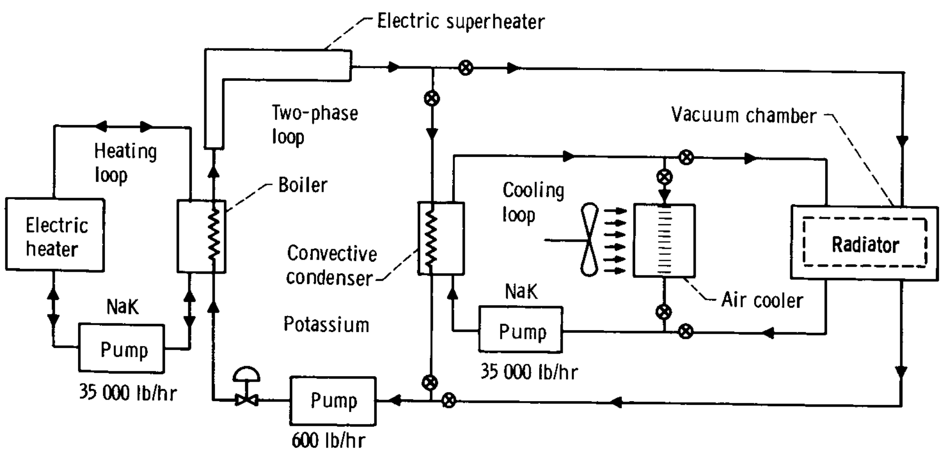


Figure 2. - Schematic test facility.

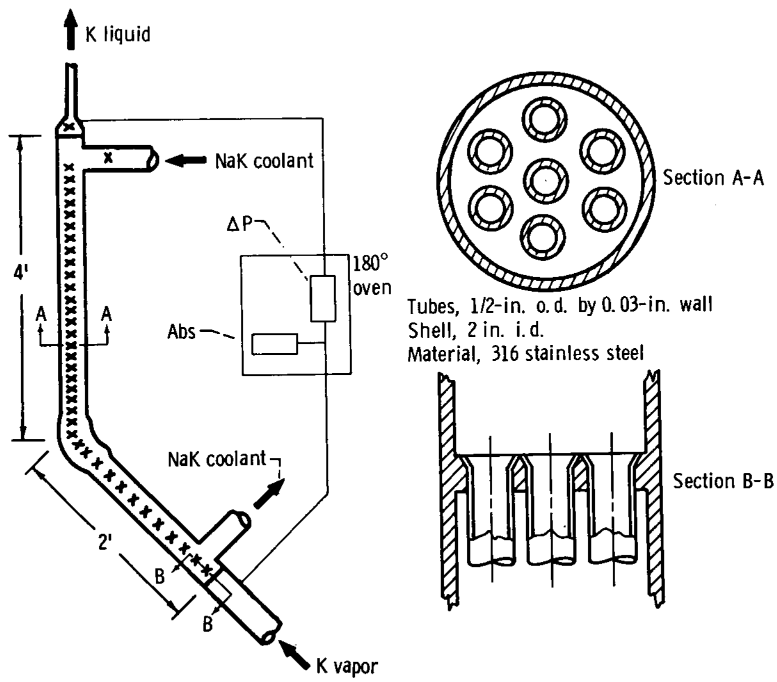


Figure 3. - Seven-tube NaK cooled condenser.

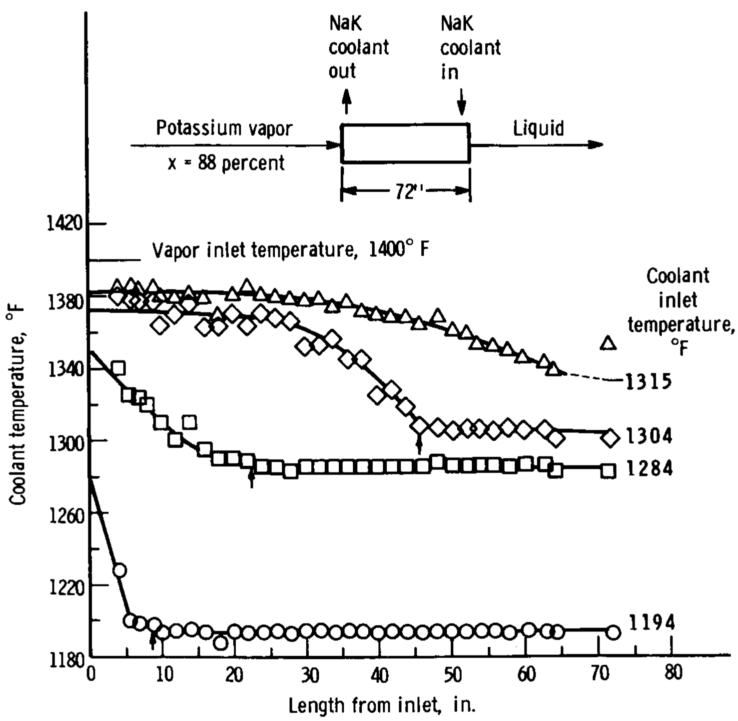


Figure 4. - Coolant shell temperature profiles. Seven-tube potassium condenser; potassium flow, 300 pounds per hour; coolant flow, 15 000 pounds per hour.

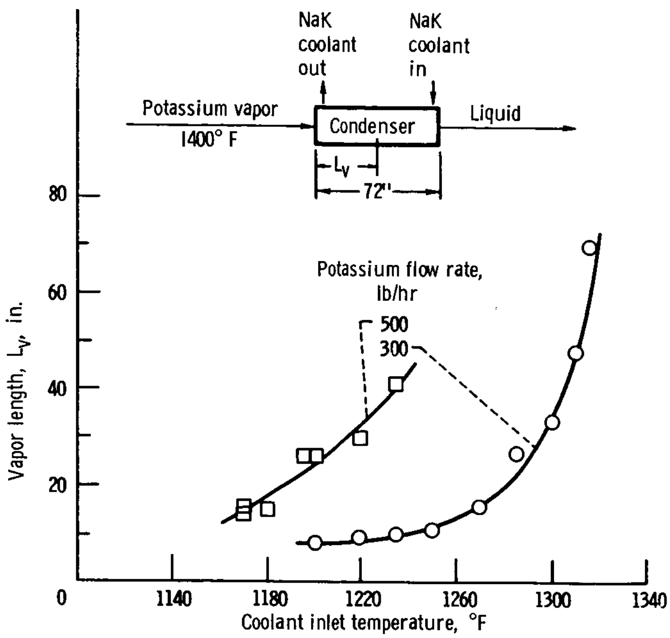


Figure 5. - Vapor lengths plotted against coolant inlet temperature. Seven-tube convective condenser.

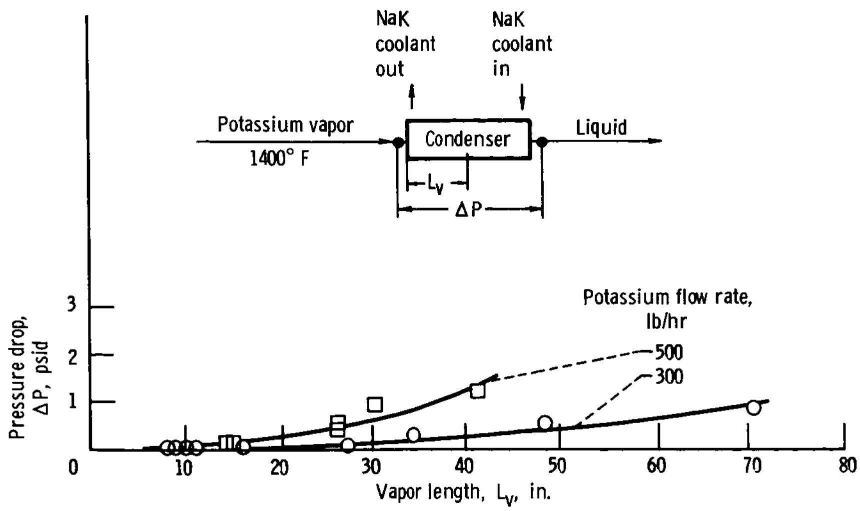


Figure 6. - Pressure drop plotted against vapor length. Seven-tube convective condenser.

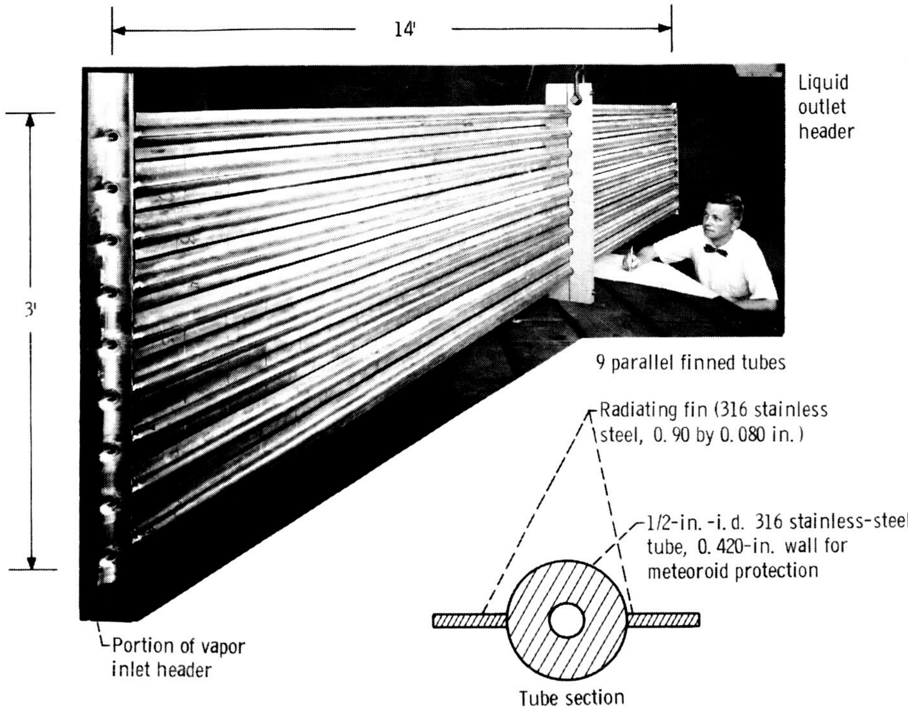


Figure 7. - Nine-tube radiative condenser.

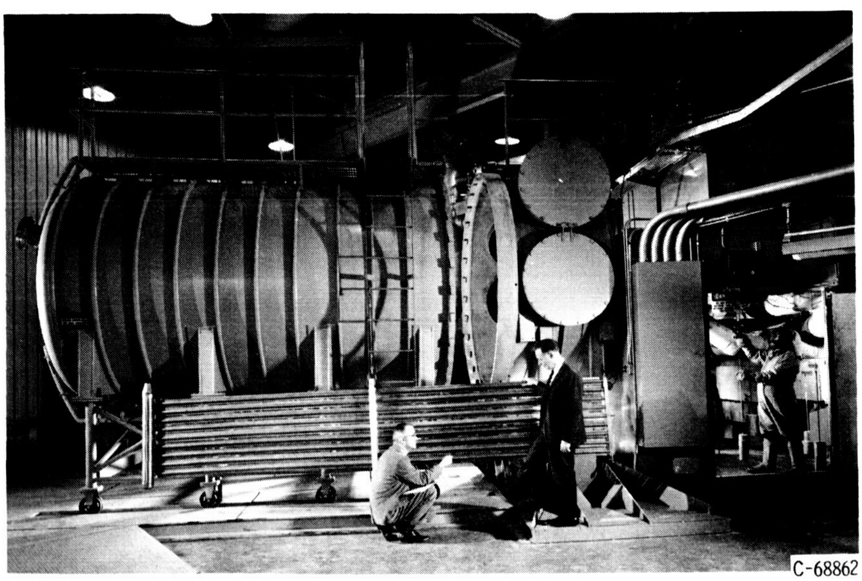


Figure 8. - Facility vacuum tank.

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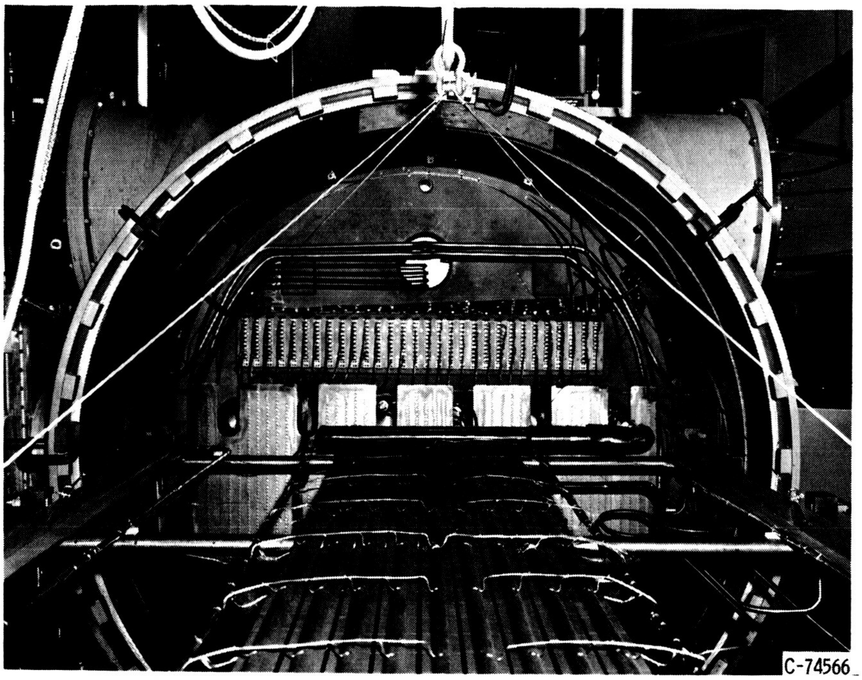

Figure 9. - Radiative condenser installation.

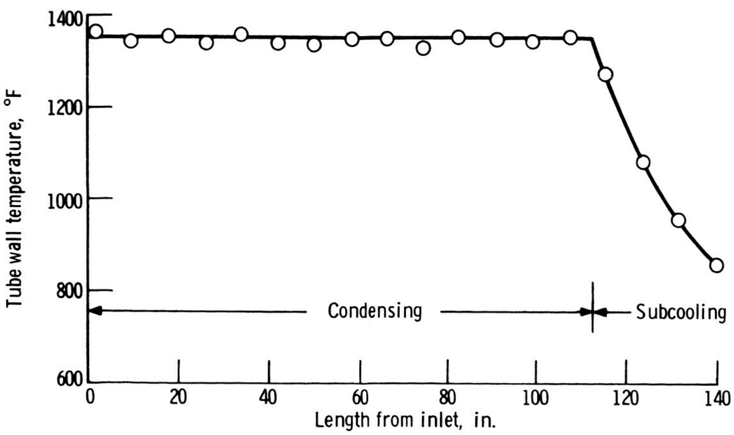

Figure 10. - Radiative condenser wall temperature profile. Vapor inlet temperature, 1426° F; potassium flow rate, 569 pounds per hour; vapor quality, 83 percent.

Figure 11. - Photograph of radiative condenser during operation at 1400° F vapor inlet temperature.

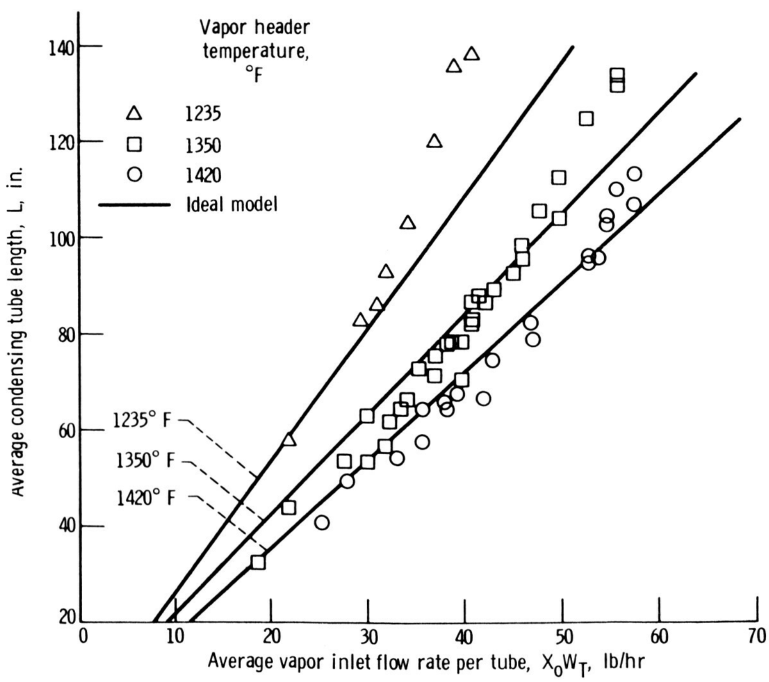

Figure 12. - Average condensing length plotted against average vapor flow per tube. Nine-tube radiative condenser.

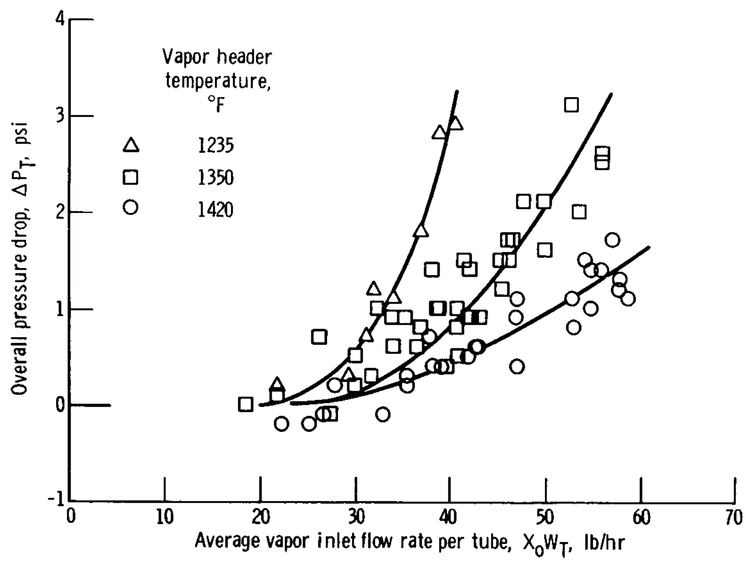

Figure 13. - Overall pressure drop plotted against average vapor flow rate per tube Nine-tube radiative condenser.