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TECHNICAL MEMORANDUM

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TITLE: Mathematical Correlation of Observed Condenser Heat Transfer Variations

ABSTRACT

A mathematical model has been developed for the SNAP-8 condenser which includes the effects of both heat transfer and pressure drop. The model identifies a substantial loss of temperature potential at certain operating conditions, and accounts for the loss of apparent heat transfer capability observed in condenser testing.

KEY WORDS: SNAP-8 condenser, new model, loss of apparent heat transfer capability

APPROVED:

DEPARTMENT HEAD R. W. Marshall, Jr.

R. W. Marshall, Jr.

NOTE: The information in this document is subject to revision as analysis progresses and additional data are acquired.

AEROJET-GENERAL CORPORATION

TM 7994:70-621

Mathematical Correlation

of

Observed Condenser Heat Transfer Variations

J. N. Hodgson

March 1970

CONTENTS

I. INTRODUCTION AND SUMMARY

The recent change in PCS-G statepoint created a need for a better understanding of how the condenser would operate at higher mercury flows and lower condensing pressures. A mathematical model was formed (Reference (1)) which identified the internal pressure characteristics of the condenser. The model identified a potential problem with the condenser at the new PCS-G statepoint due to internal pressure changes which were sufficient to cause "choked flow". A complete map was then generated (Reference (2)) which identified the limits of the condenser and also permitted bypassing the "choked flow" condition by designing with an increased number of condenser tubes.

PCS-1 test data were researched to locate the few isolated cases when the condenser had been operated at conditions at least approaching the new PCS-G statepoint. It was found that a very large drop (40-50%) in apparent heat transfer coefficient occurred as the condenser approached the PCS-G condition. With both the test data and the mathematical analysis indicating poor condenser performance at the PCS-G statepoint, it was decided to undertake a more sophisticated analysis of the condenser which simultaneously analyzed the effects of pressure drop and heat transfer. The earlier analysis used an assumed heat transfer distribution rather than calculating the actual heat transfer. The findings of this recent work are the subject of this memorandum.

The conclusions from the new mathematical model are as follows:

(1) The earlier model which has been used in the SNAP-8 program is only adequate over a limited range of conditions. It is only approximate even at the old PCS-G statepoint and is completely inadequate at the new PCS-G statepoint.

(2) The zero-pressure condition used to identify the "choked flow" state in Reference (1) has been eliminated. The pressure function

in the new model is continuous and always positive. Any effects due to shock waves or other phenomena which may occur at the undefined "choked flow" condition are not considered in the analysis.

(3) A very substantial decrease in mercury pressure and temperature can occur during the condensation process; the decrease is followed by at least a partial pressure and temperature recovery. The local temperature potential along the condenser length decreases and then increases in conjunction with the local pressure and temperature variations.

(4) The decrease in temperature potential makes the overall "apparent" heat transfer coefficient of the condenser low (by 40-50% at the new PCS-G statepoint). The actual local heat transfer coefficient is basically constant; it is the temperature potential which has decreased.

(5) The original model should not be used in evaluating the new PCS-G statepoint.

The new mathematical model makes it possible to more accurately evaluate the performance of the condenser. It is now possible to predict the performance at any set of conditions. The model is also directly applicable as a design tool if a condenser redesign is ever required. It is possible to design a condenser without a degrading "apparent" heat transfer coefficient and with an overall pressure recovery which would allow the turbine to operate at a lower back pressure without compromising mercury pump NPSH.

II. ORIGINAL MATHEMATICAL MODEL

The original mathematical model is based on the assumption of a constant mercury temperature during the condensation process. For a constant mercury temperature, the condenser performance is defined by

$$T_{sat} = \frac{\dot{W}_H \lambda X}{(C_{P_N} \dot{W}_N - C_{P_H} \dot{W}_H)} \left[\frac{1}{1 - e^{-\frac{UA}{C_{P_N} \dot{W}_N}}} \right] + T_{ni} \qquad (1)$$

less than the actual local heat transfer coefficient. Therefore, when the original model is used to calculate the condenser performance, the model predicts better performance than can actually occur. The model is only correct when the operating conditions are such that a constant mercury condensing temperature exists.

III. NEW MATHEMATICAL MODEL

The new mathematical model takes into account the changing conditions within the condenser. The changes in mercury pressure, temperature, quality, pressure drop and NaK temperature and flow are all evaluated simultaneously. The total heat transfer is the sum of the incremental contributions.

The pressure drop portion of the analysis was presented earlier in Reference (1). The heat transfer analysis has been added to the pressure drop analysis by removing the vapor-quality profiles assumed in the pressure-drop analysis and replacing them with calculated values of vapor quality. The values of vapor quality are found by computing the heat transferred at the local temperature potential.

The new model uses two computer programs which are shown in Figures (2) and (3). Figure (2) is the principle program in the analysis. This program evaluates the temperature, pressure, and vapor quality profiles for any set of input conditions. A solution is reached when the input variables result in a vapor quality distribution which just extends to the end of the condensing length. Incompatible input data result in (1) complete condensation before the interface position is reached, (2) incomplete condensation in the available condensing length, or (3) a zero or negative mercury pressure. The derivation of the analysis is presented in the Appendix.

The computer program of Figure (3) simply evaluates the effective overall heat transfer coefficient for a given set of data. The input data are the data found in the corresponding solution of the main program (Figure (2)).

where T_{sat} = condensing temperature, ^{o}F

$\dot{W}_H$ = mercury flow, lb/hr

$\dot{W}_N$ = NaK flow, lb/hr

λ = heat of vaporization, BTU/lb

X = inlet quality

C_{P_H} = liquid mercury specific heat, BTU/lb - ^{o}F

C_{P_N} = NaK specific heat, BTU/lb - ^{o}F

U = overall heat transfer coefficient, BTU/hr-ft^2 -^{o}F

A = condensing area, ft^2

T_{ni} = NaK inlet temperature, ^{o}F

Equation (1) has routinely been used throughout the SNAP-8 program for computing the theoretical condensing temperature or, in a rearranged form, for computing the overall heat transfer coefficient, U, from the test data. It has always been assumed that U is essentially a constant for all conditions. Typical mercury and NaK temperature profiles for the original mathematical model are shown schematically in Part (a), Figure (1).

For comparison, Parts (b) and (c) of Figure (1) show, schematically, what the condenser temperature profiles are actually like. Part (b) shows the condenser performance at its design condition (old PCS-G statepoint). The mercury temperature does vary, but not too significantly. But in Part (c), profiles are shown that are typical of operation at the new PCS-G statepoint. Here it is evident that the condensing temperature varies appreciably, so that the average temperature potential is significantly less than it would be if a constant mercury temperature existed, as is assumed in the original mathematical model. Consequently, if the condenser is operated under circumstances where large mercury temperature variations occur, the effective overall heat transfer coefficient is much

IV. COMPUTER RESULTS

The new mathematical model has been used to evaluate the quality and temperature distributions for a variety of condenser operating conditions. Figures (4) through (9) show data at a mercury flow of 12000 lb/hr for condensing lengths of 15, 25, and 35 in. at condensing pressures of 20 psia and 8 psia. At the high condensing pressure (20 psia) the mercury temperature remains virtually constant during the condensation process. For these conditions, the original mathematical model would have been adequate. However, when the condensing pressure is dropped to the lower value (8 psia) the variations in mercury temperature become pronounced. Now condensing length becomes important. At low condensing lengths, there is an appreicable rise in mercury temperature during the condensation process, whereas at long condensing lengths the mercury temperature drops sharply. These profiles clearly show the variations of temperature potential that can occur.

Figures (10) through (15) are the same as Figures (4) through (9) except the mercury flow is 15000 lb/hr, typical of the new PCS-G statepoint. The same remarks apply to these data at 15000 lb/hr as at 12000 lb/hr, except that the trends are more pronounced. At the new PCS-G statepoint (15000 lb/hr mercury, 8 psia condensing pressure, 30 in. condensing length) the effective overall heat transfer coefficient (based on the original model) is 40-50% less than the local coefficient, all because of the loss of temperature potential. Obviously, the original mathematical model should not be used as a design tool in PCS-G analysis as long as the statepoint requires the condenser to be so far off design.

V. COMPARISON WITH TEST DATA

Part of the motivation for developing a more extensive condenser model was the observation that the PCS-1 test data indicated an effective condenser overall heat transfer coefficient that dropped off markedly at some test conditions. It was observed that the poor condenser performance always matched those operating conditions for which a "choked flow" phenomenon was most likely. The new model has now substantiated the trends observed in the test data.

Figures (16) and (17) present values of effective overall heat transfer coefficient as calculated with the new model. Enclosure (16) is for a mercury flow of 12000 lb/hr and Enclosure (17) is for a mercury flow of 15000 lb/hr. The data show a large decline in the coefficient as condensing pressure is lowered, provided a large condensing length is used. For short condensing lengths, little change occurs.

PCS-1 test data are included in Figure (16) for comparison. The agreement with the new model is reasonably good.

Another very significant phenomenon has been observed in recent PCS-1 testing. The condenser has been analyzed during the early portion of a run before the mercury flow and condensing pressure were raised to their nominal values. Specifically, the mercury flow was 6000 lb/hr, and the condensing pressure was 2.0 psia. The unexpected phenomenon that was observed was that there was no indicated NaK temperature profile. The entire series of NaK thermocouples (which cover the condenser length from the bottom up to within 10 in. of the mercury inlet) all recorded the NaK inlet temperature even though the interface was known to be near the bottom of the condenser. This means that the majority of the condenser was not in use, and the entire condensation process was occurring in the top 10 in., or less, of the condenser. Probably this is the truest example of actual "choked flow" that has been experienced. The prevailing mercury and NaK conditions should have resulted in a condensing pressure of much less than 2.0 psia; but instead, the condensation occurred at a higher temperature (and pressure) in a shorter length, leaving the majority of the condenser unused.

A future improvement in the mathematical model would be to identify the exact conditions which determine whether the condenser will have a large unused area (as above) or whether it will simply operate at a lower condensing pressure with reduced temperature potential, but using the entire condensing length. It appears that a gross loss of condensing area is restricted to operation at very low condensing pressures (perhaps 2 psia). PCS-1 operation at condensing pressures as low as 8.0 psia has shown no loss of condensing area.

An example of the loss of condensing area that occurs at very low condensing pressure is shown in Figure (18). The data show operation at 6000 lb/hr mercury flow at a condensing pressure of 2.0 psia. The PCS-1 data show a completely flat NaK temperature profile with an available condensing length of 35 inches. The mathematical model shows the same results; the entire condensation is shown in the first three inches of the condenser.

It appears that the new condenser model is reasonably accurate and provides a new basis for predicting condenser performance.

Forthcoming tests in PCS-1 are specifically planned which evaluate the condenser at conditions which are far off-design. Further correlation of test data with the new model will be accomplished following the PCS-1 testing.

REFERENCES

(1) <u>SNAP-8 Condenser Performance at Modified PCS-G Statepoint</u>
J. N. Hodgson, Memo 4903-70-1211, 9 January 1970
   (a) Supplement #1, Memo 4903-70-1218, 23 January 1970
   (b) Supplement #2, Memo 4903-70-1222, 9 February 1970

(2) <u>Design Parameters for SNAP-8 Condenser Redesign</u>
J. N. Hodgson, Memo 4903-70-1217, 23 January 1970

EFFECT OF OPERATIONAL MODE
ON TEMPERATURE PROFILES

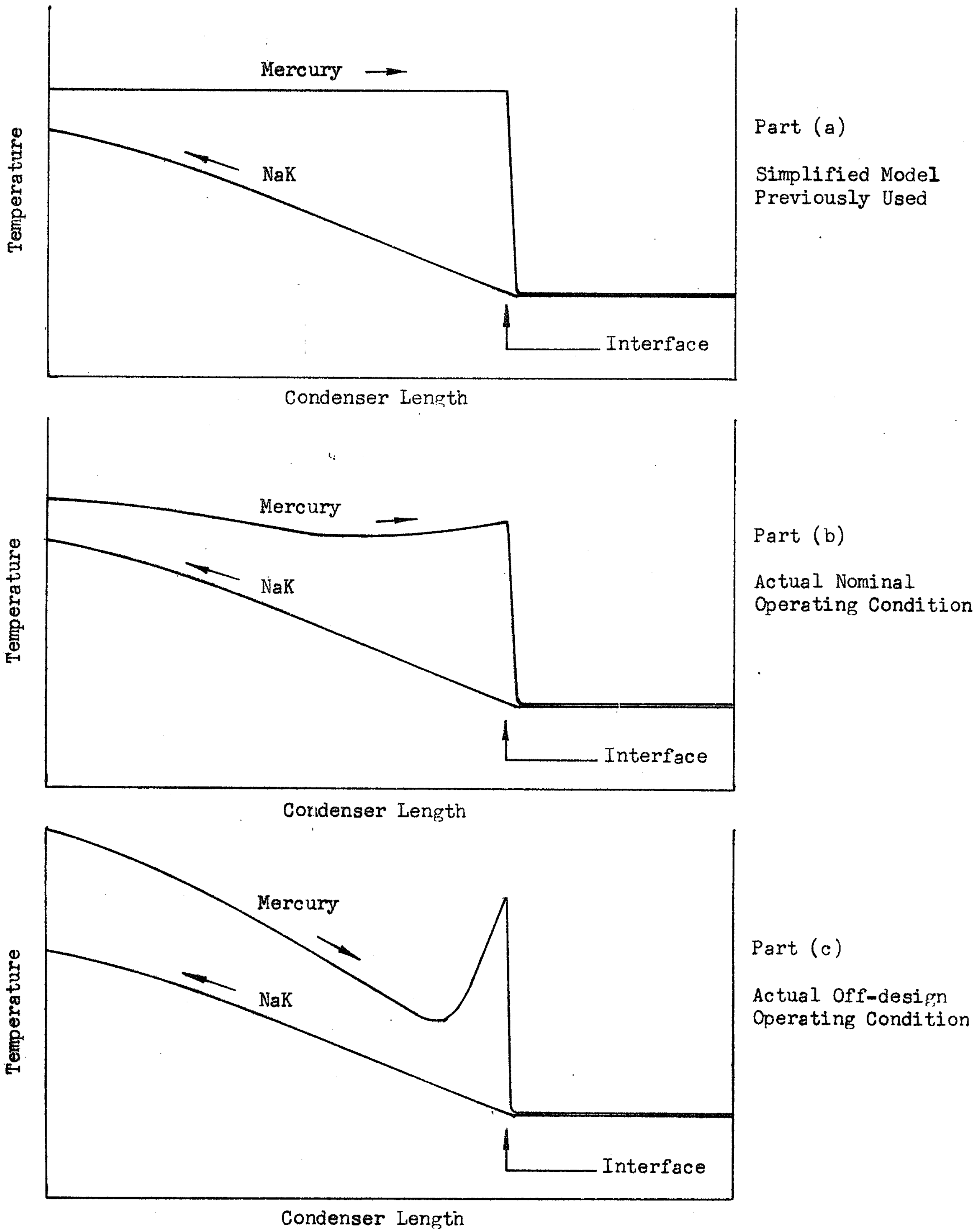


Figure 1

CONDENSER PERFORMANCE ANALYSIS

COMBINED HEAT TRANSFER AND PRESSURE DROP

```
 1 FORMAT(4X,'DIST',2X,'ULOCAL',3X,'PLOCAL',2X,'TLOCAL',3X,'TNAK',4X,'DPTOT',4X,'QUAL')
 2 PRINT 1
 3 FORMAT(4X,'(IN)',1X,'(B/HF2F)',2X,'(PSIA)',4X,'(F)',5X,'(F)',3X,'(PSIA)',2X,'(PCENT)')
 4 PRINT 3
 5 QUALIN=0.92
 6 K=-1
 7 N=-1
 8 J=-1
 9 DPM=0
10 DPTTP=0
11 DPTM=0
12 TINLET=(13050.0/(14.19-ALOG(PINLET)))-460.0
13 RHOGAS=0.01085*PINLET/(TINLET+460.0)
14 VEL1=2.09E-5*FHGTOT/RHOGAS
15 VEL2=7.23E-6*FHGTOT/RHOGAS
16 DPENT=RHOGAS*((VEL1-VEL2)**2.0+0.5*VEL2**2.0)/(2.0*386.0)
17 PLOCAL=PINLET-DPENT
18 DIST=-DELTA
19 DIST=DIST+DELTA+0.00001
20 TLOCAL=(13050.0/(14.19-ALOG(PLOCAL)))-460.0
21 DLOCAL=0.460-0.0065*DIST
22 HVAP=128.9-0.00825*(TLOCAL-400.0)
23 TNAKO=((FHGTOT/(FNAK*0.21))*((HVAP*QUALIN)+(0.0325*(TINLET-TNAKIN))))+TNAKIN
24 N=N+1
25 IF(N-1)26,26,27
26 TNAK=TNAKO
27 J=J+1
28 IF(J-1)29,29,30
29 ACOND1=0
30 ACOND2=0.70*DIST-0.00507*DIST**2.0
31 DELTAA=ACOND2-ACOND1
32 ACOND1=ACOND2
33 DELTAQ=ULOCAL*DELTAA*(TLOCAL-TNAK)
34 TNAK=TNAK-(DELTAQ/(FNAK*0.21))
```

(CONTINUED)

Figure 2

```
35 DELTAX=DELTAQ/(FHGTOT*HVAP)
36 K=K+1
37 IF(K-1)38,38,39
38 QUAL=QUALIN
39 QUAL=QUAL-DELTAX
40 RHOLIQ=0.463-4.92E-5*(TLOCAL-600.0)
41 RHOGAS=0.01085*PLOCAL/(TLOCAL+460.0)
42 AVAP=0.785*(DLOCAL)**2.0/(1.0+(RHOGAS*((1.0-QUAL)/QUAL)/RHOLIQ))
43 VVAP=FHGTOT*QUAL/(3600.0*73.0*AVAP*RHOGAS)
44 VISVAP=0.132+0.00022*(TLOCAL-600.0)
45 VISLIQ=2.09-0.0012*(TLOCAL-600.0)
46 RENUM=RHOGAS*VVAP*DLOCAL*3600.0*12.0/VISVAP
47 FFAC=0.046/(RENUM**0.2)
48 DPDLG=FFAC*(VVAP)**2.0*RHOGAS/(193.0*DLOCAL)
49 XTT=(((1.0-QUAL)/QUAL)**0.9)*((RHOGAS/RHOLIQ)**0.5)*((VISLIQ/VISVAP)**0.1)
50 PHI=1.30+(5.82*XTT)-(3.20*XTT**2.0)+(0.283*XTT**3.0)
51 DPDLTP=PHI**2.0*DPDLG
52 DPTP=DPDLTP*DELTA
53 PM2A=FHGTOT*VVAP*QUAL/(386.0*3600.0*73.0)
54 A2=0.785*DLOCAL**2.0
55 IF(DIST-DELTA)57,56,56
56 DPM=(PM2A-PM1A)/((A1+A2)/2.0)
57 PM1A=PM2A
58 A1=A2
60 DPTTP=DPTTP+DPTP
61 DPTM=DPTM+DPM
62 DPTOT=DPTTP+DPTM+DPENT
63 PRINT 64,DIST,ULOCAL,PLOCAL,TLOCAL,TNAK,DPTOT,QUAL
64 FORMAT(2F8.0,F8.2,2F8.0,2F8.2)
65 PLOCAL=PLOCAL-DPTP-DPM
66 IF(DIST-CONDL)19,67,67
67 STOP
LISTING COMPLETED
```

Figure 2 (cont.)

CONDENSER OVERALL PERFORMANCE

```
  7 FORMAT(2X,'FHGTOT',4X,'FNAK',2X,'TNAKIN',3X,'TNAKO',2X,'PINLET',4X,'TDLM',2X,'QUALIN',2X,'TINLET',3
 14 PRINT  7
 16 QUALIN=0.92
 21 TINLET=(13050.0/(14.19-ALOG(PINLET)))-460.0
 22 RHOGAS=0.01085*PINLET/(TINLET+460.0)
 23 VEL1=2.09E-5*FHGTOT/RHOGAS
 24 VEL2=7.23E-6*FHGTOT/RHOGAS
 25 DPENT=RHOGAS*((VEL1-VEL2)**2.0+0.5*VEL2**2.0)/(2.0*386.0)
 26 PIN=PINLET-DPENT
 27 TINLET=(13050.0/(14.19-ALOG(PIN)))-460.0
 28 TNINT=(((FHGTOT*0.032)*(TINLET-TNAKIN-2.0))/(FNAK*0.21))+TNAKIN
 35 HVAP=128.0-0.00825*(TINLET-400.0)
 42 TNAKO=((FHGTOT/(FNAK*0.21))*((HVAP*QUALIN)+(0.0325*(TINLET-TNAKIN))))+TNAKIN
 49 TDLM=((TINLET-TNAKO)-(TINLET-TNINT))/ALOG((TINLET-TNAKO)/(TINLET-TNINT))
 56 QHG=FHGTOT*HVAP*QUALIN
 63 ACOND=0.70*CONDL-0.00507*CONDL**2.0
 70 UOVRAL=QHG/(TDLM*ACOND)
 77 PRINT 84,FHGTOT,FNAK,TNAKIN,TNAKO,PINLET,TDLM,QUALIN,TINLET,ACOND,UOVRAL
 84 FORMAT(4F8.0,F8.2,F8.0,F8.2,F8.0,F8.2,F8.0)
 91 STOP
LISTING COMPLETED
```

Figure 3

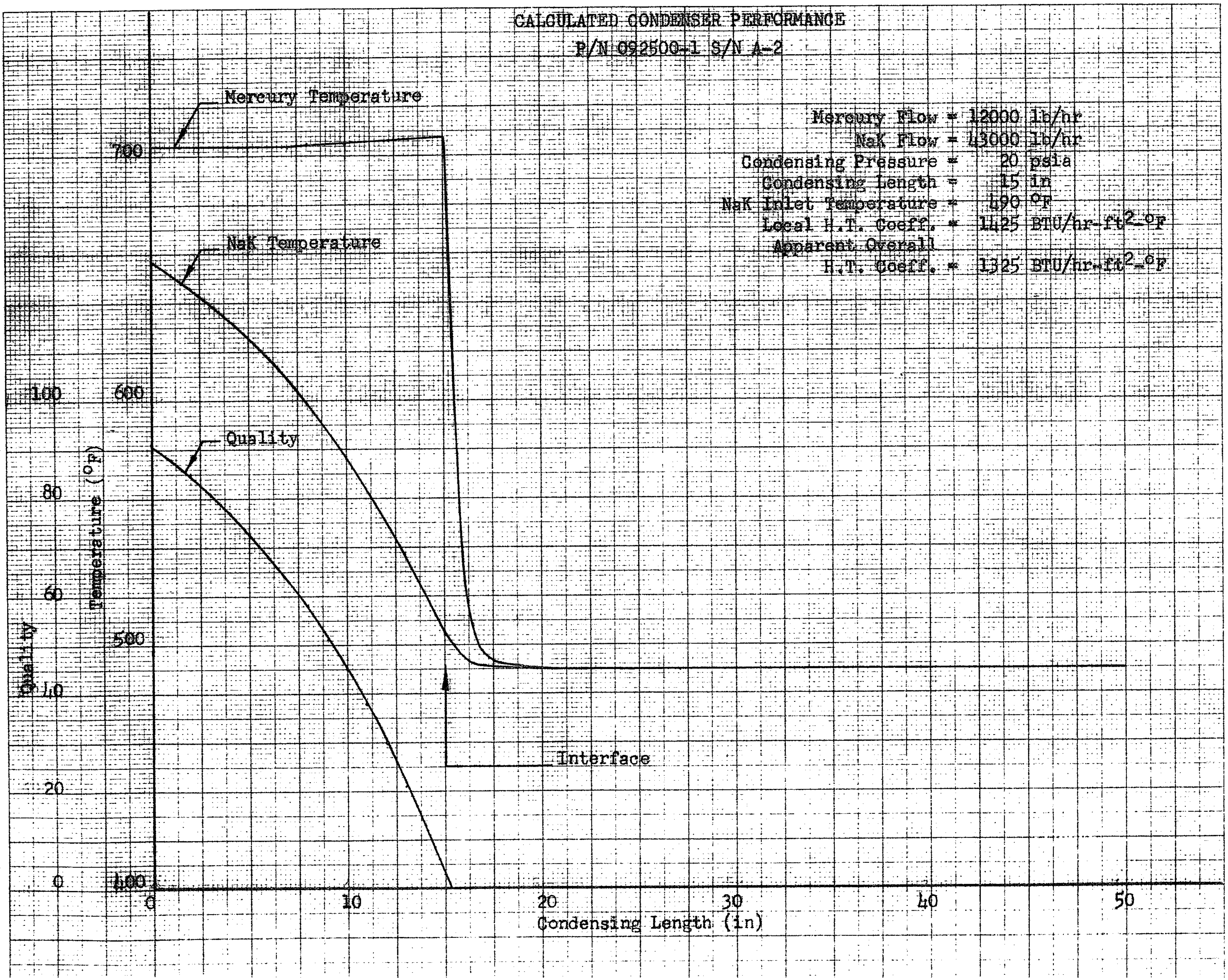

CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Temperature
NaK Temperature
Quality
Mercury Flow = 12000 lb/hr
NaK Flow = 43000 lb/hr
Condensing Pressure = 20 psia
Condensing Length = 15 in
NaK Inlet Temperature = 490 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1325 BTU/hr-ft2-°F
Interface
Temperature (°F)
Quality
700
600
500
400
100
80
60
40
20
0
0
10
20
30
40
50
Condensing Length (in)

Figure 4

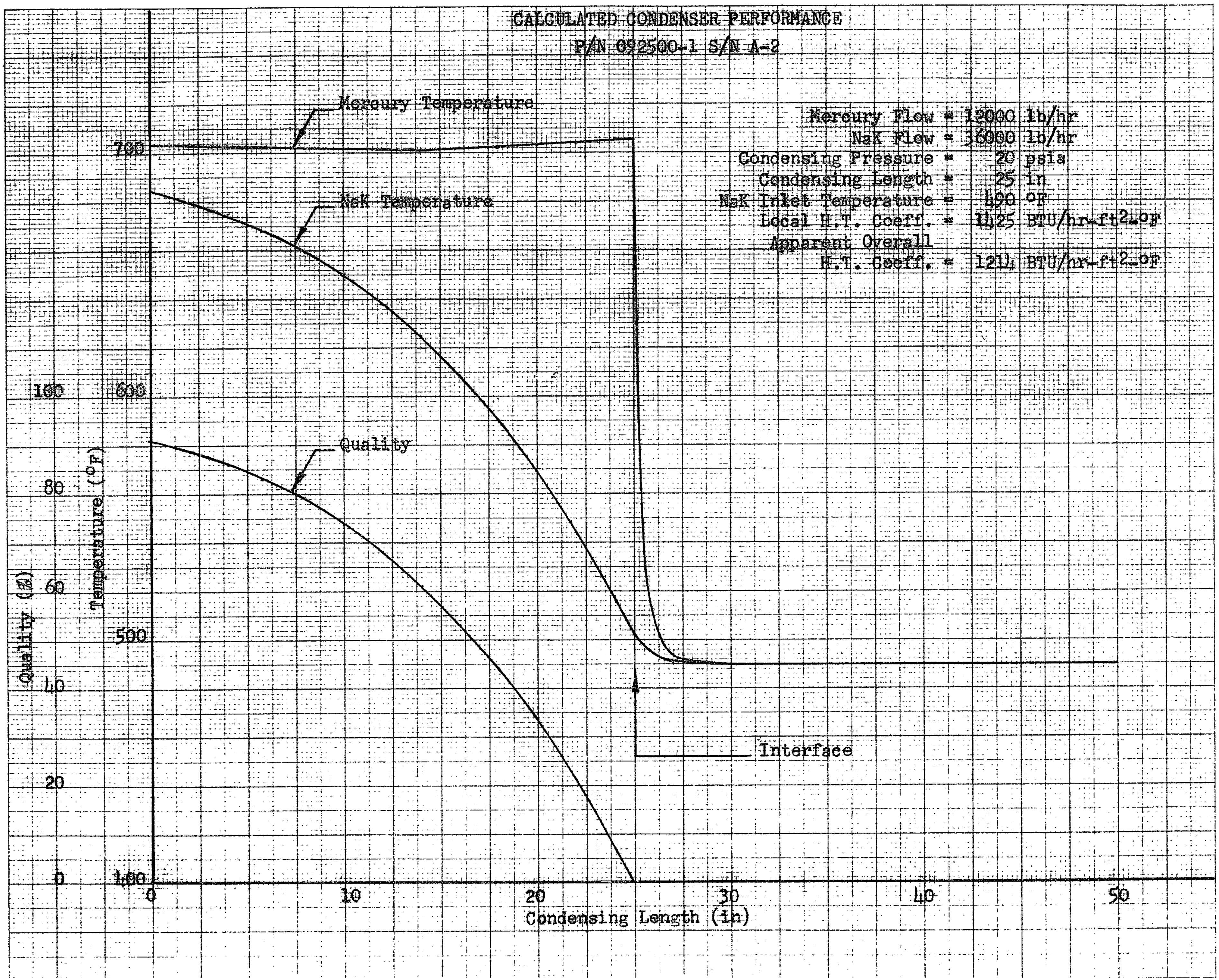

CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Temperature
NaK Temperature
Quality
Interface
Mercury Flow = 12000 lb/hr
NaK Flow = 36000 lb/hr
Condensing Pressure = 20 psia
Condensing Length = 25 in
NaK Inlet Temperature = 490 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1214 BTU/hr-ft2-°F
Quality (%)
Temperature (°F)
700
600
500
400
100
80
60
40
20
0
0
10
20
30
40
50
Condensing Length (in)

Figure 5

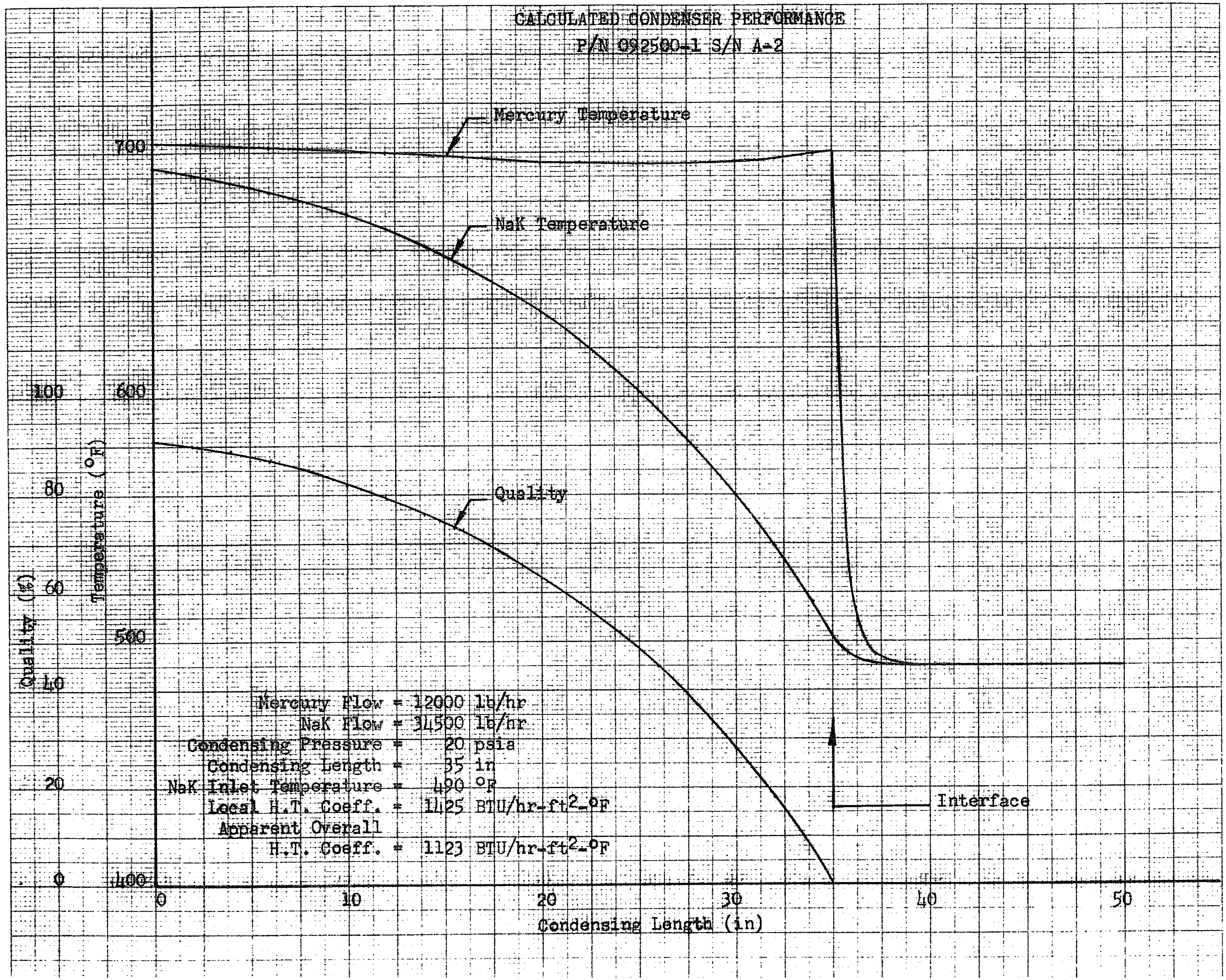

CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Temperature
NaK Temperature
Quality
Interface
Temperature (°F)
Quality (%)
700
600
500
400
100
80
60
40
20
0
0
10
20
30
40
50
Condensing Length (in)
Mercury Flow = 12000 lb/hr
NaK Flow = 34500 lb/hr
Condensing Pressure = 20 psia
Condensing Length = 35 in
NaK Inlet Temperature = 490 °F
Local H.T. Coeff. = 1425 BTU/hr-ft²-°F
Apparent Overall
H.T. Coeff. = 1123 BTU/hr-ft²-°F

Figure 6

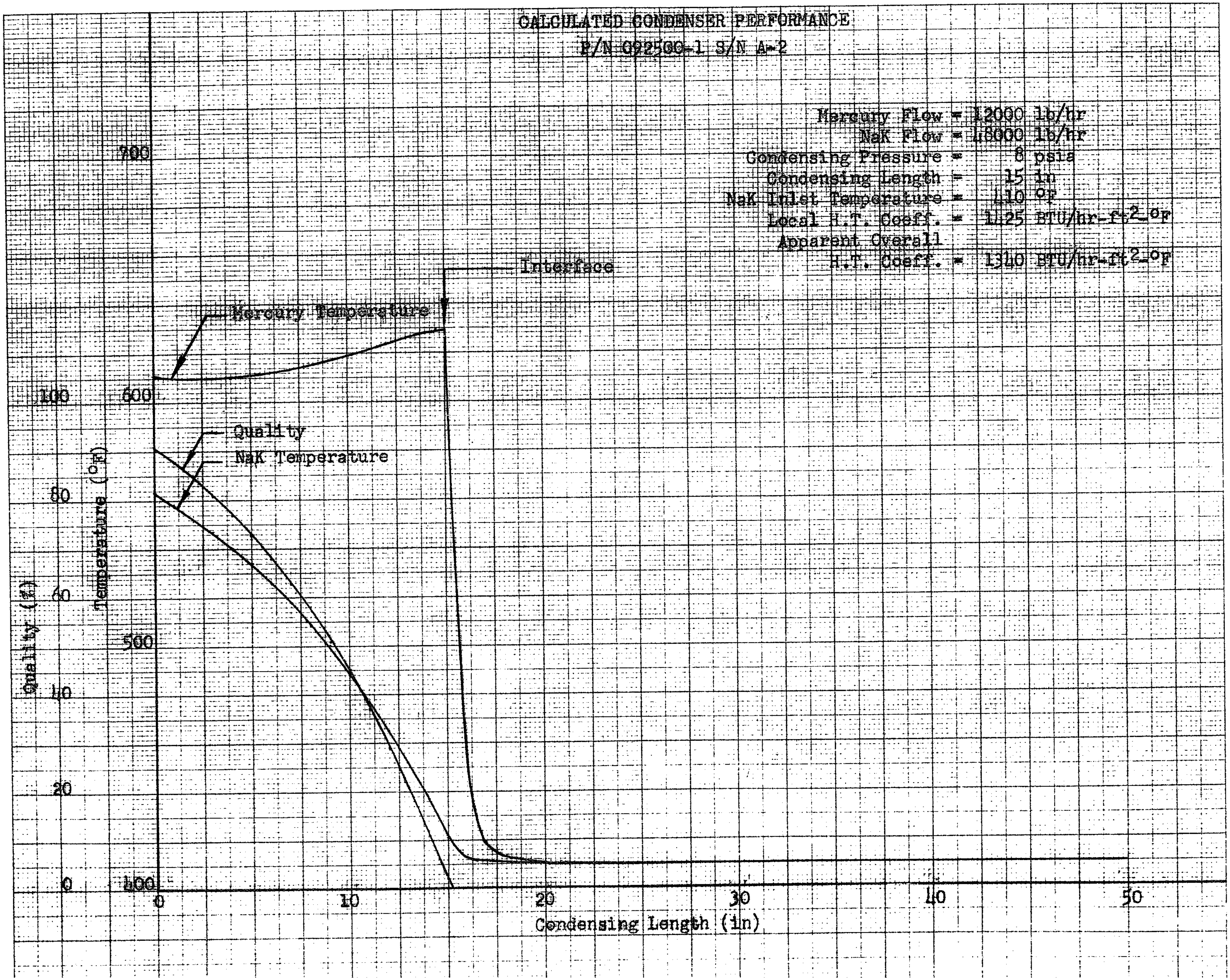

CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Flow = 12000 lb/hr
NaK Flow = 48000 lb/hr
Condensing Pressure = 8 psia
Condensing Length = 15 in
NaK Inlet Temperature = 410 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1340 BTU/hr-ft2-°F
Interface
Mercury Temperature
Quality
NaK Temperature
Quality (%)
Temperature (°F)
700
600
500
400
100
80
60
40
20
0
0
10
20
30
40
50
Condensing Length (in)

Figure 7

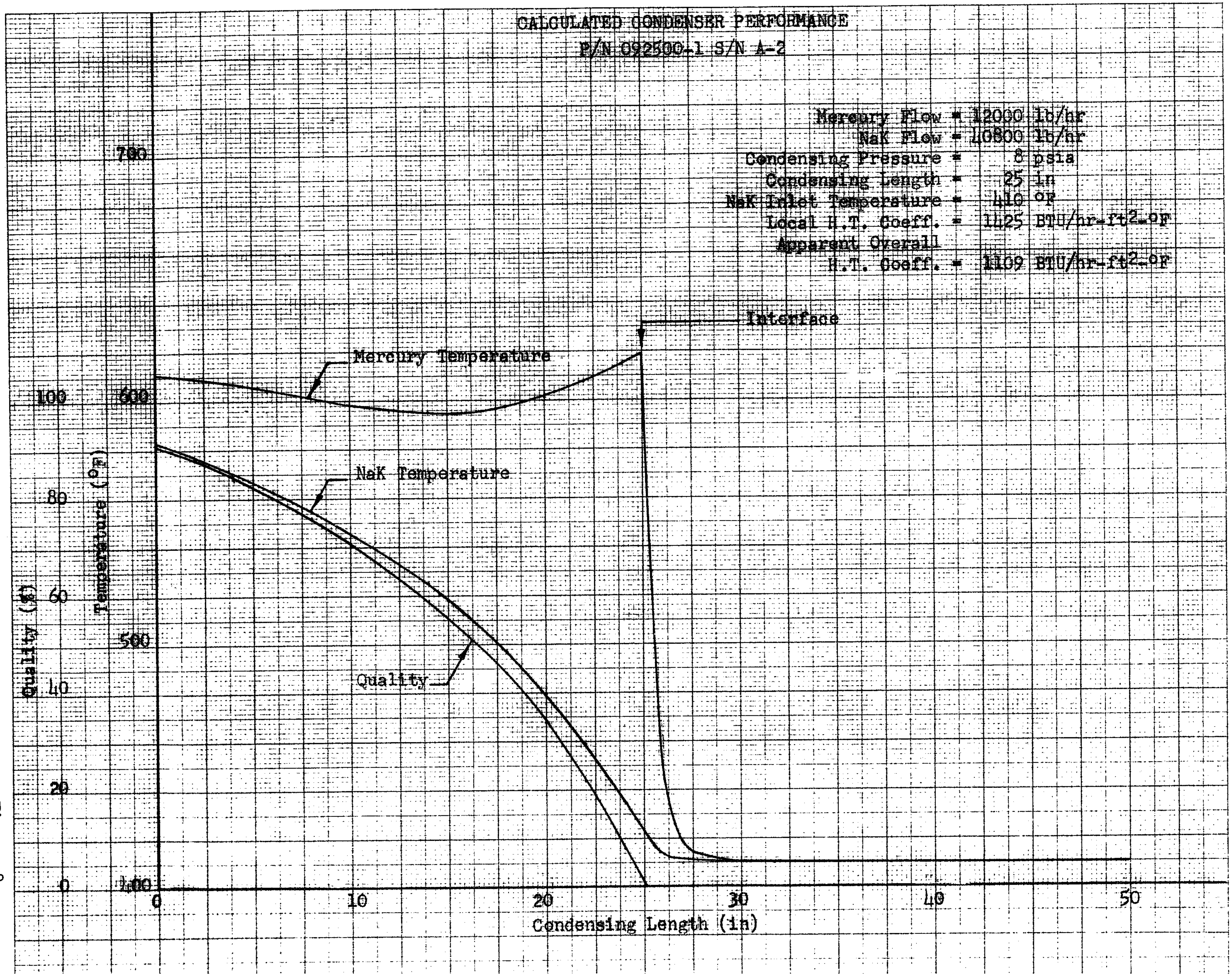
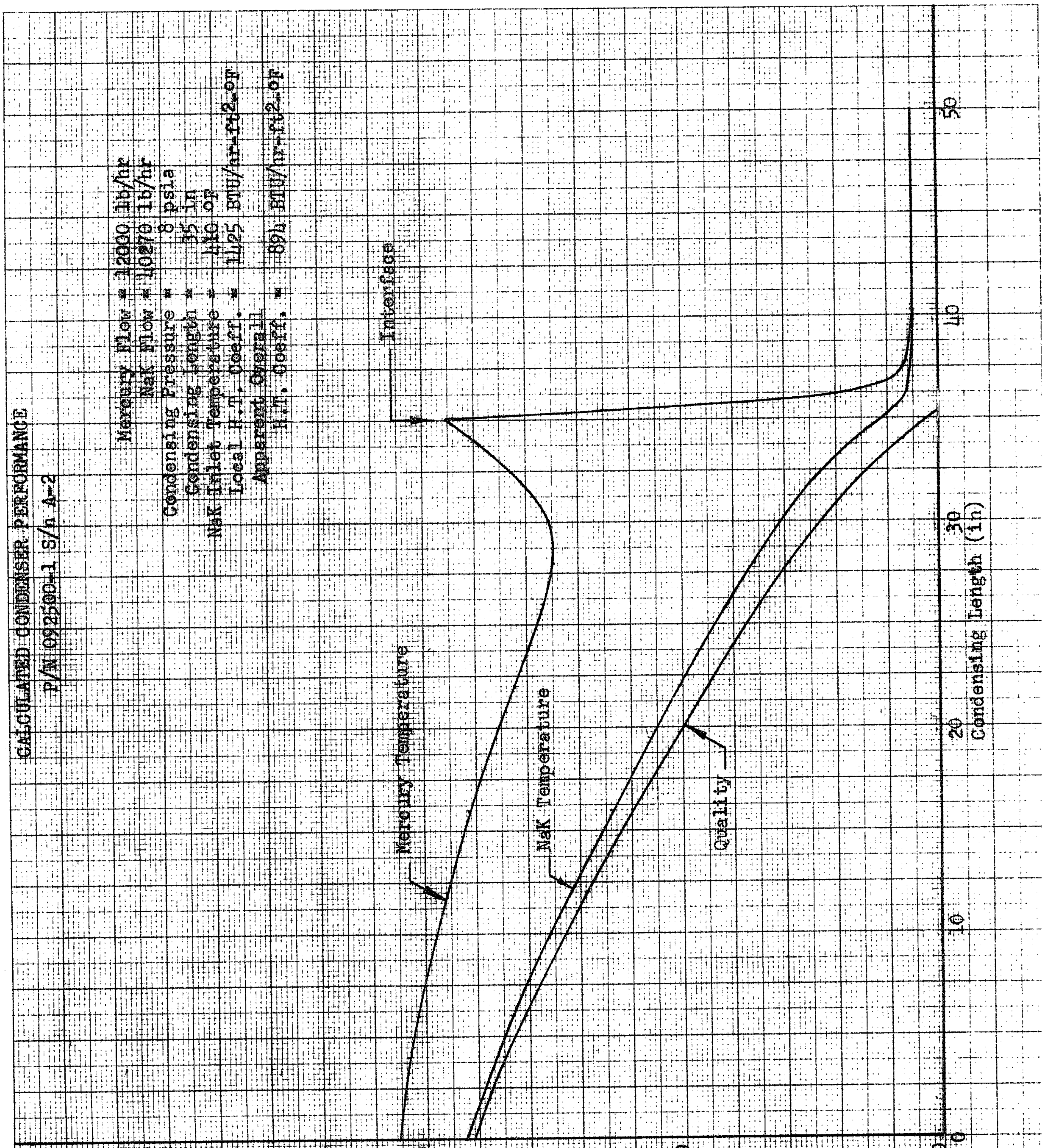

CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Flow = 12000 lb/hr
NaK Flow = 10800 lb/hr
Condensing Pressure = 8 psia
Condensing Length = 25 in
NaK Inlet Temperature = 410 °F
Local H.T. Coeff. = 1125 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1109 BTU/hr-ft2-°F
Interface
Mercury Temperature
NaK Temperature
Quality
Quality (%)
Temperature (°F)
700
600
500
400
100
80
60
40
20
0
10
30
50
Condensing Length (in)

Figure 8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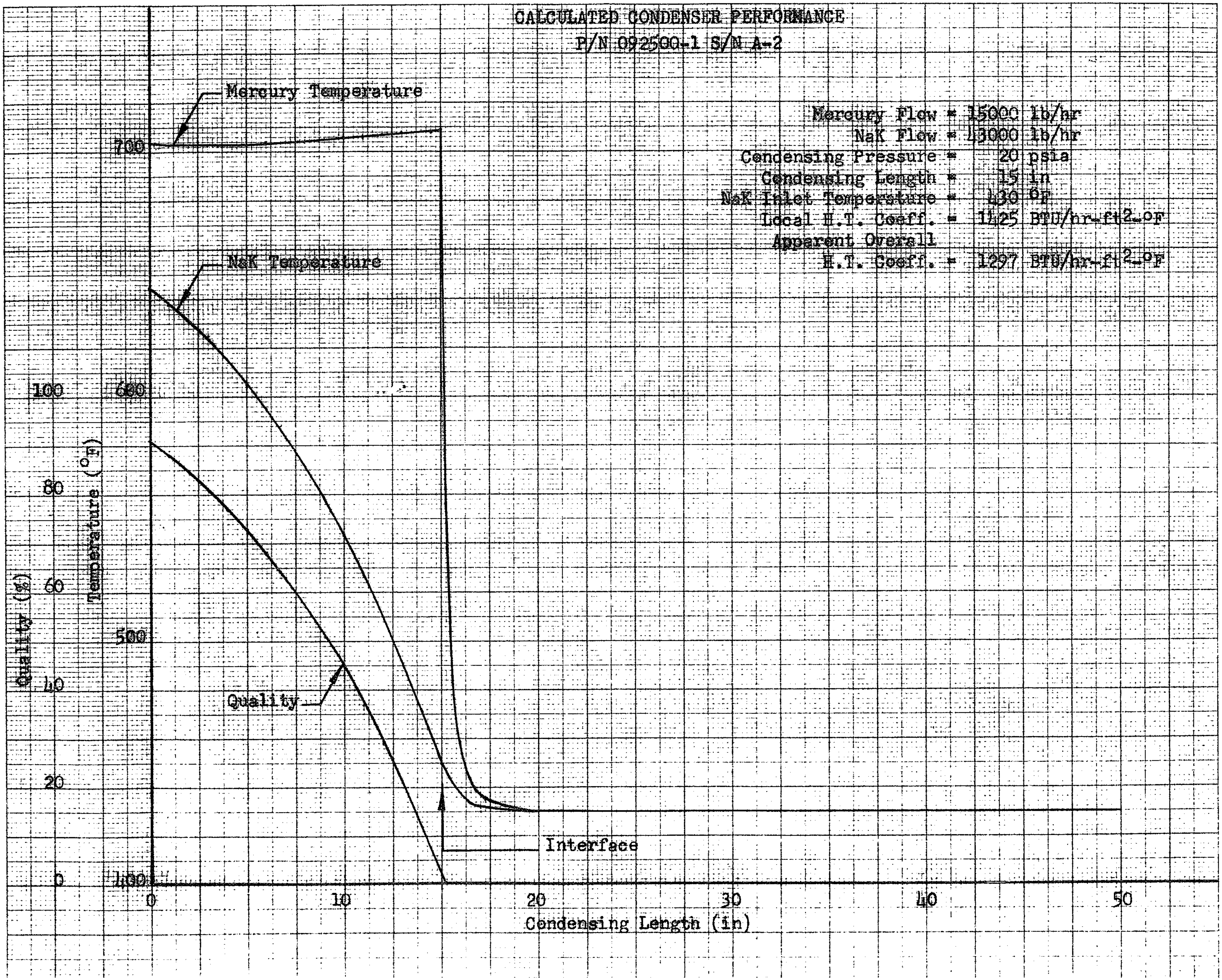
P/N 092500-1 S/n A-2
Mercury Flow = 12000 lb/hr
NaK Flow = 10270 lb/hr
Condensing Pressure = 8 psia
Condensing Length = 35 in
NaK Inlet Temperature = 410 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall H.T. Coeff. = 894 BTU/hr-ft2-°F
Interface
Mercury Temperature
NaK Temperature
Quality
Temperature (°F)
700
600
500
400
Quality (%)
100
80
60
40
20
0
0
10
20
30
40
50
Condensing Length (in)

Figure 9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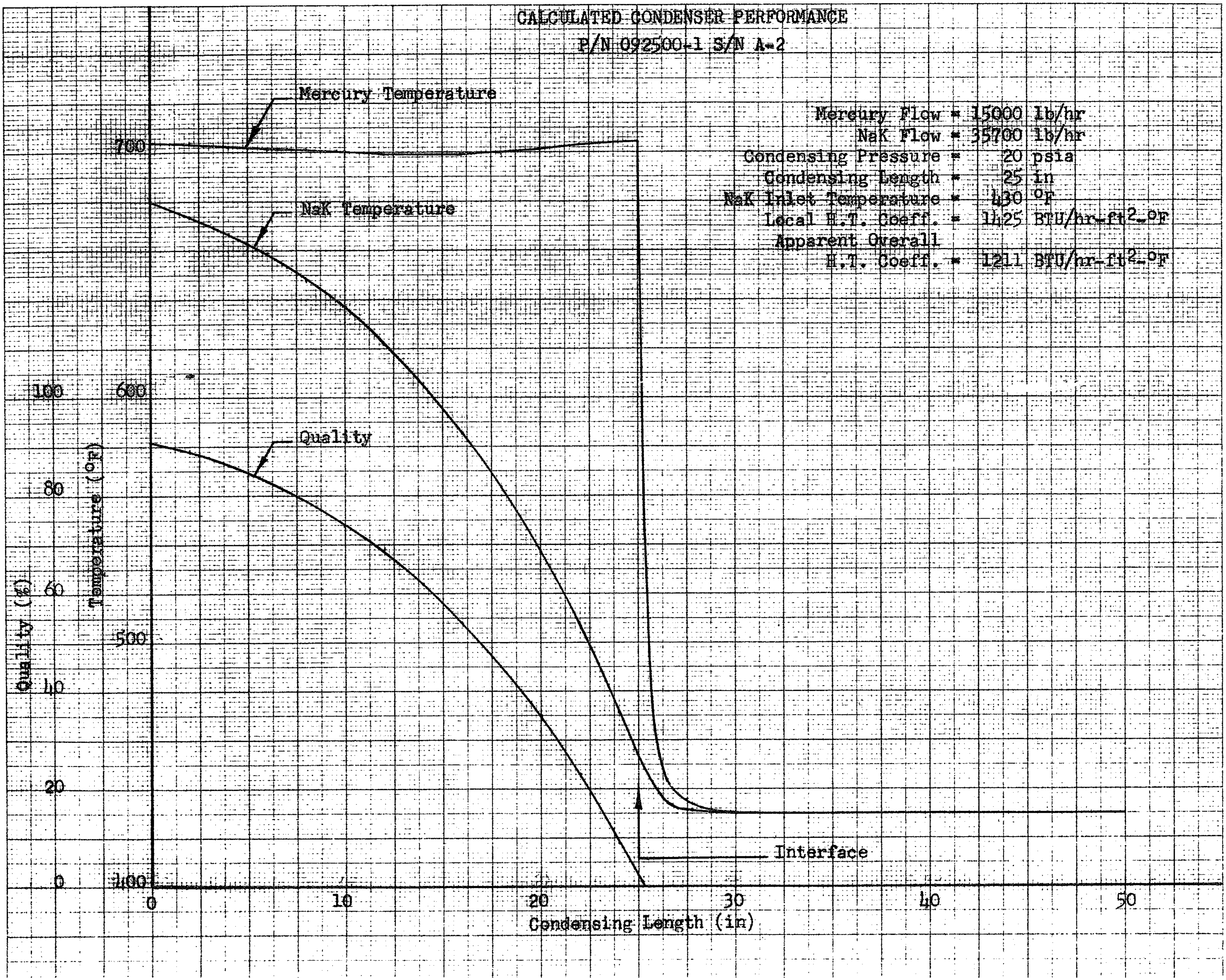
P/N 092500-1 S/N A-2
Mercury Flow = 15000 lb/hr
NaK Flow = 13000 lb/hr
Condensing Pressure = 20 psia
Condensing Length = 15 in
NaK Inlet Temperature = 430 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1297 BTU/hr-ft2-°F
Mercury Temperature
NaK Temperature
Quality
Interface
Quality (%)
Temperature (°F)
100
80
60
40
20
0
700
600
500
400
0
10
20
30
40
50
Condensing Length (in)

Figure 10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NaK Temperature
Quality
Mercury Flow = 15000 lb/hr
NaK Flow = 35700 lb/hr
Condensing Pressure = 20 psia
Condensing Length = 25 in
NaK Inlet Temperature = 430 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1211 BTU/hr-ft2-°F
Interface
Temperature (°F)
Quality (%)
Condensing Length (in)
700
600
500
400
100
80
60
40
20
0
10
20
30
40
50

Figure 11

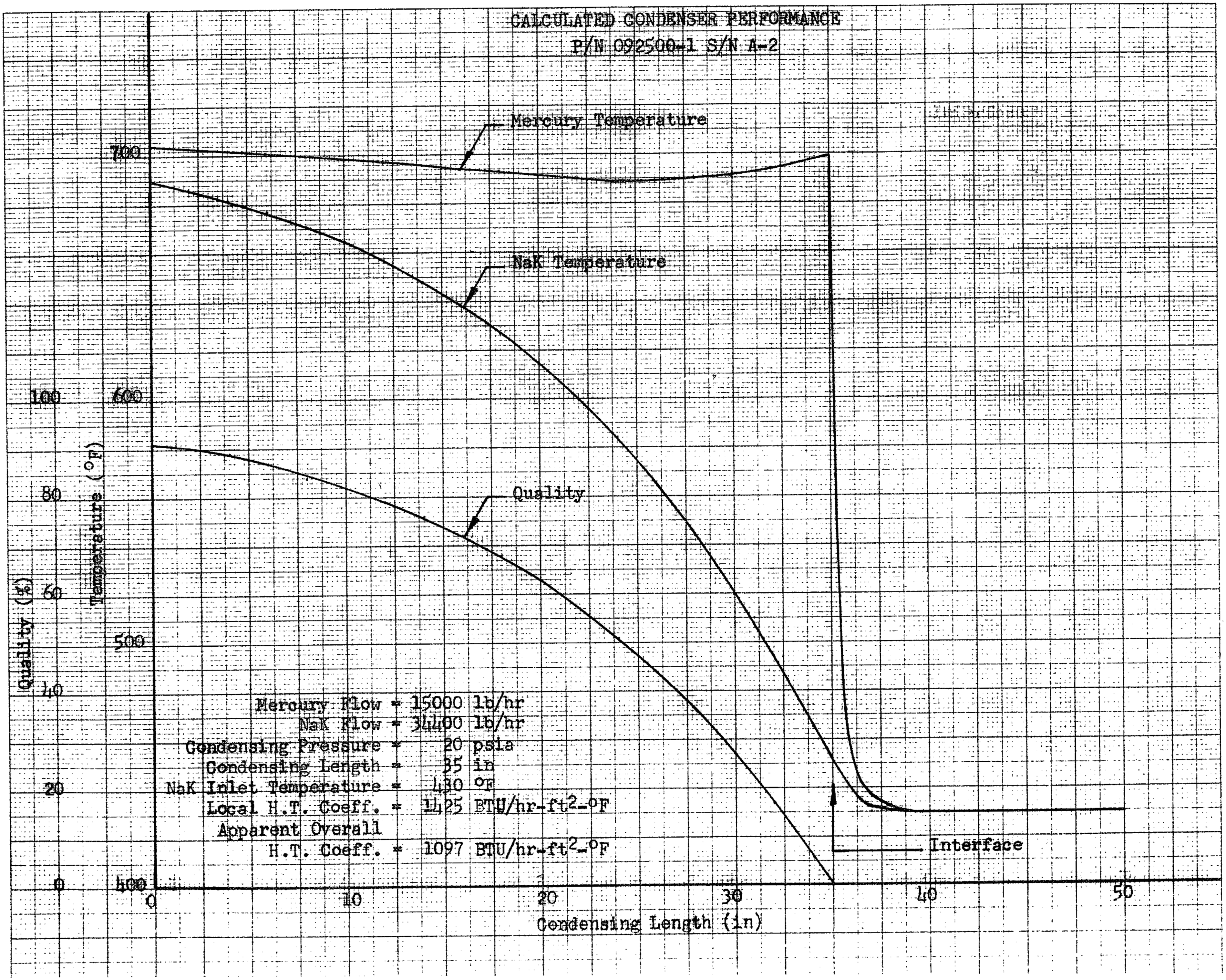
CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Temperature
NaK Temperature
Quality
Interface
Mercury Flow = 15000 lb/hr
NaK Flow = 34400 lb/hr
Condensing Pressure = 20 psia
Condensing Length = 35 in
NaK Inlet Temperature = 430 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1097 BTU/hr-ft2-°F
Quality (%)
Temperature (°F)
Condensing Length (in)
0
10
20
30
40
50
400
500
600
700
0
20
40
60
80
100

Figure 12

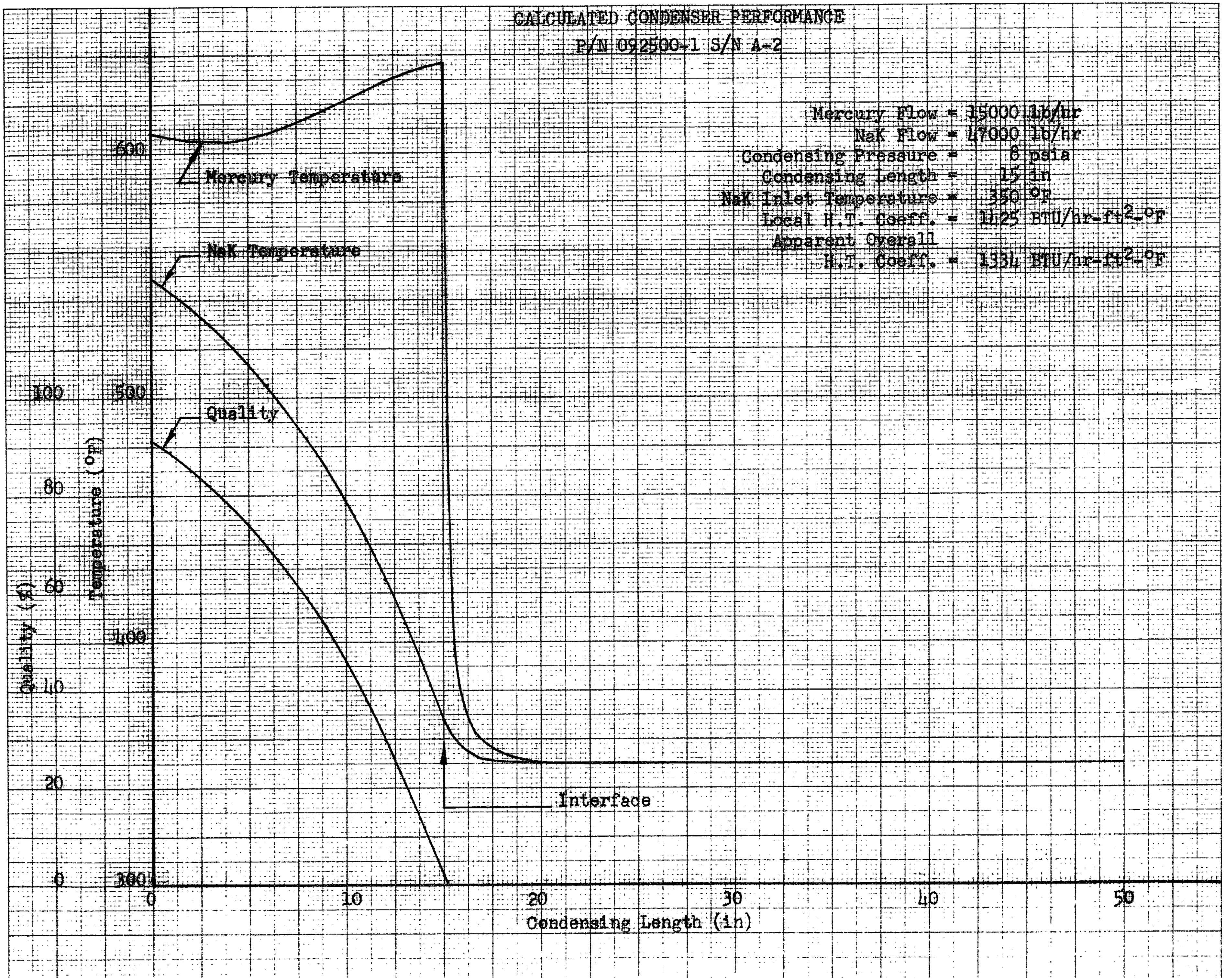
CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Flow = 15000 lb/hr
NaK Flow = 47000 lb/hr
Condensing Pressure = 8 psia
Condensing Length = 15 in
NaK Inlet Temperature = 350 °F
Local H.T. Coeff. = 1125 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1334 BTU/hr-ft2-°F
Mercury Temperature
NaK Temperature
Quality
Interface
Quality (%)
Temperature (°F)
100
80
60
40
20
0
600
500
400
300
0
10
20
30
40
50
Condensing Length (in)

Figure 13

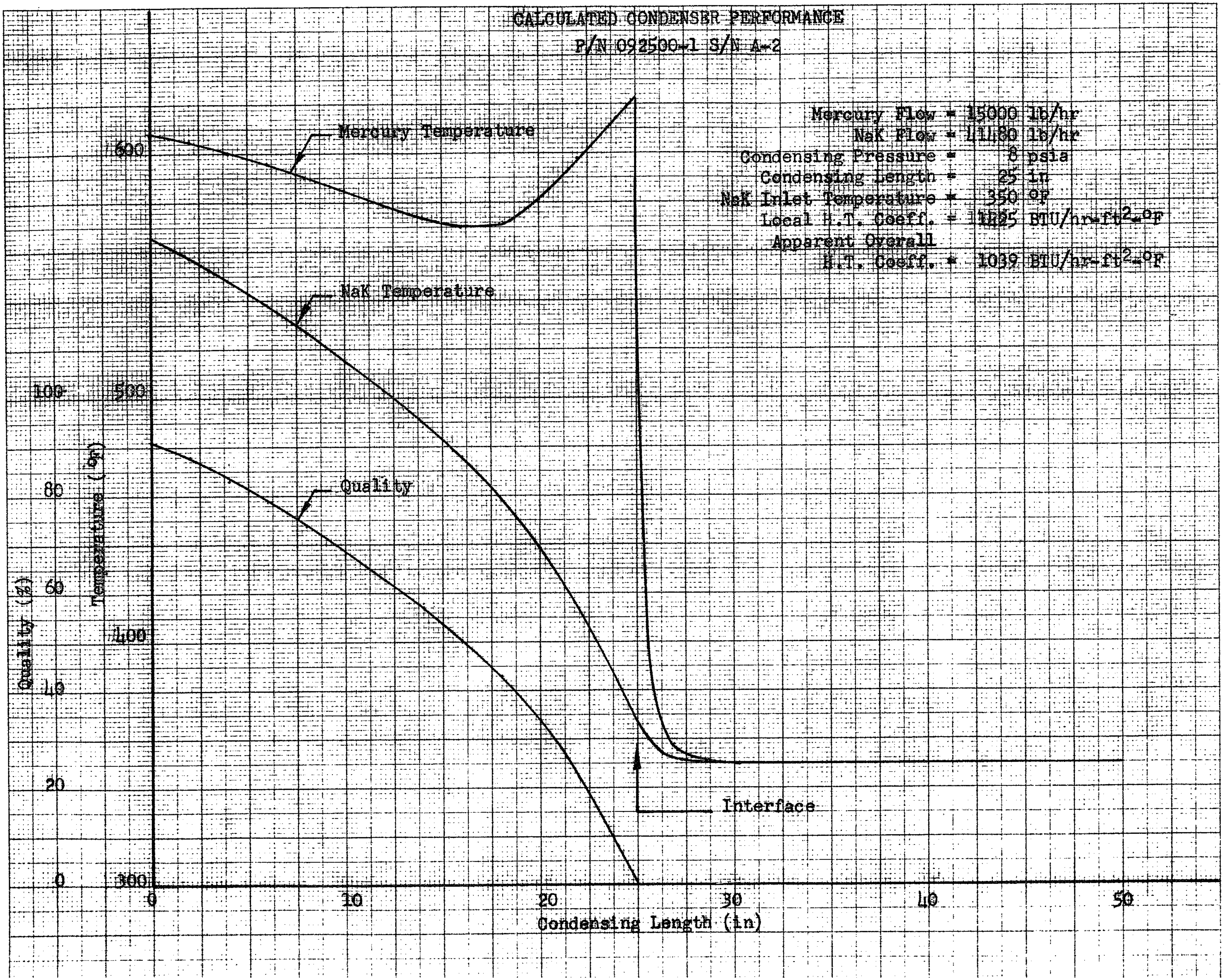
CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Flow = 15000 lb/hr
NaK Flow = 11480 lb/hr
Condensing Pressure = 8 psia
Condensing Length = 25 in
NaK Inlet Temperature = 350 °F
Local H.T. Coeff. = 1125 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 1039 BTU/hr-ft2-°F
Mercury Temperature
NaK Temperature
Quality
Interface
Quality (%)
Temperature (°F)
100
80
60
40
20
0
600
500
400
300
0
10
20
30
40
50
Condensing Length (in)

Figure 14

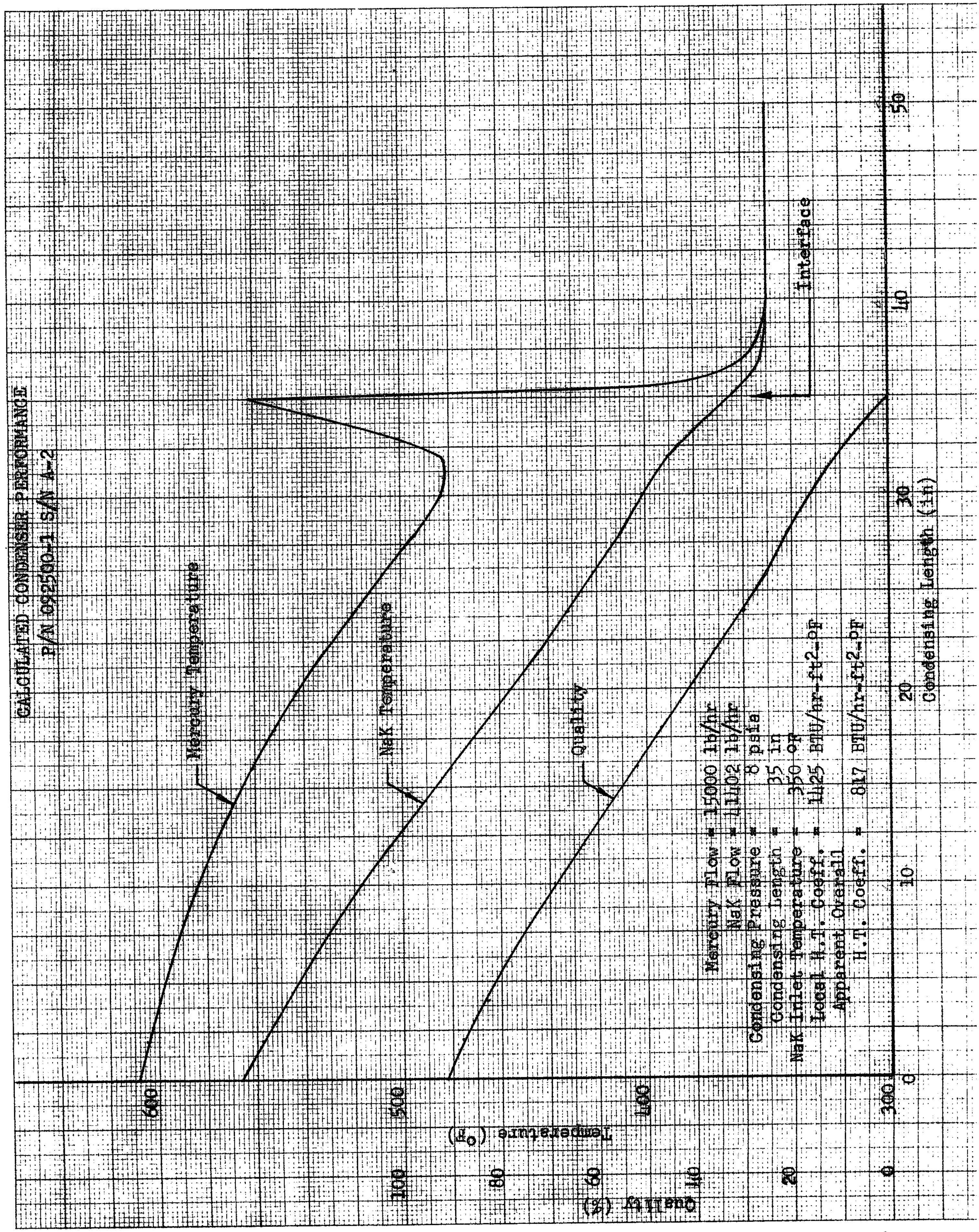
CALCULATED CONDENSER PERFORMANCE
P/N 092500-1 S/N A-2
Mercury Temperature
NaK Temperature
Quality
Interface
Mercury Flow = 15000 lb/hr
NaK Flow = 11402 lb/hr
Condensing Pressure = 8 psia
Condensing Length = 35 in
NaK Inlet Temperature = 350 °F
Local H.T. Coeff. = 1425 BTU/hr-ft2-°F
Apparent Overall
H.T. Coeff. = 817 BTU/hr-ft2-°F
Temperature (°F)
Quality (%)
Condensing Length (in)
600
500
400
300
100
80
60
40
20
0
0
10
20
30
40
50

Figure 15

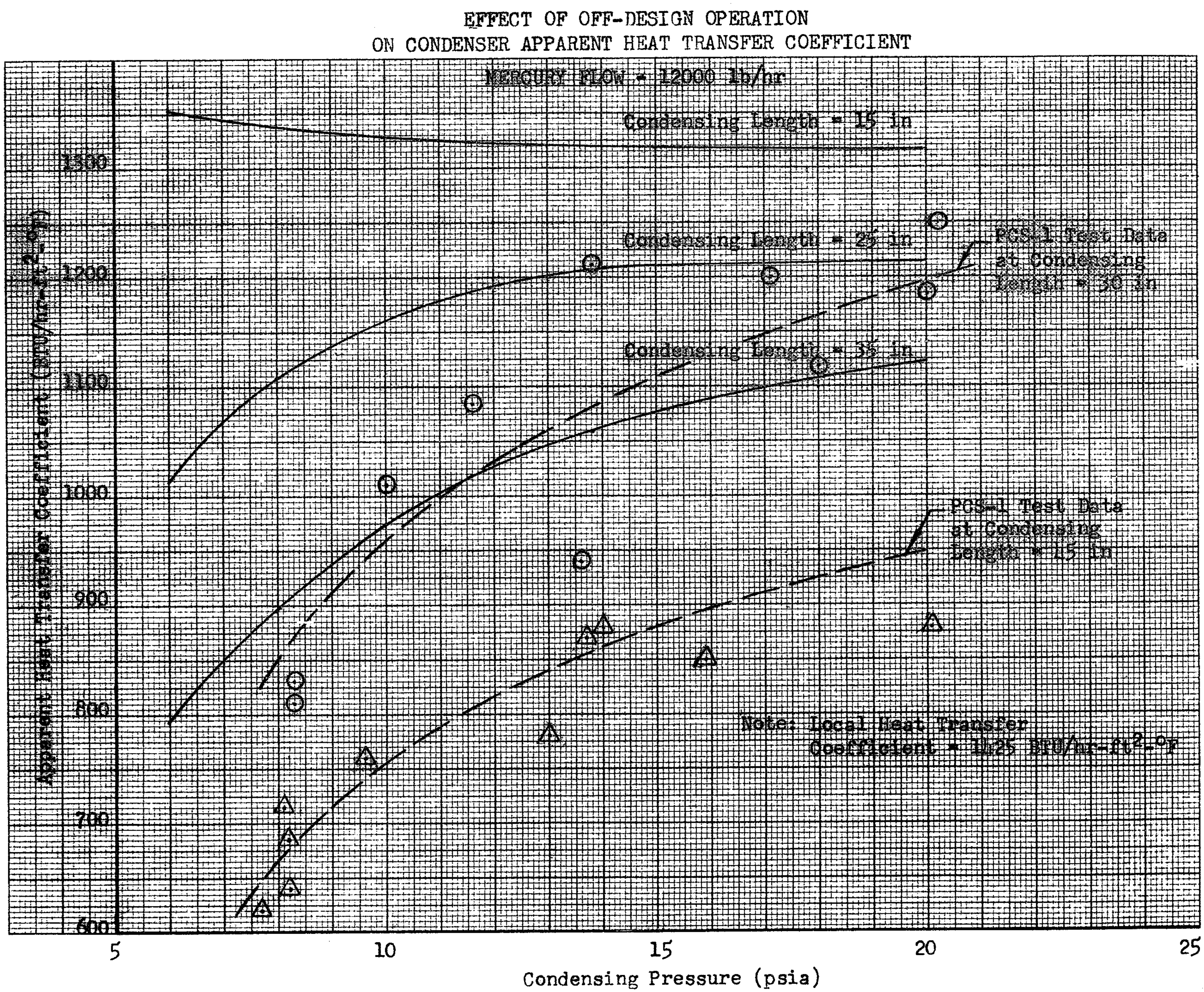
EFFECT OF OFF-DESIGN OPERATION
ON CONDENSER APPARENT HEAT TRANSFER COEFFICIENT
MERCURY FLOW = 12000 lb/hr
Condensing Length = 15 in
Condensing Length = 25 in
Condensing Length = 35 in
PCS-1 Test Data at Condensing Length = 30 in
PCS-1 Test Data at Condensing Length = 15 in
Note: Local Heat Transfer Coefficient = 1425 BTU/hr-ft2-°F
Apparent Heat Transfer Coefficient (BTU/hr-ft2-°F)
600
700
800
900
1000
1100
1200
1300
5
10
15
20
25
Condensing Pressure (psia)

Figure 16

EFFECT OF OFF-DESIGN OPERATION
ON CONDENSER APPARENT HEAT TRANSFER COEFFICIENT

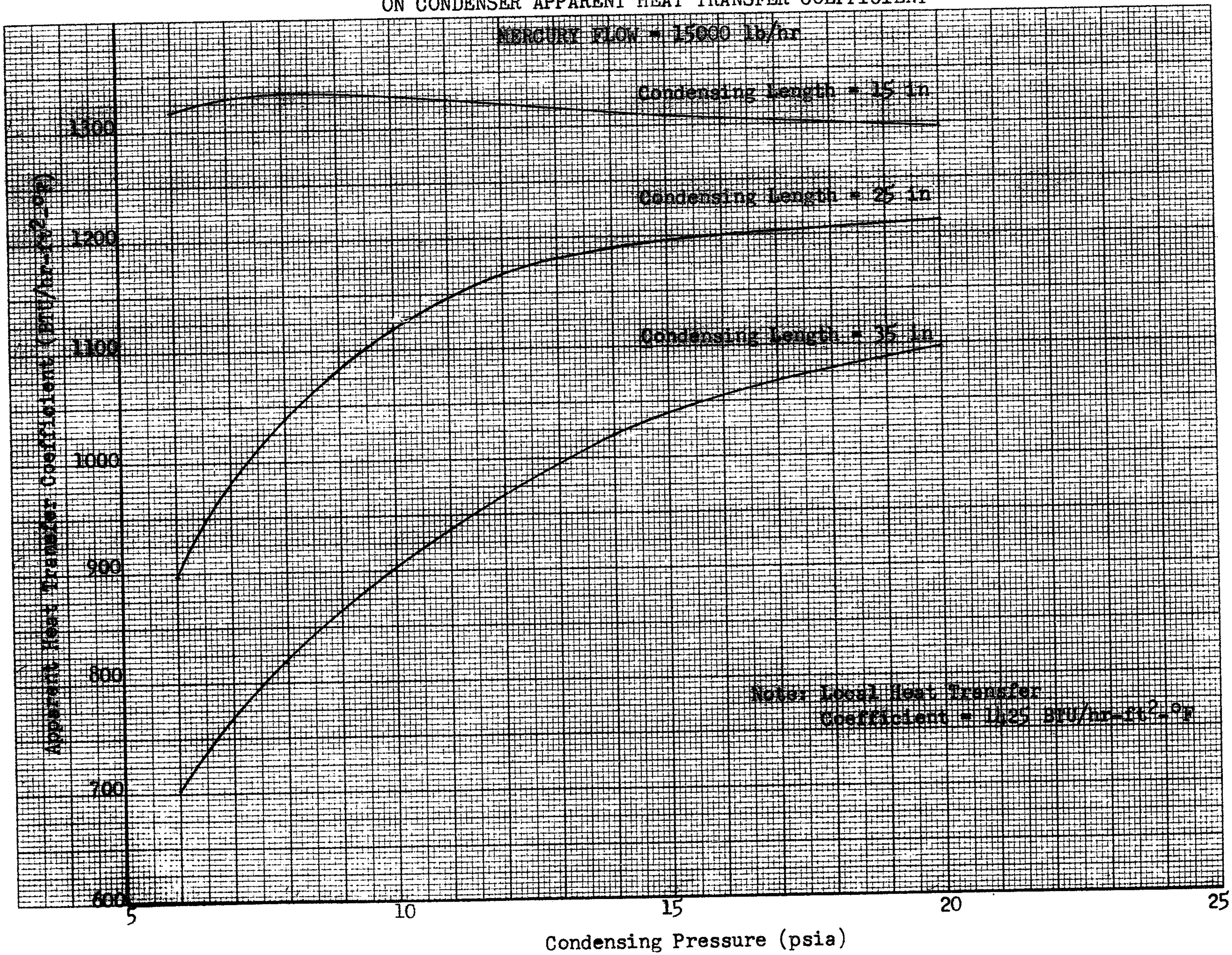

Figure 17

CONDENSER PERFORMANCE AT VERY LOW PRESSURES

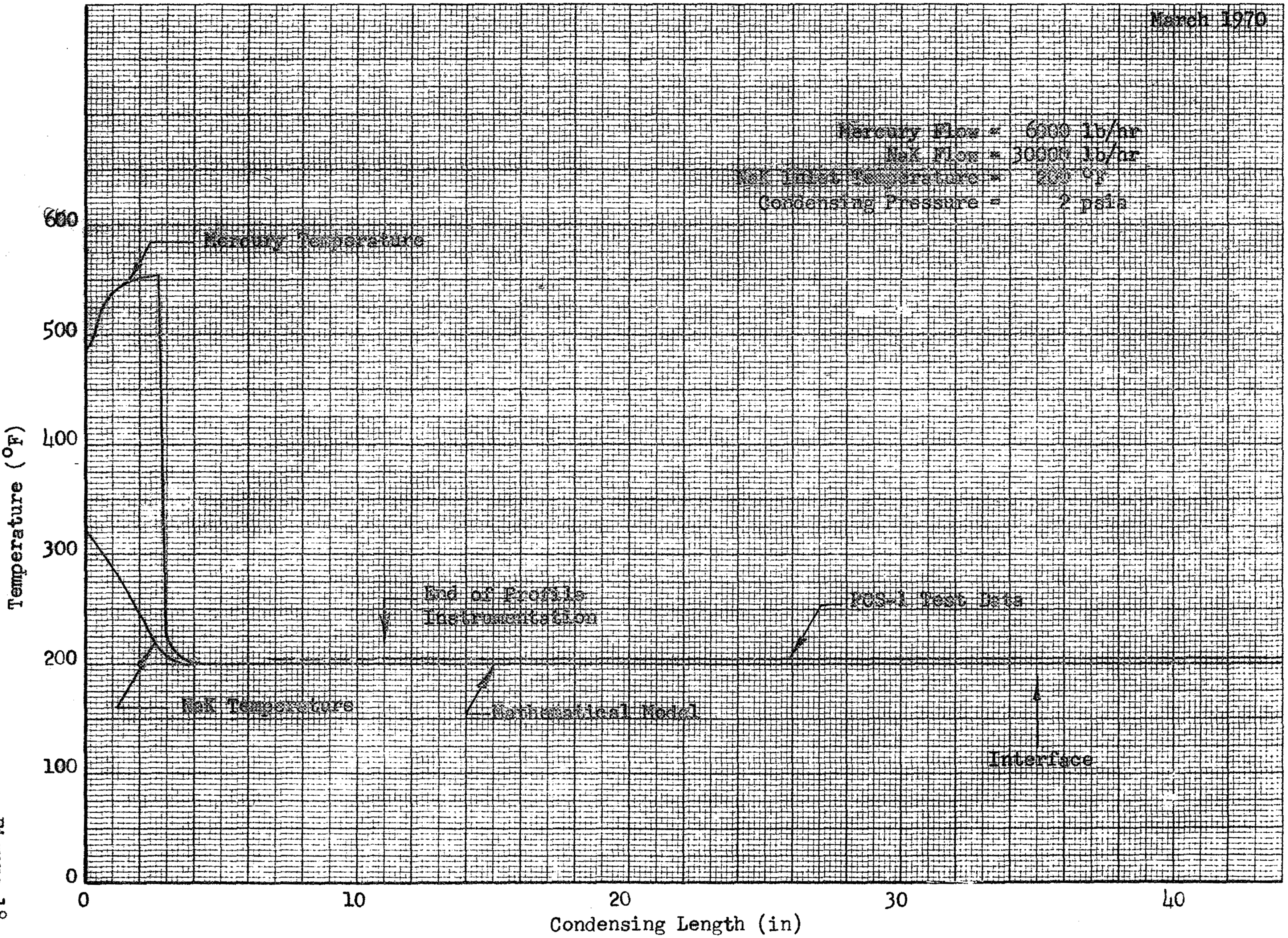

Figure 18

APPENDIX A

CONDENSER PERFORMANCE ANALYSIS

Appendix A - Condenser Performance Analysis

The following parameters are used in the analysis:

Name	Description	Units
A1	Tube cross-sectional area at entrance to incremental length	in^2
A2	Tube cross-sectional area at exit to incremental length	in^2
ACOND1	Heat transfer area down to entrance to incremental length	ft^2
ACOND2	Heat transfer area down to exit from incremental length	ft^2
AVAP	Cross-sectional area of tube available to vapor	in^2
CONDL	Condensing length	in
DELTA	Incremental length	in
DELTAA	Heat transfer area of incremental area	ft^2
DELTAQ	Heat transfer of incremental length	BTU/HR
DELTAX	Quality change in incremental length	
DIST	Distance along tube measured from mercury inlet	in
DLOCAL	Local diameter of single tube	in
DPDLG	Vapor phase pressure gradient	psi/in
DPDLTP	Two phase pressure gradient	psi/in
DPENT	Condenser entrance pressure drop	psi
DPM	Momentum pressure drop	psi
DPTM	Total momentum pressure drop	psi
DPTOT	Total pressure drop	psi
DPTP	Two phase pressure drop	psi
DPTTP	Total two phase pressure drop	psi
FFAC	Friction factor	
FHGTOT	Total mercury flow	lb/hr

Name	Description	Units
FNAK	NaK flow	lb/hr
HVAP	Heat of vaporization	BTU/lb
PHI	Lockhart-Martinelli parameter	
PINLET	Inlet vapor pressure	psia
PLOCAL	Local vapor pressure	psia
PM1A	Pressure equivalent of momentum X cross-sectional area at entrance to increment length	psi
PM2A	Pressure equivalent of momentum X cross-sectional area at exit of increment length	psi
QUAL	Vapor quality	
QUALIN	Inlet quality	
RENUM	Reynolds number	
RHOGAS	Vapor density	lb/in^3
RHOLIQ	Liquid density	lb/in^3
TINLET	Inlet vapor temperature	^oR
TLOCAL	Local vapor temperature	^oR
TNAK	Local NaK temperature	^oF
TNAKIN	NaK inlet temperature	^oF
TNAKO	NaK outlet temperature	^oF
ULOCAL	Local heat transfer coefficient	$BTU/hr-ft^2-{}^oF$
VEL1	Condenser inlet vapor velocity	in/sec
VEL2	Vapor velocity after expension into inlet manifold	in/sec
VISLIQ	Liquid viscosity	lb/ft-hr
VISVAP	Vapor viscosity	lb/ft-hr
VVAP	Vapor velocity	in/sec
XTT	Lockhart-Martinelli two-phase flow modulus for turbulent gas, turbulent liquid	

The following relationships were developed to define geometrical and fluid properties:

$$ACOND = 0.70\ (DIST) - 0.00507\ (DIST)^2 \qquad A\text{-}1$$

$$DLOCAL = 0.460 - 0.0065\ DIST \qquad A\text{-}2$$

$$HVAP = 128.9 - 0.00825\ (TLOCAL - 400) \qquad A\text{-}3$$

$$RHOGAS = 0.01085\ \left(\frac{PLOCAL}{TLOCAL}\right) \qquad A\text{-}4$$

$$RHOLIQ = 0.463 - 4.92 \times 10^{-5}\ (TLOCAL - 1060) \qquad A\text{-}5$$

$$VISVAP = 0.132 + 2.2 \times 10^{-4}\ (TLOCAL - 1060) \qquad A\text{-}6$$

$$VISLIQ = 2.09 - 1.2 \times 10^{-3}\ (TLOCAL - 1060) \qquad A\text{-}7$$

$$TLOCAL = \frac{13050}{14.19 - \ln PLOCAL} \qquad A\text{-}8$$

The cross-sectional area available to the vapor is given by:

$$AVAP = \frac{\frac{\pi}{4}\ (DLOCAL)^2}{1 + \frac{RHOGAS}{RHOLIQ}\ \left(\frac{1 - QUAL}{QUAL}\right)} \qquad A\text{-}9$$

The vapor velocity is given by:

$$VVAP = \frac{FHGTOT \times QUAL}{3600 \times 73 \times AVAP \times RHOGAS} \qquad A\text{-}10$$

The vapor quality is a function of the condenser heat transfer. The quality is determined at each increment by computing the heat transfer resulting from the temperature potential at that increment. The starting NaK temperature is the NaK outlet temperature which is given by:

$$TNAKO = \left(\frac{FHGTOT}{0.21 \times FNAK}\right) \left[\left(HVAP \times QUALIN\right) + 0.0325\left(TINLET - TNAKIN\right)\right] - TNAKIN \qquad A\text{-}11$$

The heat transferred in an increment is given by:

$$\text{DELTAQ} = \text{ULOCAL} \times \text{DELTAA} \left(\text{TLOCAL} - \text{TNAK}\right) \qquad \text{A-12}$$

$$\text{where} \quad \text{DELTAA} = \text{ACOND2} - \text{ACOND1} \qquad \text{A-13}$$

The change in NaK temperature from one increment to the next is:

$$\frac{\text{DELTAQ}}{0.21 \times \text{FNAK}} \qquad \text{A-14}$$

The change in quality from one increment to the next is:

$$\text{DELTAX} = \frac{\text{DELTAQ}}{\text{FHGTOT} \times \text{HVAP}} \qquad \text{A-15}$$

The mercury temperature at each increment is the saturation temperature at the local pressure. The pressure is found by using the Lockhart-Martinelli two-phase flow theory. The theory is used by first determining the pressure drop of the vapor as if it flowed alone in the tube. The pressure gradient for the vapor is given by:

$$\text{DPDLG} = \frac{2 \times \text{FFAC}(\text{VVAP})^2\text{RHOGAS}}{386 \times \text{DLOCAL}} \qquad \text{A-16}$$

$$\text{where FFAC} = \frac{0.046}{(\text{RENUM})^{0.2}} \qquad \text{A-17}$$

$$\text{RENUM} = \frac{3600 \times 12 \times \text{RHOGAS} \times \text{VVAP} \times \text{DLOCAL}}{\text{VISVAP}} \qquad \text{A-18}$$

The two phase pressure gradient is related to the vapor phase pressure gradient by:

$$\text{DPDLTP} = (\text{PHI})^2\ \text{DPDLG} \qquad \text{A-19}$$

The Lockhart-Martinelli parameter, PHI, is a function of the flow regime, quality, and fluid properties. The parameter PHI can be expressed as:

$$\text{PHI} = 1.30 \times 5.82\ (\text{XTT}) - 3.20\ (\text{XTT})^2 + 0.283\ (\text{XTT})^3 \qquad \text{A-20}$$

where XTT is the flow modulus for turbulent gas and turbulent liquid.

The flow modulus is defined as:

$$XTT = \left(\frac{1 - QUAL}{QUAL}\right)^{0.9} \left(\frac{RHOGAS}{RHOLIQ}\right)^{0.5} \left(\frac{VISLIQ}{VISVAP}\right)^{0.1} \quad A-21$$

The pressure change due to momentum change is a function of the assumption made regarding the liquid velocity. The best overall condenser pressure drop correlation has been found to occur when the liquid velocity is assumed to be zero. For zero liquid velocity, the pressure change due to momentum is:

$$DPM = \frac{FHGTOT \times \Delta(VVAP \times QUAL)}{386 \times 73 \times 3600 \left(\frac{A1 + A2}{2}\right)} \quad A-22$$

An additional pressure loss in the condenser occurs at the entrance. The entering flow expands into the inlet manifold area and then enters the individual tubes. The pressure loss is:

$$DPENT = \frac{RHOGAS}{2 \times 386} \left[(VEL1 - VEL2)^2 + 0.5\ (VEL2)^2\right] \quad A-23$$

$$\text{where } VEL1 = 2.09 \times 10^{-5}\ \frac{FHGTOT}{RHOGAS} \quad A-24$$

$$VEL2 = 7.23 \times 10^{-6}\ \frac{FHGTOT}{RHOGAS} \quad A-25$$

COMPUTER PROGRAM

The foregoing analysis has been programmed for computer solution. The program is given in Figure A-1. The required input to the program is:

1.	Total mercury flow	FHGTOT
2.	Condensing length	CONDL
3.	Inlet pressure	PINLET
4.	Increment length	DELTA
5.	NaK flow	FNAK
6.	NaK inlet temperature	TNAKIN
7.	Local heat transfer coefficient	ULOCAL

```
35 DELTAX=DELTAQ/(FHGTOT*HVAP)
36 K=K+1
37 IF(K-1)38,38,39
38 QUAL=QUALIN
39 QUAL=QUAL-DELTAX
40 RHOLIQ=0.463-4.92E-5*(TLOCAL-600.0)
41 RHOGAS=0.01085*PLOCAL/(TLOCAL+460.0)
42 AVAP=0.785*(DLOCAL)**2.0/(1.0+(RHOGAS*((1.0-QUAL)/QUAL)/RHOLIQ))
43 VVAP=FHGTOT*QUAL/(3600.0*73.0*AVAP*RHOGAS)
44 VISVAP=0.132+0.00022*(TLOCAL-600.0)
45 VISLIQ=2.09-0.0012*(TLOCAL-600.0)
46 RENUM=RHOGAS*VVAP*DLOCAL*3600.0*12.0/VISVAP
47 FFAC=0.046/(RENUM**0.2)
48 DPDLG=FFAC*(VVAP)**2.0*RHOGAS/(193.0*DLOCAL)
49 XTT=(((1.0-QUAL)/QUAL)**0.9)*((RHOGAS/RHOLIQ)**0.5)*((VISLIQ/VISVAP)**0.1)
50 PHI=1.30+(5.82*XTT)-(3.20*XTT**2.0)+(0.283*XTT**3.0)
51 DPDLTP=PHI**2.0*DPDLG
52 DPTP=DPDLTP*DELTA
53 PM2A=FHGTOT*VVAP*QUAL/(386.0*3600.0*73.0)
54 A2=0.785*DLOCAL**2.0
55 IF(DIST-DELTA)57,56,56
56 DPM=(PM2A-PM1A)/((A1+A2)/2.0)
57 PM1A=PM2A
58 A1=A2
60 DPTTP=DPTTP+DPTP
61 DPTM=DPTM+DPM
62 DPTOT=DPTTP+DPTM+DPENT
63 PRINT 64,DIST,ULOCAL,PLOCAL,TLOCAL,TNAK,DPTOT,QUAL
64 FORMAT(2F8.0,F8.2,2F8.0,2F8.2)
65 PLOCAL=PLOCAL-DPTP-DPM
66 IF(DIST-CONDL)19,67,67
67 STOP
LISTING COMPLETED
```

Figure A-1

CONDENSER PERFORMANCE ANALYSIS
COMBINED HEAT TRANSFER AND PRESSURE DROP

```
 1 FORMAT(4X,'DIST',2X,'ULOCAL',3X,'PLOCAL',2X,'TLOCAL',3X,'TNAK',4X,'DPTOT',4X,'QUAL')
 2 PRINT  1
 3 FORMAT(4X,'(IN)',1X,'(B/HF2F)',2X,'(PSIA)',4X,'(F)',5X,'(F)',3X,'(PSIA)',2X,'(PCENT)')
 4 PRINT  3
 5 QUALIN=0.92
 6 K=-1
 7 N=-1
 8 J=-1
 9 DPM=0
10 DPTTP=0
11 DPTM=0
12 TINLET=(13050.0/(14.19-ALOG(PINLET)))-460.0
13 RHOGAS=0.01085*PINLET/(TINLET+460.0)
14 VEL1=2.09E-5*FHGTOT/RHOGAS
15 VEL2=7.23E-6*FHGTOT/RHOGAS
16 DPENT=RHOGAS*((VEL1-VEL2)**2.0+0.5*VEL2**2.0)/(2.0*386.0)
17 PLOCAL=PINLET-DPENT
18 DIST=-DELTA
19 DIST=DIST+DELTA+0.00001
20 TLOCAL=(13050.0/(14.19-ALOG(PLOCAL)))-460.0
21 DLOCAL=0.460-0.0065*DIST
22 HVAP=128.9-0.00825*(TLOCAL-400.0)
23 TNAKO=((FHGTOT/(FNAK*0.21))*((HVAP*QUALIN)+(0.0325*(TINLET-TNAKIN))))+TNAKIN
24 N=N+1
25 IF(N-1)26,26,27
26 TNAK=TNAKO
27 J=J+1
28 IF(J-1)29,29,30
29 ACOND1=0
30 ACOND2=0.70*DIST-0.00507*DIST**2.0
31 DELTAA=ACOND2-ACOND1
32 ACOND1=ACOND2
33 DELTAQ=ULOCAL*DELTAA*(TLOCAL-TNAK)
34 TNAK=TNAK-(DELTAQ/(FNAK*0.21))

(CONTINUED)
```

Figure A-1 (cont.)