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CENTRAL FILES NUMBER

67-3-38

DATE: March 21, 1967 COPY NO. 17

SUBJECT: Heat Transfer Performance of the MSRE Main Heat Exchanger and Radiator

TO: Distribution

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ABSTRACT

Soon after the operating power of the MSRE was raised to a significant level, the heat-transfer capability of the main heat exchanger and of the coolant radiator was found to be less than had been predicted. A design review indicated that the heat exchanger had been designed correctly using the existing data but that the radiator airside heat-transfer coefficient had been calculated using air physical properties that were evaluated at the metal surface temperature rather than the average "film" temperature. The poor performance of the main heat exchanger was partially explained by recent measurements of the fuel salt thermal conductivity which indicated a value of 0.83 $\frac{\text{Btu}}{\text{Hr-ft-°F}}$ rather than 2.75 $\frac{\text{Btu}}{\text{Hr-ft-°F}}$ which was used in the calculations. Even after the correct physical property data was applied, the observed performance was still below the calculated by about 14% and 25% respectively for the heat exchanger and the radiator. No causes for the low heat transfer have been found other than possible errors in other physical properties or in the heat-transfer correlations. There is no evidence of gas filming or of the buildup of a scale on the tubes.

This report covers the design review of the heat exchanger and radiator and also the evaluation of the actual performance.

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INTRODUCTION

The nominal power chosen for the design of the MSRE was 10 Mw. Escalation of the reactor power in April and May, 1966 showed that the heat-transfer capability of the air-cooled radiator was less than anticipated and, in fact, limited the attainable heat removal to about 7.5 Mw. The overall heat-transfer coefficient of the primary heat exchanger was also below the predicted value, resulting in somewhat larger fuel-coolant temperature differences than had been planned.

After the first indications of low heat transfer, we reexamined the reactor data to determine heat-transfer coefficients as accurately as possible and to see if the coefficients varied with power level or operating time. Meanwhile we reviewed the original design work to see if there were errors in calculational method or physical properties that could account for the apparent discrepancy between predicted and observed performance.

EXPECTED AND OBSERVED PERFORMANCE

Primary Heat Exchanger

The performance of this heat exchanger determines the temperature difference which must exist between the fuel and coolant salts at any specified power level. (The temperature changes of the fuel and of the coolant as they pass through the exchanger depend not on the exchanger but on the flow rates and heat capacities.) Poorer performance means at a given power the fuel must be hotter and/or the coolant cooler.

Nominal operating conditions, listed on the flowsheets, are: power, 10 Mw; fuel entering the heat exchanger at 1225°F, leaving the heat exchanger at 1175°F; coolant leaving the heat exchanger at 1100°F and entering the heat exchanger at 1025°F. Actually, it was expected that temperature differences would be somewhat smaller than these because of conservatism in the heat-exchanger design and in the predicted flow rates.

Temperature differences observed during operation are shown in Fig. 1. Shown for comparison are lines representing the nominal relations, i.e.,

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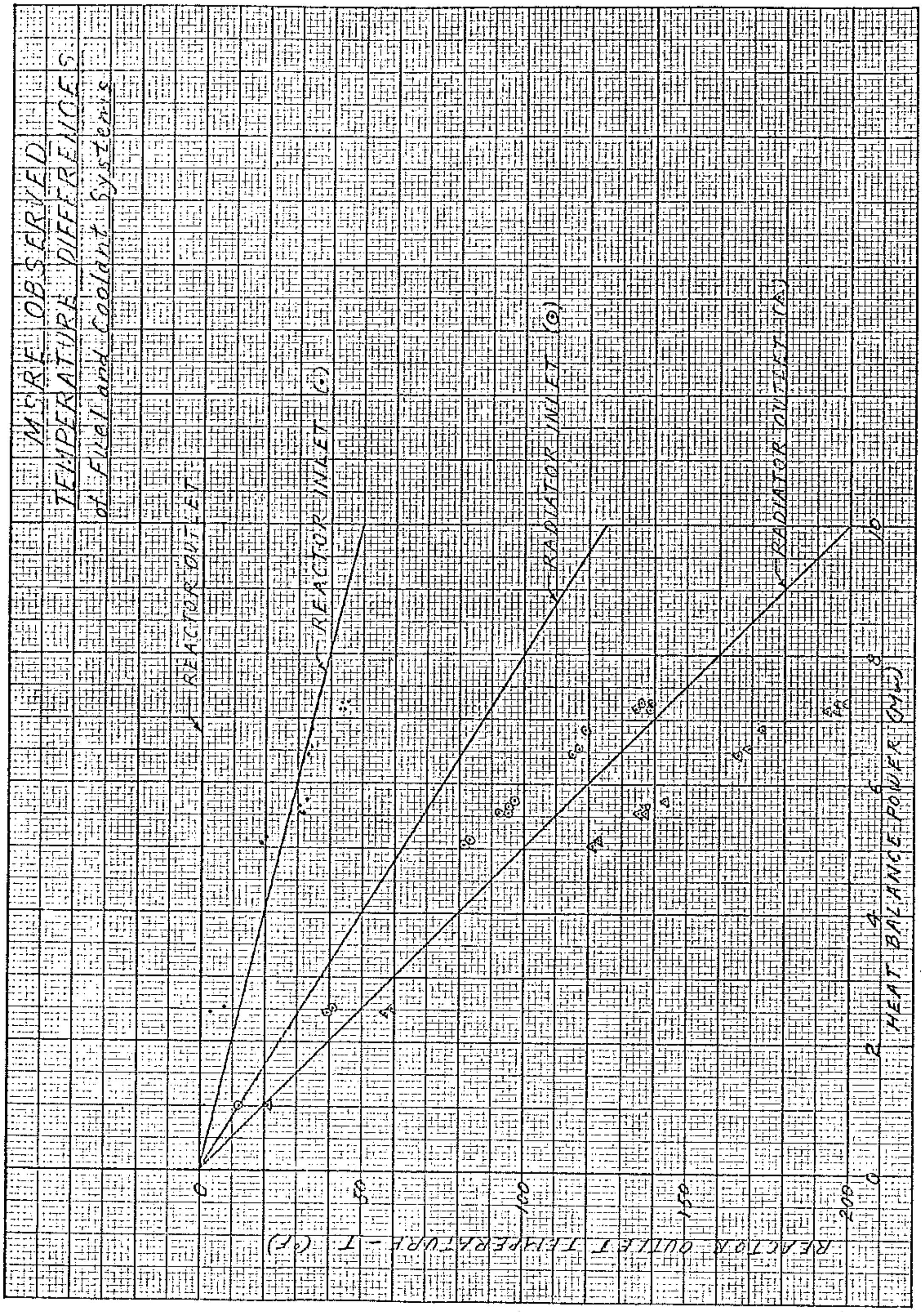

FIGURE-1

straight lines from zero through the nominal 10-Mw values. (Changes in viscosities, flow rates, and heat capacities with salt temperature or power level, which would cause the lines to curve, are ignored in this simple comparison.) It is apparent that the temperature differences between the fuel and coolant systems, instead of being somewhat smaller than nominal as might have been expected, were actually larger. Thus the performance of the heat exchanger was recognized to be substantially below expectations.

It is worth nothing here that the power levels in Fig. 1 are values obtained from heat balances which is our best estimate of the actual heat load on the exchanger. Some confusion existed for a while because the calibration of the neutron instruments used for continuous indication of reactor power changed during the operation. This change was later related to increasing water temperature in the nuclear instrument shaft and the erroneously high values of power from the neutron indication were revised downward.

<u>Radiator</u>

In the case of the radiator, the temperature difference between the salt in the tubes and the air outside is quite large (about 850°F at full power). Thus only small fractional changes result from variations in ambient air temperature or the limited changes which can be made in salt temperatures. As a result, poorer performance in the radiator cannot simply be compensated by taking a larger ΔT at a chosen power. Poorer performance means the attainable power level is lower.

Neglecting the secondary effects of salt and ambient air temperatures, the heat removal depends on the air flow distribution over the tubes. The flow distribution in turn depends on the pressure drop available across the radiator and the position of the radiator doors. The driving ΔP depends on the number of blowers running, the blower blade angle, the position of the bypass damper and the positions of the doors.

Table 1 lists heat-removal rates for several conditions. The predicted rates were calculated assuming the nominal coolant salt temperatures for a 1200°F mean core temperature and ambient air at 100°F. Salt and air temperatures were not greatly different at the times of the observations. Total air flow rates were also close to those used in the predictions. The comparison therefore shows that the performance of the radiator, i.e., the heat-transfer coefficient for the tubes was significantly less than predicted.

Table 1

MSRE HEAT REMOVAL RATES

CORE OUTLET TEMPERATURES ~ 1200°F

Conditions at Radiator	Predicted (Mw)	Measured (Mw)
One Blower on; By-Pass open	3.8	4.1
One Blower on; By-Pass closed	6.5	5.8
Both Blowers on; By-Pass open	6.1	5.9
Both Blowers on; By-Pass closed	10.0	7.5 (max)

DESIGN REVIEW

Primary Heat Exchanger

The primary heat exchanger is a conventional cross-baffled, U-tube exchanger. Fuel salt circulates on the shell side at 1200 gpm and coolant salt circulates at 850 gpm through the tubes. The exchanger now contains 159 half-inch tubes on a triangular pitch. For a more detailed description, see Reference 1. Design calculations and considerations are presented in Reference 2.

The methods used in the design of the MSRE heat exchanger are those commonly followed in designing heat exchangers of this type. The tube-side coefficient was computed from the Sieder-Tate equation, and the shell-side coefficient was computed from a correlation by Kern.[3] Implicit in the use of these procedures is the assumption that the fused salts behave as normal fluids, i.e., that conventional relationships can be used to compute heat-transfer coefficients in fused salt systems. Although there are some inconsistencies in the literature concerning this assumption, we believe that these inconsistencies could be resolved, given adequate knowledge of fused salt physical properties, scaling and corrosion characteristics, and this sort of thing. That this basic assumption is valid for salt flowing inside tubes is shown in Reference 4 while Reference 5 shows the same for salt flowing on the outside of a tube bundle. Thus we conclude that the calculational methods were appropriate.

The design calculations tend to give a conservatively low prediction of the heat-transfer capability (effective UA) for four reasons. First, the correlation for shell-side coefficient by Kern is conservative, i.e., his design curve falls below the data points rather than through the mean. This would tend to make the predicted shell-side coefficient low by 0-20%. Since the shell-side resistance is about a third of the total, the effect of this conservatism on the predicted overall coefficient, U, is about 0-6%. Second, the predicted coefficient would also be low because an additional resistance of about 11% was added arbitrarily to allow for scale. This was done even though it has been shown, both in and out of pile, that the salts do not corrode or deposit scale on INOR-8 under MSRE operating conditions. The third conservative approximation was in the definition of the effective heat-transfer surface, A. Here no credit was taken for the bent part of the tubes, i.e., the active length of the tube was taken to be the straight portion between the thermal barrier near the tubesheet and the last baffle. This approximation was made in recognition that the thermal efficiency of the return bends might be less than that of the straight portions. Nevertheless, this region contains 7 to 8 percent of the total tube area and will transfer a significant amount of heat. Finally an additional 8% of active heat-transfer area was added to the computed requirement as a contingency factor. The net result is that a deliberate margin for error of over 20% was included in the design.

Between the design and the operation of the heat exchanger, some modifications were made. When the heat exchanger was hydraulically tested with water before being installed in the reactor, the pressure drop was excessive and the tubes vibrated. To reduce the high pressure drop, the outermost row of four tubes was removed and the corresponding holes in the baffle plates were plugged. To alleviate the tube vibration problem, an impingement baffle was placed at the fuel salt inlet. In addition the tubes were "laced" with rods next to each baffle plate to restrain the lateral movement of the tubes. A laced structure was also built up in the return bend to make these tube projections behave as a unit, and the tubes essentially support each other. No attempt was made to measure the overall heat-transfer coefficient, but it does not appear that these changes were enough to affect the conservatism in the original design. The effect of the rods and impingement baffle is probably negligible. The loss in heat transfer by the removal of the four tubes is also relatively small. The heat-transfer area of the removed tubes was only about 2.5% of the total; the effect on capacity was probably less because these particular tubes, by virtue of their proximity to the shell, would be expected to have heat-transfer coefficients below the average.

At this point we concluded that the design methods were appropriate, the assumptions conservative, and that subsequent modifications should not have used up the margin of safety believed to be provided in the design. One factor remained to be evaluated — the physical properties used in the computations. Table 2 lists the values that were used.[2] Table 3 shows how the calculated resistances depend on the physical properties.

Table 2

PHYSICAL PROPERTIES USED IN DESIGN
of
MSRE PRIMARY HEAT EXCHANGER

		Fuel	Coolant
Salt:	thermal conductivity, k(Btu/hr-ft-°F)	2.75	3.5
	viscosity, μ(lb/ft-hr)	17.9	20.0
	density, ρ(lb/ft^3)	154.3	120
	specific heat, C(Btu/lb-°F)	0.46	0.57
	prandtl number	3.00	3.26
Metal:	thermal conductivity, k(Btu/hr-ft-°F)	12.2	

Table 3
HEAT TRANSFER RESISTANCES IN PRIMARY HEAT EXCHANGER
and
DEPENDENCE ON PHYSICAL PROPERTIES

Resistance	Calculated Value $\frac{°F}{Btu/hr\text{-}ft^2}$	Inversely Proportional To:
Inside film	1.85×10^{-4}	$\frac{C^{1/3}\ k^{2/3}}{\mu^{0.47}}$ (coolant)
Tube wall	3.35×10^{-4}	k (metal)
Outside film	2.81×10^{-4}	$\frac{C^{1/3}\ k^{2/3}}{\mu^{0.22}}$ (fuel)

The physical properties values were the best available at the time. More recent information[13,14] indicates that the conceivable error in some of them is enough to have considerable effect on the overall heat transfer. This will be discussed later in this memo.

Radiator

The heat-transfer surfaces of the radiator consist of 120 unfinned 3/4-inch tubes, each about 30 ft long. The S-shaped tube bundle, consisting of 10 staggered banks of 12 tubes each, is located in a horizontal air duct so that air blows across the tubes at right angles. Doors can be lowered just upstream and downstream of the tubes to vary the air flow over them. A bypass duct with a controlled damper and the option of either one or two blowers provide other means of varying the air pressure drop across the radiator. (For a detailed description, see Reference 1.)

The basic design calculations for the radiator are reported in References 6 and 7. Predicted performance for various combinations of doors, damper and blowers are given in References 8 and 9.

As in the design of the primary heat exchanger, the Sieder-Tate equation was used to calculate the heat-transfer coefficient on the inside of the tubes. The same comments as to validity of method and accuracy of salt properties apply in both designs. In the radiator, however, only 2% of the calculated heat-transfer resistance is inside the tubes, so no conclusions with regard to accuracy of the inside film calculations can be drawn from the observed performance.

Over 95% of the resistance is on the air side. This coefficient was calculated using an equation by Colburn recommended by McAdams.[10] This equation is well-proven for cross-flow geometries identical in all essentials to the MSRE radiator. The difficulty with applying the equation to the MSRE design is the very large difference between the tube temperature and the bulk temperature of the air. The physical properties of the air vary so much over this range that relatively large variations in the heat-transfer coefficient can be calculated depending on which temperature is selected for the evaluation of the physical properties. The MSRE design calculation used air properties at the temperature of the outside surface of the tubes. The procedure recommended by McAdams is to evaluate the properties at a "film temperature" defined as the average of the surface and the bulk air temperatures. Had this been done, the outside film coefficient (and the overall coefficient) calculated for the MSRE radiator would have been lower by some 14%. Even lower values would have resulted if the physical properties had been evaluated nearer the temperature of the bulk of the air.

The surface area of the radiator tubes was increased by about 4% over the calculated minimum for 10-Mw heat removal. In light of the unconservative evaluation of the outside coefficient, this increase would appear to be far too small to provide any factor of safety.

A heat-transfer coefficient calculated using the recommended air film temperature would still be greater than the observed value by about 25%. A contingency factor of this magnitude would not be unreasonable when the large air-to-tube surface temperature difference is considered and when the unconventional geometry of the tube bundle within its enclosure is considered.

ANALYSIS OF PERFORMANCE

Primary Heat Exchanger

Why was the performance of the heat exchanger not as good as was expected? In seeking the probable answers to this question, it is desirable to make a comparison on the basis of overall heat-transfer coefficient. There are two difficulties or sources of error in making this comparison. First is the question of effective area in this U-tube heat exchanger. The design calculations give a value of U for the straight portion of the tubes while the observed performance will yield a value for effective UA which also includes the surface area in the bends. Since the shell-side flow distribution is a combination of parallel and cross flows in both the straight section and in the bends, the heat-transfer characteristics should be approximately the same. Therefore, the total outside area of the tubes will be used in evaluating the overall coefficient. The second important source of error is the effect of temperature measurement errors on the "observed" UA.

The primary heat-exchanger performance was originally evaluated from data taken on May 26, 1966 when the reactor was operated at several different power levels. Two methods were used in computing the overall heat-transfer coefficient. First, conventional calculation was performed and second, a rather novel technique was used which eliminates certain types of thermocouple errors. Each method will be discussed separately.

In the "conventional" calculation, the following equation was used:

$$U = \frac{Q}{A\ \Delta t_{lm}}$$

where U = overall heat-transfer coefficient

Q = heat transferred, computed from a coolant salt heat balance

Δt_{lm} = applicable log mean temperature difference for a heat exchanger of this geometry

A = 279 ft^2 - total heat-transfer area based on tube OD and including return bends in U-tubes. Does not include area behind thermal barrier plate.

For this calculation the accuracy of the temperature measurements was questioned, even though a thermocouple calibration bias is periodically measured and automatically added (or subtracted) by the data logger. On May 27, the reactor was operated for a period of time at 10 kw. At this power level the entire fuel and coolant salt systems were essentially isothermal, and all thermocouples used in this analysis should have read the same. Temperature correction factors were then determined to force the temperatures to be the same. These correction factors were of an additive (or subtractive) nature. It was then assumed that the correction factors were independent of power level and temperature over the range of interest, and all temperatures from May 26 were corrected accordingly.

The second technique used in computing overall coefficients is important because it is not influenced by thermocouple errors, provided that the errors themselves are not functions of temperature over the range of interest. The basic equation, which is presented in Reference 11, was modified to fit the geometry of the MSRE heat exchanger (see appendix). The final equation takes the form:

$$\frac{d\left[T_{fo} + T_{fi} - T_{co} - T_{ci} + \Delta T_c \sqrt{1 + \left(\frac{F_c Cp_c}{F_f Cp_f}\right)^2}\right]}{d\left[T_{fo} + T_{fi} - T_{co} - T_{ci} - \Delta T_c \sqrt{1 + \left(\frac{F_c Cp_c}{F_f Cp_f}\right)^2}\right]} = e^{\frac{UA}{F_c Cp_c}\sqrt{1 + \left(\frac{F_c Cp_c}{F_f Cp_f}\right)^2}}$$

where T = measured salt temperatures

F = mass flow rates

Cp = Heat capacity

subscripts

f - fuel

c - coolant salt

i - heat exchanger inlet

o - heat exchanger outlet

To use this equation, the term in the numerator on the left is plotted against the term in the denominator. From the slope of the resulting line, the overall coefficient (U) can be calculated.

The results of these two calculation procedures are shown in Figure 2. Theoretically the results should be identical but in practice there is some disagreement between the two, especially at the lower powers. There are at least two possible reasons for the disagreement.

1. Some of the parameters used in the second procedure do not appear in the first procedure (F_f and Cp_f, for example). In addition, the parameter F_f is a calculated number rather than a measured one.
2. The temperature corrections used in the first procedure may have been slightly in error.

Generally, and especially at the higher power levels, the agreement between the two methods is good. At low powers the effects of thermocouple errors become increasingly important when using the conventional procedure. This is evident from the progressively larger scatter in the data at the lower powers.

Both procedures indicate a power dependence on the heat-transfer coefficient. This power dependence can be attributed to the change in physical properties of the salt with temperature. As the reactor power is raised from zero to 7.5 Mw, the average salt temperature in the heat exchanger drops from 1225°F to about 1135°F (if the reactor outlet temperature is held constant at 1225°F). As a first approximation, the heat-transfer film coefficients of the fuel and coolant salts will vary as $(Cp/\mu)^{0.4}$. This relation assumes that both the inside and outside coefficient vary with the 0.4 power and that the temperature dependent changes of thermal conductivity and density are negligible in the range of interest. The change in overall heat-transfer coefficient with reactor power was calculated by the above relation using the viscosity and specific heat temperature dependence presented in the MSRE Design Data Sheets.[12] The results of these calculations are also shown as the dashed line on Fig. 2. The absolute value of the coefficient was normalized to 600 $\frac{\text{Btu}}{\text{hr-ft}^2\text{-°F}}$ at 7.5 Mw and therefore only the slope of the line is significant. The slopes

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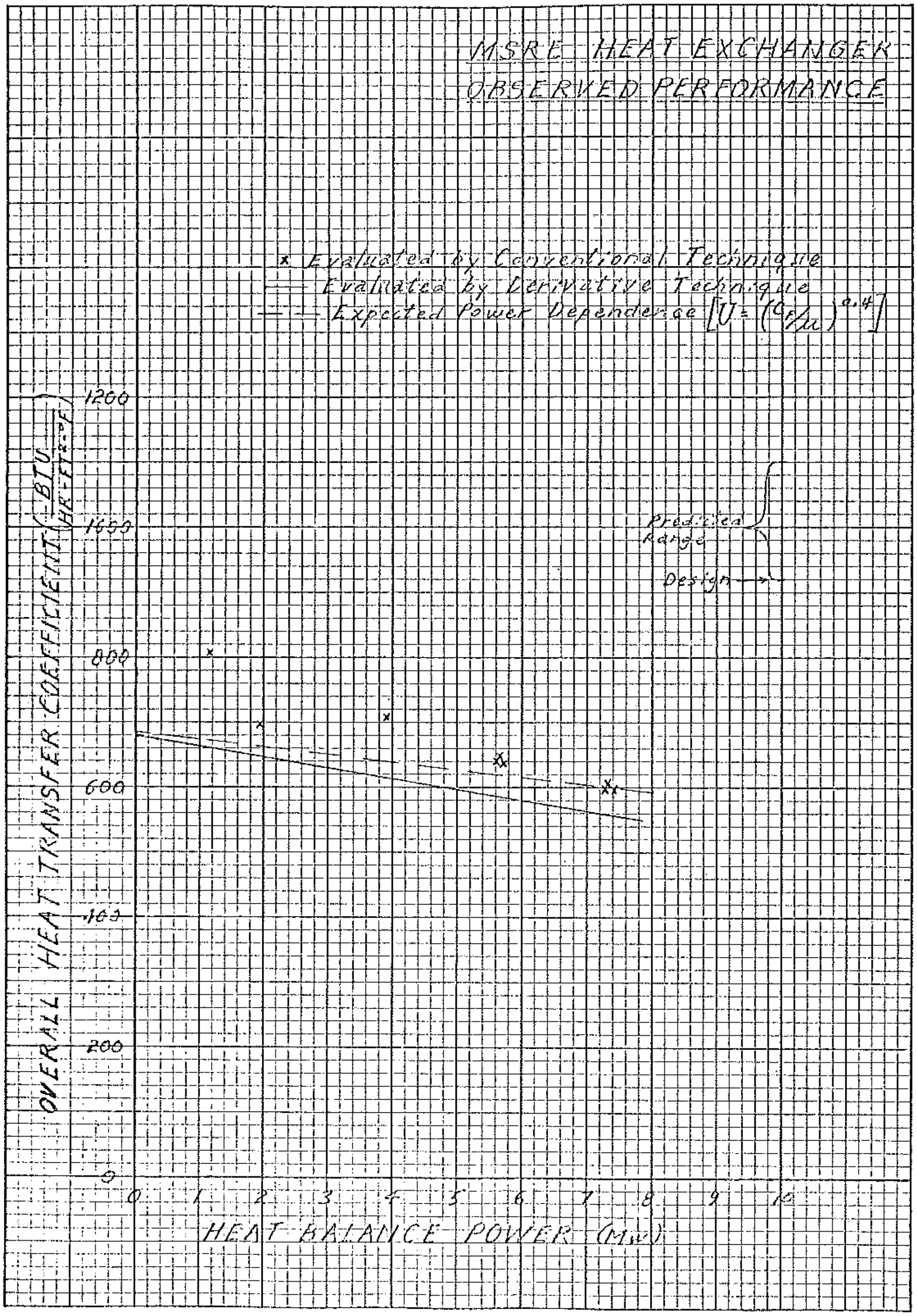

FIGURE - 2

of these three plots of Heat-Transfer Coefficient vs Power were 24, 17, and 12 respectively for the conventional, derivative, and $(Cp/\mu)^{0.4}$ calculations. If the derivative method is assumed to give the correct value, the $(Cp/\mu)^{0.4}$ dependence agrees within 30%. A more refined calculation evaluating the temperature effects on the fuel and coolant separately would tend to improve the agreement.

A second possible cause of the low and power-dependent heat transfer would be gas filming of the tubes. A series of pressure release experiments to determine the circulating gas void of the fuel salt has been completed. These experiments consisted of raising the fuel-system pressure from the normal value of 5 psig to 15 psig and suddenly venting off the excess pressure. Data from these experiments showed no indication of a reduction in heat transfer as the system pressure was vented off. If a gas film had been present, the changing pressure would have changed the thickness of the film and would have reduced the heat transfer.

Several sets of data have been collected over the life of the reactor and have been analyzed by the derivative method to determine if any significant decrease in heat transfer has occurred with time. The data for this evaluation cover a span of 4,470 and 7,100 hours respectively of fuel and coolant salt circulation and 21,220 Mw-hrs of integrated power. Figure No. 3 shows the results of this analysis which seem to indicate a gradual decrease in the heat-transfer rate with time. However, this apparent decrease is believed to be within the uncertainty band of the data. The data for the last three points were taken over a full range of reactor powers with more closely controlled operating conditions so that these points should represent the best data. Another sensitive index to changes in heat transfer is the heat-balance power divided by the difference between the reactor outlet and the radiator outlet salt temperatures. This index for full-power operation is also plotted on Figure No. 3 and shows that the heat transfer has been constant.

In summary, the observed performance of the heat exchanger is about half of the predicted value. The power-dependent decrease in heat transfer is caused by temperature dependent changes in the physical properties of the salt. There is no obvious reason for the heat transfer to be so much less than predicted other than discrepancies in the salt physical properties.

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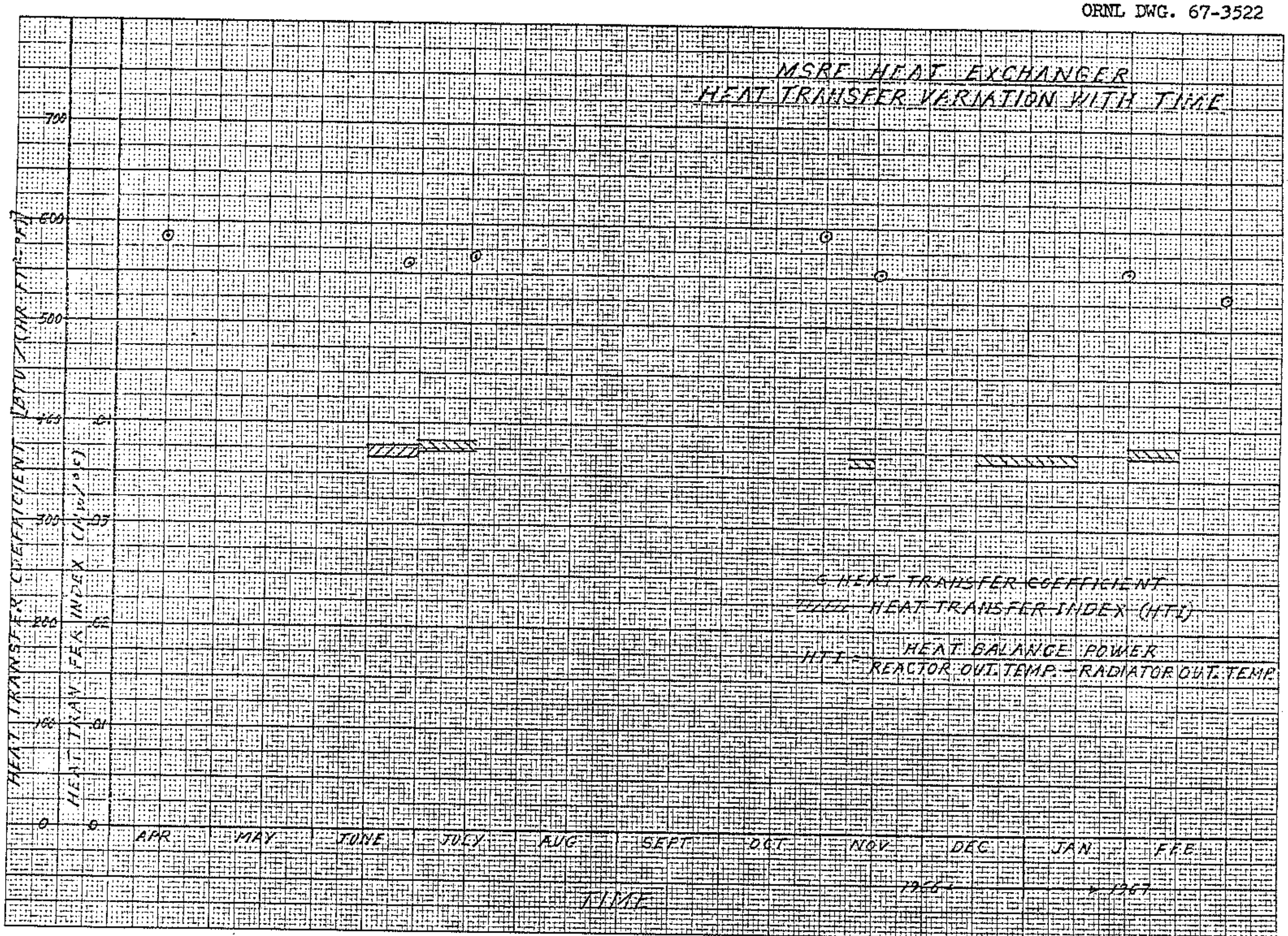

FIGURE - 3

Radiator

The design and instrumentation of the radiator and its enclosure were not intended to provide data for an accurate determination of the radiator heat-transfer coefficient. However, with the data that is available, the heat-transfer coefficient can be estimated for any condition when the radiator doors are fully open. The best evaluation can be made when the bypass damper is also closed because this is the only condition where the air outlet temperature can be measured. The observed performance of the radiator is shown on Figure 4, which is a plot of the heat-transfer coefficient vs radiator air-pressure drop.

The observed heat-transfer coefficient at full power was 38.5 as compared to the corrected design value of 51.5 $\frac{\text{Btu}}{\text{Hr-ft}^2\text{-°F}}$. The following equation was used to calculate the observed heat-transfer coefficients:

$$U = \frac{Q}{A\ \Delta t_{lm}}$$

where U = overall heat transfer

Q = heat transferred, computed from a coolant salt heat balance

A = 706 ft^2 - heat transfer area based on tube OD

Δt_{lm} = applicable log mean temperature difference which is approximately equal to the average salt temperature minus the average air temperature

The inaccuracy in applying this equation lies primarily in the measurement of the outlet air temperature. The air temperature of the total air flow is measured at the top of the radiator stack. This air flow includes the flow through the bypass damper as well as the flow from the annulus blowers. The outlet air temperatures were calculated from an air and salt heat balance around the radiator. The air flows across the radiator were obtained for the two conditions when the bypass damper was fully closed by subtracting the predicted annulus blower flow from the measured total flow rate. The air flow through the radiator with the bypass damper open was calculated from a flow-ΔP relation (Flow = $C\sqrt{\Delta P}$) that was obtained from the two conditions when the bypass damper was fully closed. However, the

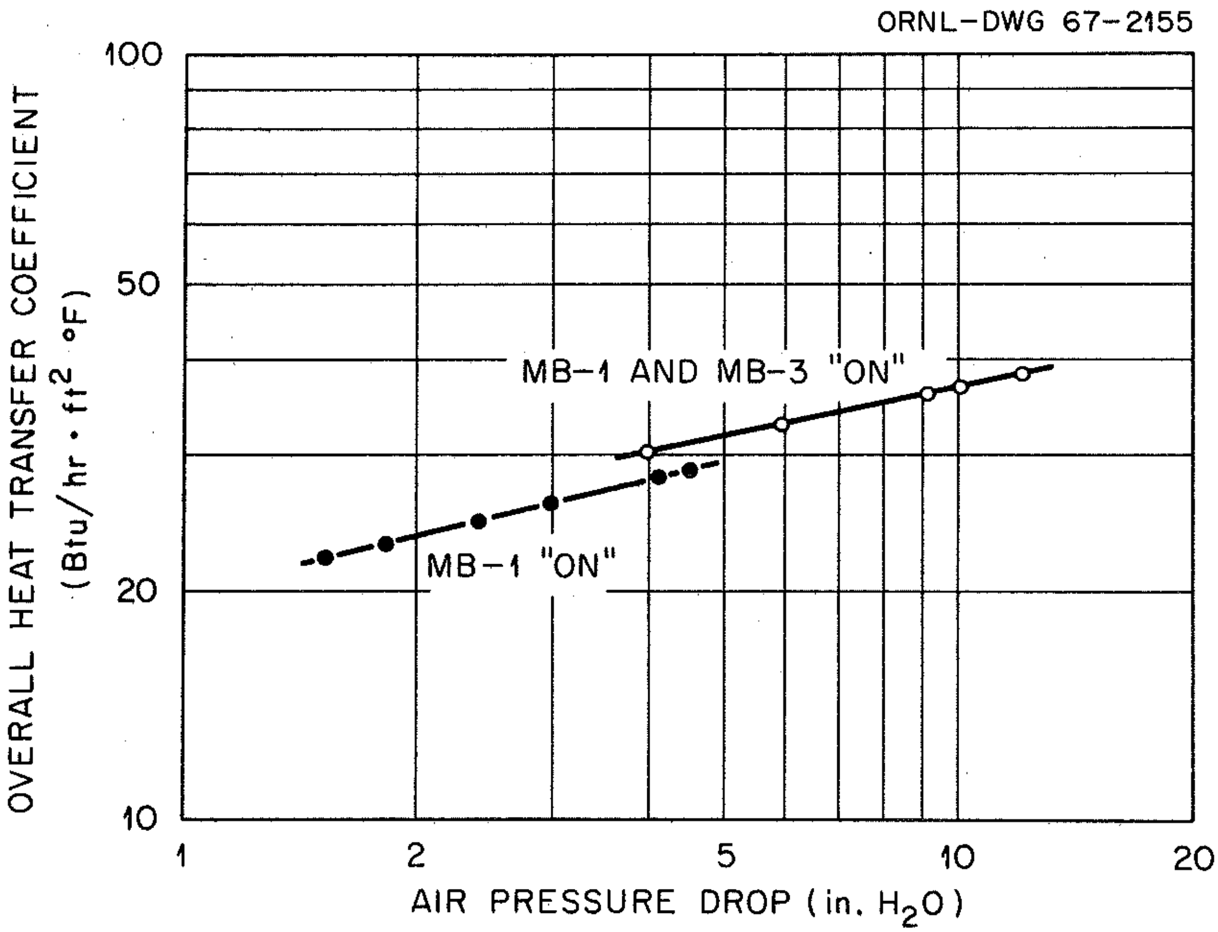

FIGURE 4. OBSERVED PERFORMANCE OF MSRE RADIATOR.

calculated outlet air temperatures for the two cases with the bypass damper closed were significantly below the measured stack outlet temperature. This is an impossibility which indicates an error in either the stack air flow calibration, the temperature measurement, or in the heat balance. We assumed that the heat balances and temperature measurements were correct, and the outlet air temperatures for the various other radiator conditions were corrected to be consistent with the measured values when the bypass damper was closed. As shown in Figure 4, there was an increase in heat-transfer coefficient when the second main blower was turned on. This increase is probably caused by the direct impingement of air from the main blowers on the radiator tubes. The observed heat-transfer coefficient varies with the 0.575 power of the air flow rate. This compares with a theoretical value of 0.6 (Ref. 10).

CONCLUSIONS

Reasons for Discrepancies

After the calculations of the main heat exchanger had been reviewed and found correct, we believed the most likely cause of the reduced heat transfer was errors in the physical property data.

Experimental determinations of the thermal conductivity of MSRE salt were made and reported by Cooke[13]. Up to the present time, the determination included only fuel salt and the value was found to be 0.83 Btu/hr-ft-°F (the value used in the heat-exchanger calculation was 2.75 Btu/hr-ft-°F). The opinion was expressed by Mr. Cooke (personal communication, 7/18/66) that although measurements had not been made on coolant salt, its thermal conductivity would probably be less than the old value by about the same factor as the fuel salt.

Thermal Conductivity Measurements

	Fuel	Coolant
Value used in original design	2.75	3.5
Value determined by Cooke	0.83	None made yet
Estimate for Coolant Salt $\frac{0.83 \times 3.5}{2.75}$	----	1.06

Using the measured value of thermal conductivity for the fuel and ~ 1.0 for the coolant, an overall "U" of 648 was calculated for 7.2 Mw. This compares to a measured "U" of 555 to 590 depending on the particular set of data. Although the agreement is not as close as one might want, it is quite close when compared to the original calculated "U" of ~ 1100.

There is experimental evidence that the reduced heat transfer and the power dependence of the heat transfer are not caused by a gas film on the fuel side of the heat-exchanger tubes. The existing data also indicates that there has been essentially no loss in heat transfer with time. Therefore, the heat-transfer discrepancy that still exists at a reduced degree, may still be caused by errors in the physical property data. However, if the new value of the thermal conductivity had been used and if the same 20% overdesign had been applied, the heat exchanger would have been adequate.

The same general comments are applicable to the air radiator except that the physical properties of air were not evaluated at the correct temperature. However, even if the correct procedure had been followed, a contingency factor of 1.34 would have been required to obtain a radiator of the desired capacity. The more significant design error was the omission of an adequate degree of overdesign.

Considering the heat exchanger and radiator performance together, the system design is consistent. The radiator is capable of rejecting about the same quantity of heat that the heat exchanger can transfer within the existing operational limits in regard to the maximum fuel and minimum coolant temperatures.

In summary, the basic design calculations of both the heat exchanger and the radiator followed the accepted heat-transfer correlations and the only definite reasons for the low heat transfer are the errors in the physical property data. The thermal conductivity of the coolant salt is below the value used in the heat exchanger design and the physical properties of air were evaluated at the wrong temperature in the radiator design. In all other respects the heat exchanger and radiator have performed satisfactorily. There has been no evidence either of gas filming of the heat-exchanger tubes or of a decrease in performance by a buildup of scale.

Methods of Improving Performance

The maximum power capability of the reactor can be increased only by improving the heat-transfer performance of both the radiator and the main heat exchanger. Other than replacement with a larger unit, the heat-exchanger capability can be increased only by increasing the flow rates of the fuel and coolant systems. Increasing the temperature difference between the fuel and coolant system is undesirable because of adverse effects on the thermal cycle and stress rupture life of the reactor system.

A study was completed to determine the maximum flow rates that could be obtained and to determine the increase in heat-transfer performance that would result from these flow increases. The pumps are capable of accepting larger diameter impellers and of operating at a higher horsepower rating so that increased flow is possible. The flow can also be increased by increasing the rotational speed of the pumps with a higher frequency power supply. Slightly higher flow rates can be obtained with the higher rotating speed because the horsepower rating of the drive motors can be increased at the higher speed. The calculations indicated that the maximum possible flow rates for the fuel and coolant systems would be 1530 and 1044 gpm respectively. However, the higher flow rate in the fuel system may be objectionable because of the possibility of increased gas entrainment into the circulating fuel stream. Using the maximum flow rates and the present maximum and minimum system temperatures, the maximum heat-transfer capability of the heat exchanger would be 8.1 Mw as compared to 7.2 Mw at the present flow conditions. While the heat rejection rate of the radiator could be increased by several methods, none of these can be easily accomplished. Additional surface area could be provided by adding additional tubes or by adding some type of fins to the tubes, but either would be difficult and time consuming. Additional air capacity could be provided, but this too, would be a major undertaking. The radiator air flow would have to be increased by a factor of about 1.8 to remove 10 Mw. This would increase the air-pressure drop to ~ 35 in. H_2O and the power requirements of the blowers to about 2900 hp. Neither the present blowers nor the building electrical system is capable of meeting these requirements.

One additional blower could possibly increase the radiator heat removal to the 8.1-Mw level which would be consistent with the maximum possible power of the heat exchanger. However, the blowers would be operating very close to their surge limit, new drive motors might be required to avoid an overload condition, and the existing building electrical system would be unable to supply the third blower.

In conclusion, the difficulties in raising the power capability of the reactor far outweigh any advantages that could be gained from the relatively small power increase that can be reasonably achieved. Since the objectives of the MSRE can be accomplished at the present power level, no attempts to increase the power capability of the MSRE are planned.

APPENDIX

EXPERIMENTAL DETERMINATION OF U

The overall heat-transfer coefficient may be determined by, measuring, simultaneously, a temperature difference (ΔT) and a heat-removal rate (Q), and then applying the equation $Q = UA\Delta T$. At the MSRE, the value of Q is calculated routinely by the on-line computer* by determining the heat removed by the coolant salt. The coolant loop is provided with thermocouple wells and a venturi flowmeter so that temperature differences and the flow rate can be measured accurately. The value of heat transfer area (A) is known so only the value of ΔT remains to be evaluated.

The evaluation of the ΔT across the primary heat exchanger depends upon the measurement of several temperatures which are subject to biases and random errors. A procedure was developed by P. N. Haubenreich for evaluation of the HRT heat exchanger, which minimizes or eliminates the effects of those biases and errors in the calculation of U (Ref. 11). This procedure was modified for use in the MSRE and the development is shown below:

Nomenclature:

T_{ci}		temperature of coolant salt entering the HX
T_{co}		temperature of coolant salt leaving the HX
T_{fi}		temperature of fuel salt entering the HX
T_{fo}		temperature of fuel salt leaving the HX
ΔT_c	=	$T_{co} - T_{ci}$
ΔT_f	=	$T_{fi} - T_{fo}$
$\overline{\Delta T}$	=	log mean temperature difference
F_c		mass flow rate of coolant salt
F_f		mass flow rate of fuel salt
$(Cp)_c$		specific heat of coolant salt
$(Cp)_f$		specific heat of fuel salt

*A BR-340 digital computer

$$Q = F_c(Cp)_c \; \Delta T_c \tag{1}$$

$$Q = UA \; \overline{\Delta T} \tag{2}$$

where

$$\overline{\Delta T} = \frac{\sqrt{\Delta T_f^{\,2} + \Delta T_c^{\,2}}}{\ln \dfrac{T_{fo} + T_{fi} - T_{co} - T_{ci} + \sqrt{\Delta T_f^{\,2} + \Delta T_c^{\,2}}}{T_{fo} + T_{fi} - T_{co} - T_{ci} - \sqrt{\Delta T_f^{\,2} + \Delta T_c^{\,2}}}} \tag{3}$$

using Equations 1 and 2

$$Q = F_c(Cp)_c \; \Delta T_c = UA\overline{\Delta T}$$

$$\lambda \equiv \frac{UA}{F_c(Cp)_c} = \frac{\Delta T_c}{\overline{\Delta T}} \tag{4}$$

and substituting for $\overline{\Delta T}$

$$\lambda = \frac{\Delta T_c}{\sqrt{\Delta T_f^{\,2} + \Delta T_c^{\,2}}} \ln \frac{T_{fo} + T_{fi} - T_{co} - T_{ci} - \sqrt{\Delta T_f^{\,2} + \Delta T_c^{\,2}}}{T_{fo} + T_{fi} - T_{co} - T_{ci} - \sqrt{\Delta T_f^{\,2} + \Delta T_c^{\,2}}} \tag{5}$$

now within the heat exchanger

$$F_c(Cp)_c \; \Delta T_c = F_f(Cp)_f \; \Delta T_f \tag{6}$$

and rearranging

$$\Delta T_f = \frac{F_c \; (Cp)_c}{F_f \; (Cp)_f} \; \Delta T_c = K \; \Delta T_c$$

where

$$K \equiv \frac{F_c\ (Cp)_c}{F_f\ (Cp)_f}$$

substituting ΔT_f Equation 5 and simplifying

$$\lambda = \frac{1}{1 + K^2} \quad \ln \frac{T_{fo} + T_{fi} - T_{co} - T_{ci} + \Delta T_c \sqrt{1 + K^2}}{T_{fo} + T_{fi} - T_{co} - T_{ci} - \Delta T_c \sqrt{1 + K^2}}$$

define

$$\lambda' \equiv \lambda \sqrt{1 + K^2} = \frac{UA}{F_c\ (Cp)_c} \sqrt{1 + K^2} \tag{7}$$

so

$$\lambda' = \ln \frac{T_{fo} + T_{fi} - T_{co} - T_{ci} + \Delta T_c \sqrt{1 + K^2}}{T_{fo} + T_{fi} - T_{co} - T_{ci} - \Delta T_c \sqrt{1 + K^2}} \tag{8}$$

which simplifies to

$$e^{\lambda'} = \frac{T_{fo} + T_{fi} - T_{co} - T_{ci} + \Delta T_c \sqrt{1 + K^2}}{T_{fo} + T_{fi} - T_{co} - T_{ci} - \Delta T_c \sqrt{1 + K^2}}$$

or

$$T_{fo} + T_{fi} - T_{co} - T_{ci} + \Delta T_c \sqrt{1 + K^2} = e^{\lambda'} \left[T_{fo} + T_{fi} - T_{co} - T_{ci} - \Delta T_c \sqrt{1 + K^2} \right] \tag{9}$$

It may be assumed that the values of F_f, F_c, $(Cp)_f$ and $(Cp)_c$ are known or can be calculated and that if the temperatures were known, a value of UA could be calculated directly from Eq. (9). The true temperature, however, is not necessarily the same as the indicated because of measurement errors.

Let us say that

$$T = T' + \theta + X \tag{10}$$

when T and T' are the true and indicated temperatures, respectively, θ is a bias and X is a random error. Substituting 10 into 9 we get

$$\begin{aligned}
&T'_{fo} + T'_{fi} - T'_{co} - T'_{ci} + \Delta T'_c\sqrt{1 + K^2} \\
&\quad + \theta_{fo} + X_{fo} + \theta_{fi} + X_{fi} - \theta_{co} - X_{co} - \theta_{ci} - X_{ci} \\
&\quad + (\theta_{co} + X_{co} - \theta_{ci} - X_{ci})\sqrt{1 + K^2} \\
&\quad = e^{\lambda'}\left[T'_{fo} + T'_{fi} - T'_{co} - T'_{ci} - \Delta T'_c\sqrt{1 + K^2}\right] \\
&\quad + \theta_{fo} + X_{fo} + \theta_{fi} + X_{fi} - \theta_{co} - X_{co} - \theta_{ci} - X_{ci} \\
&\quad - (\theta_{co} + X_{co} - \theta_{ci} - X_{ci})\sqrt{1 + K^2}
\end{aligned} \tag{11}$$

Now since θ's and X's are independent of the temperature function we can differentiate with respect to $(T'_{fo} + T'_{fi} - T'_{co} - T'_{ci} - \Delta T'_c\sqrt{1 + K^2})$ and get:

$$\frac{d\left[T'_{fo} + T'_{fi} - T'_{co} - T'_{ci} + \Delta T'_c\sqrt{1 + K^2}\right]}{d\left[T'_{fo} + T'_{fi} - T'_{co} - T'_{ci} - \Delta T'_c\sqrt{1 + K^2}\right]} = e^{\lambda'} \tag{12}$$

or rewriting

$$\frac{d\left[\alpha(T') + \beta(T')\right]}{d\left[\alpha(T') - \beta(T')\right]} = e^{\lambda'} \tag{12a}$$

where

$$\alpha(T') = T'_{fo} + T'_{fi} - T'_{co} - T'_{ci}$$

$$\beta(T') = \Delta T'_c \sqrt{1 + K^2}$$

Since Eq. (12) is true, the raw temperature data can be plotted with the slope of the resulting curve being equal to $e^{\lambda'}$. At the MSRE, each time a heat balance is taken by the on-line computer, instantaneous values of $\alpha(T')$ and $\beta(T')$ are calculated stored on magnetic tape. All available temperature data can be plotted so that a "best statistical" slope can be determined.

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