TECHNICAL REPORT

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TEST AND OBSERVATION RESULTS OF BIMETALLIC AND INVERTED
BUCKET TYPE TRAPS ON DRIP AND TRACER LINE APPLICATIONS

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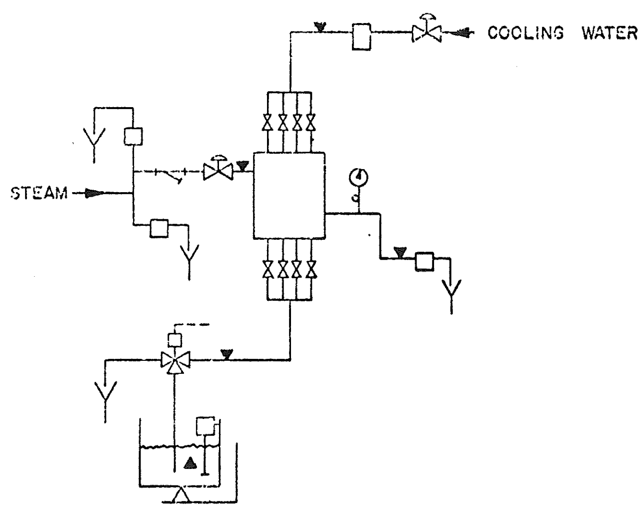
Most tracer lines are presently equipped with bimetallic, thermostatic, thermodynamic, or inverted bucket traps to control condensate removal. In our Technical Report I, we compared the inverted bucket trap to the thermodynamic type trap, and conclusively determined the superiority of the inverted bucket trap. In this report, we will again use the inverted bucket trap as a source of comparison; this time to the bimetallic trap. Our initial study utilized a series of tests set up specifically for the thermodynamic and inverted bucket traps. However, we soon realized that, due to the amount of sub-cooling inherent to the bimetallic principle, these same tests could not be reapplied. The bimetallic trap, being a temperature activated unit, has to rely on radiation and/or convection losses through the trap body. Many manufacturers of this type of trap recommend that a considerable length of pipe be left free of insulation to aid in the under-cooling or quick radiation of the unit. This is done so the trap does not have to rely totally on the radiation of the body housing. However, the heart of the unit is in the housing itself and, therefore, can only react to the temperature present in the unit. The inverted bucket trap, due to its separation principle, does not need the cooling leg and, therefore, can work with the fully insulated pipe. Also, in colder climates, the trap itself can be insulated without having an adverse effect on its operation. Based on this information, a test was run to establish the effect sub-cooling would have on a piece of heat exchanging equipment using either a bimetallic or inverted bucket type trap. We were more concerned with a comparison between the two units and the principles being tested than with the overall efficiency of the heat exchanger itself. So we tested the inverted bucket trap and then the bimetallic trap to establish a comparison chart between the two operating principles.



TEST
SETUP

Test Parameters

Steam Pressure: 11.5 bar

Cooling Water Flow Test #1: 5.50 l/min

Flow Test #2: 9.25 l/min

Flow Test #3: 13.00 l/min

Cooling Water Temperature Entering Heat Exchanger: 11.1°C

Heat Transfer Area Test #1: 288.68 cm^2

Test #2: 433.02 cm^2

Test #3: 577.36 cm^2

First, the heat transfer rate is established for the inverted bucket trap draining the shell of the heat exchanger. A bimetallic trap is then put in its place and tested at the same parameters to determine the heat transfer rate. The inverted bucket trap is then retested so that the heat transfer rates at the beginning and the end of each test can be compared. This assures that there has not been a significant amount of fouling of the heat exchanger tubes during the test. Next, a new set of parameters is established and this test is repeated. The test data is then converted to meaningful results by performing a number of calculations. The total heat transfer rate can be calculated by using the following formula.

$$q = (E_F - E_I - E_N)(60/t)$$

Where:

q = heat transfer rate in kcal/hr

E_F = heat content of tank at the end of test in kcal

E_I = heat content of tank at the start of test in kcal

E_N = heat content of water entering heat exchanger in kcal

In order to calculate the total enthalpy or heat of the tank water system, the water equivalent weight of the tank must first be calculated. This is necessary because it obviously requires fewer kilocalories to raise the temperature of the metal tank 1°C than to raise the temperature of the water 1°C.

The formula to determine the water equivalent weight is as follows.

$$W_e = W_C \cdot \frac{cpc}{cpw} \qquad [kg]$$

Where:

W_e = water equivalent weight of the tank [kg]

W_C = weight of the tank [kg]

cpc = specific heat of container [kcal/kg/°C]

cpw = specific heat of water [kcal/kg/°C]

W_e = .117 W_C (container is stainless steel; water is in -10 to 60°C range; container temperature $\approx$ H_2O temperature)

And:

$$W_I = W_S - W_C + W_e$$

$$= W_S - W_C + .117\ W_C$$

$$= W_S - .883\ W_C \quad [kg]$$

$$W_F = W_E - W_C + W_e$$

$$= W_E - .883\ W_C \quad [kg]$$

Where:

W_I = initial weight of water plus water equivalent weight of the container [kg]

W_F = final weight of water plus water equivalent weight of the container [kg]

W_S = initial weight of water plus container [kg]

W_E = final weight of water plus container [kg]

Therefore:

$$q = [\{(W_E - .883\ W_C)\ (i_{FSP})\} - \{(W_S - .883\ W_C)\ (i_{ISP})\} - \{(W_E - W_S)\ (i_{INSP})\}] \cdot (60/t)$$

Where:

i_{FSP} = specific enthalpy of water at final temperature [kcal/kg]

i_{ISP} = specific enthalpy of water at initial temperature [kcal/kg]

i_{INSP} = Specific enthalpy of water at entering temperature in heat exchanger [kcal/kg]

Or:

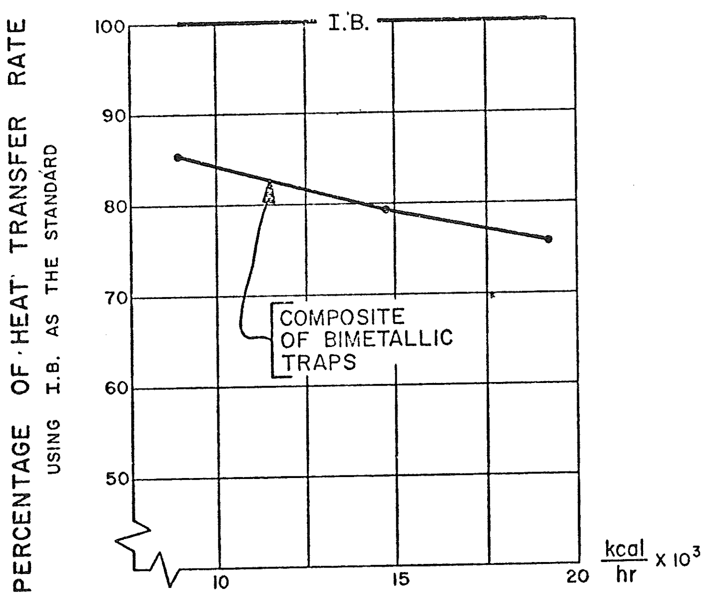
$$q = [(W_F \cdot i_{FSP}) - (W_I \cdot i_{ISP}) - (\Delta W \cdot i_{INSP})]\ (60/t)$$

Or:

$$q = (E_F - E_I - E_N)\ (60/t)$$

The information was then consolidated and plotted on the following graph. This graph represents a percentage average of several manufacturers of bimetallic traps compared to several inverted bucket type traps of one manufacturer.

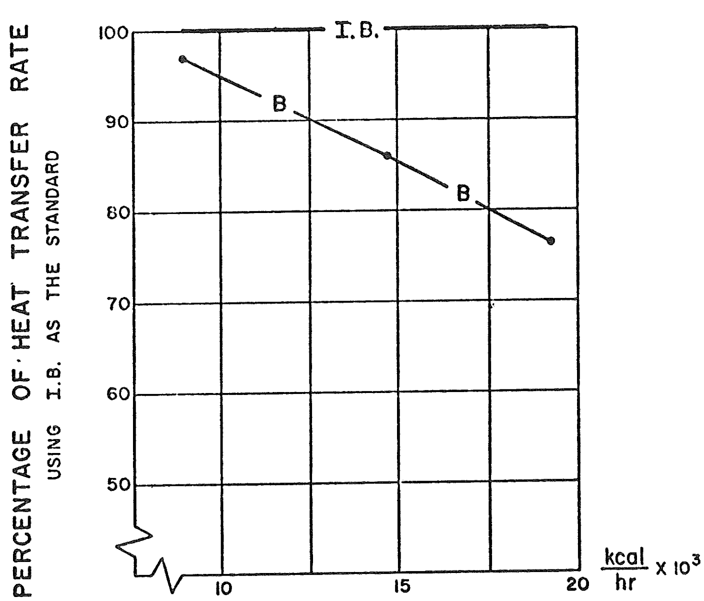
HEAT EXCHANGER PERFORMANCE
I.B.-VS- BIMETALLIC TRAP
PRELIMINARY RESULTS



RATE OF HEAT TRANSFER WITH I.B.

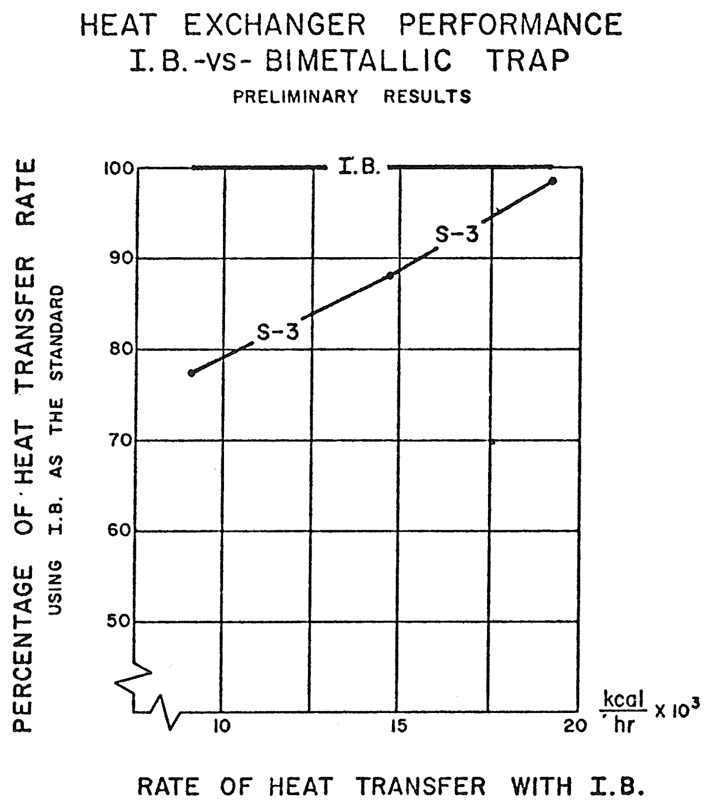
Although the bimetallic traps were new and had been ordered for tracer line applications; the test results, in most cases, showed under-cooling of the condensate from 10 to 40°C. The steam pressure of 11.5 bar was left unchanged. The X axis shows the rate of heat transfer in the heat exchanger for various conditions when drained with an inverted bucket trap. The Y axis shows the average percentage of this heat transfer rate attained by different types of bimetallic traps. The graph shows an average of all tested traps, however the performances varied from quite near to far from that of the inverted bucket trap. Graphs C and D on Pages 9 and 10 show bimetallic traps which were close to the heat transfer rate originally obtained by the inverted bucket traps. The traps were then retested at the same conditions on a steam loss stand as described in Technical Report I. It was found that the trap in Graph C was consuming 4 kg/hr of steam and the trap in Graph D, 5 kg/hr of steam. Those traps whose performance was farthest from the inverted bucket type trap reached a substantial sub-cooling of 40 to 50°C. By testing the bimetallic traps, it became apparent that their subsequent performances were quite erratic. The response in opening and closing was unpredictable and, although the test conditions were closely observed and held constant during the test, all units tested gave different results on different occasions. After retesting, the data was averaged. The erratic behavior was found to occur when the trap was retested several months after the original testing. By opening these traps, it was found that sediment, which is contained in the steam and condensate, had covered the bimetallic elements forming a layer of insulation and causing increased friction. This covering of sediment had a significant impact on the characteristics of the opening and closing curve. Since the efficiency comparison showed that the reactions of the bimetallic elements were inconsistent, the efficiency test setup was used to observe the operating characteristics under normal conditions but at different pressures and condensate loads. New traps of different manufacturers were chosen and monitored under the same conditions as the inverted bucket trap.

HEAT EXCHANGER PERFORMANCE
I.B.-VS- BIMETALLIC TRAP
PRELIMINARY RESULTS



RATE OF HEAT TRANSFER WITH I.B.

GRAPH C



GRAPH D

Test Parameters

Steam Pressure: 6.5 bar and 11.5 bar

Cooling Water Flow Test #1: 5.70 l/min

Flow Test #2: 9.50 l/min

Flow Test #3: 13.25 l/min

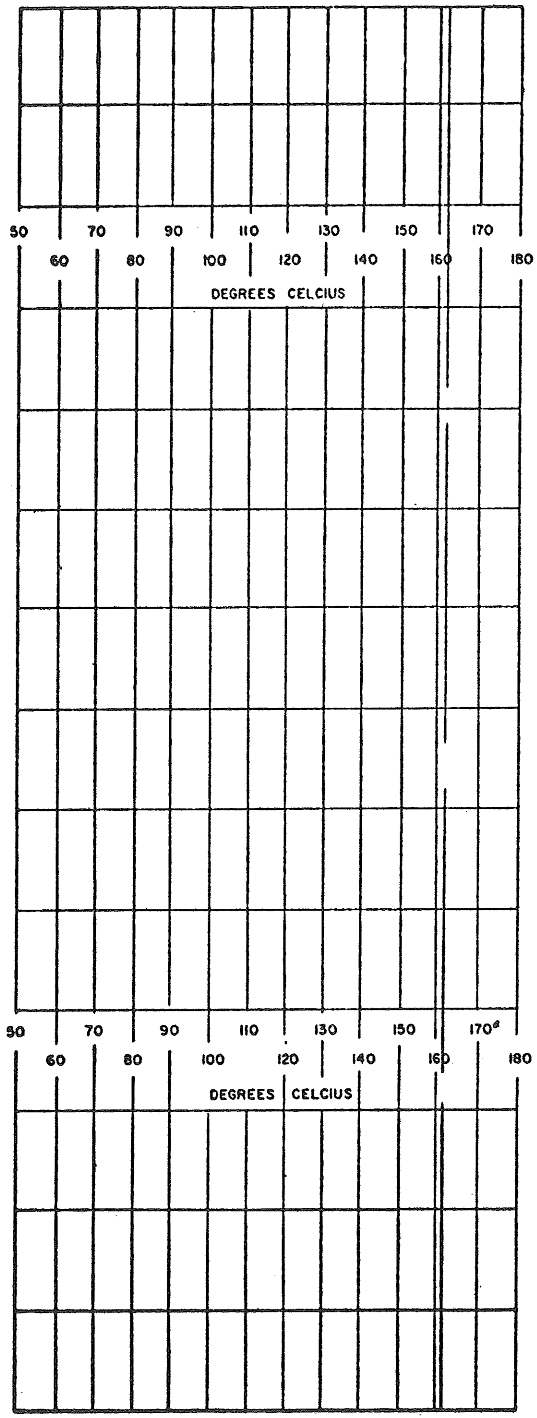
Cooling Water Temperature Entering Heat Exchanger: 11.1°C

First, the inverted bucket trap was tested at both 6.5 and

11.5 bar. In both cases, the inverted bucket trap pulled a straight line on the recorder indicating that the trap was discharging condensate at steam temperature. (See Graphs E and F.)

Bimetallic trap 2A showed a steam loss of only .85 kg/hr at the low pressure, but went up to 2.5 kg/hr at the high pressure. (See Graphs G-1, G-2, H-1 and H-2.) The unit itself reacted quite differently under low and high pressures. The potential condensate load of 11.5 kg/hr showed different curves for the 6.5 and 11.5 bar systems. Sub-cooling of only 13°C was found at the low pressure, while at the high pressure sub-cooling reached 37°C. In both cases, the unit took a semi-fixed position and did not cycle on and off until it reached a potential condensate load of 30 kg/hr at the low pressure and 39.5 kg/hr at the high pressure. Note that sub-cooling was noticeable at the low pressure from 32 to 40°C and at the high pressure from 40 to 70.5°C.

SHT. 1 OF 1



TRAP NO. IB2

STEAM TEMPERATURE ≈ 162°C

STEAM PRESSURE ≈ 6.5 BAR

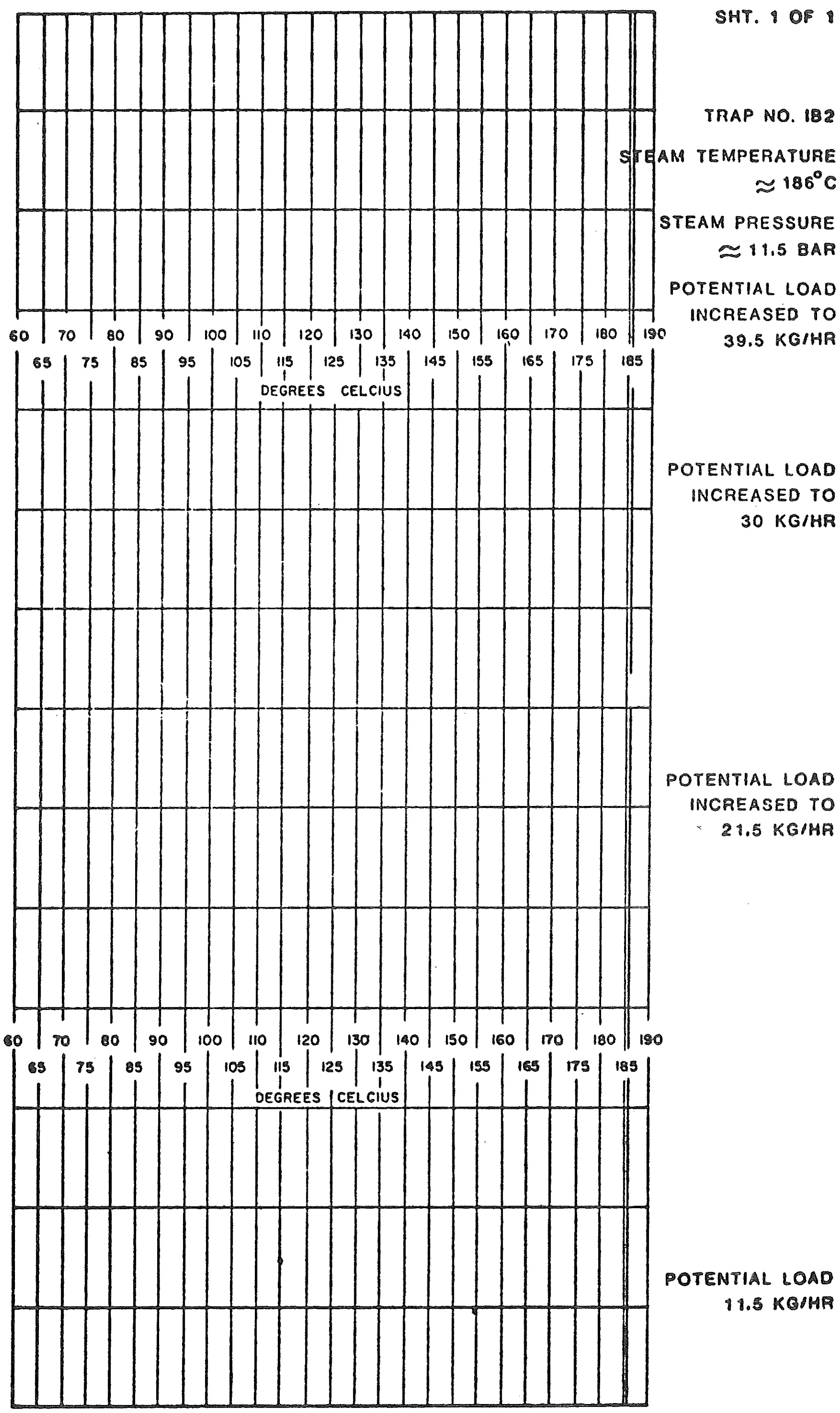
POTENTIAL LOAD INCREASED TO 39.5 KG/HR

POTENTIAL LOAD INCREASED TO 30 KG/HR

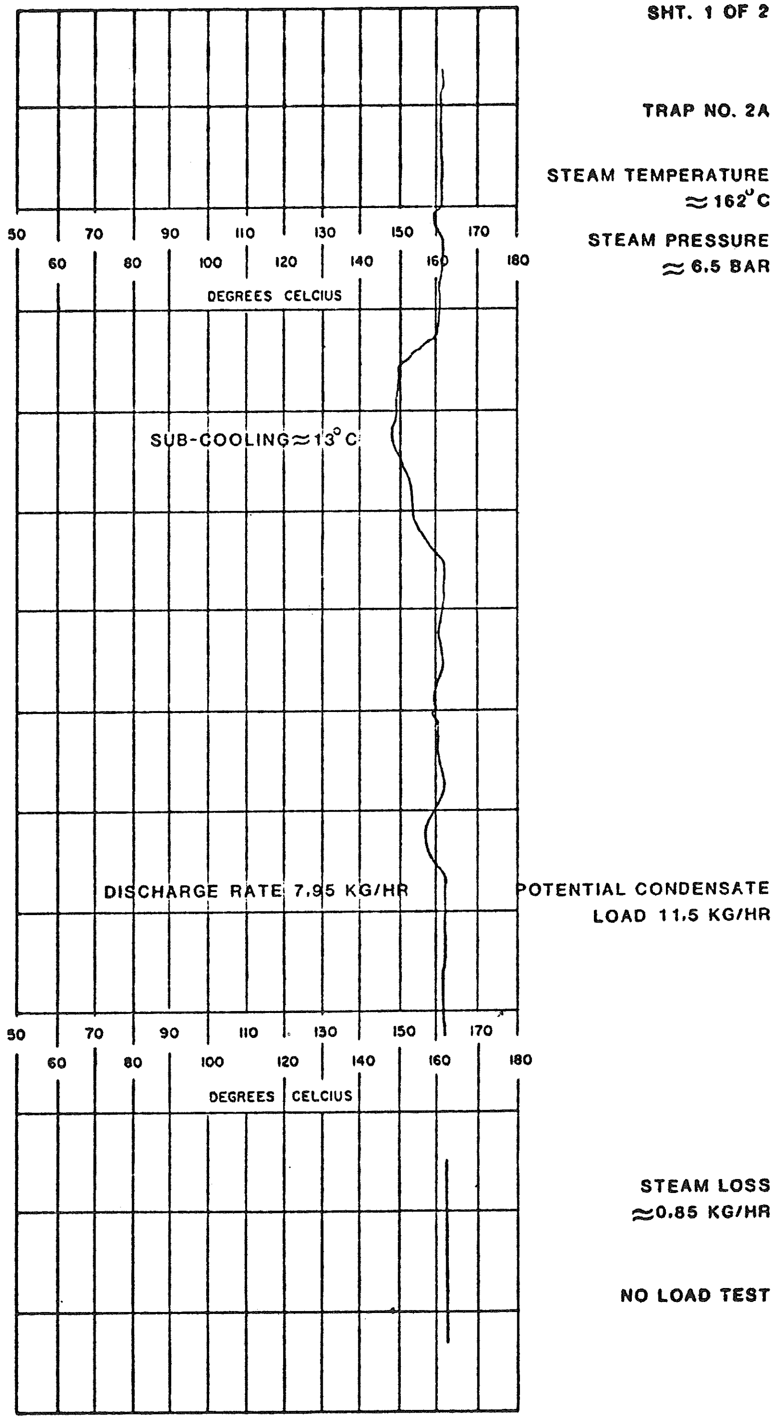
POTENTIAL LOAD INCREASED TO 21.5 KG/HR

POTENTIAL LOAD 11.5 KG/HR

GRAPH E


SHT. 1 OF 1
TRAP NO. 1B2
STEAM TEMPERATURE ≈ 186°C
STEAM PRESSURE ≈ 11.5 BAR
POTENTIAL LOAD INCREASED TO 39.5 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL LOAD INCREASED TO 30 KG/HR
POTENTIAL LOAD INCREASED TO 21.5 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL LOAD 11.5 KG/HR

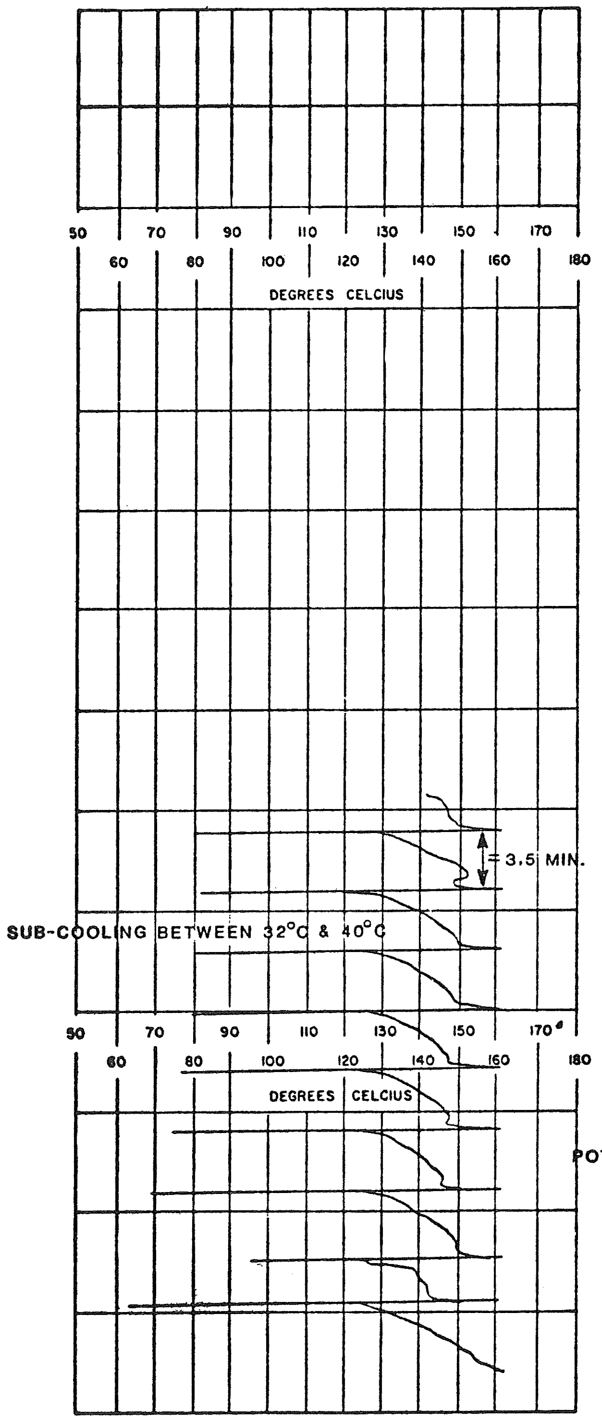
GRAPH F


SHT. 1 OF 2
TRAP NO. 2A
STEAM TEMPERATURE ≈ 162°C
STEAM PRESSURE ≈ 6.5 BAR
50 60 70 80 90 100 110 120 130 140 150 160 170 180
DEGREES CELCIUS
SUB-COOLING ≈ 13° C
DISCHARGE RATE 7.95 KG/HR
POTENTIAL CONDENSATE LOAD 11.5 KG/HR
STEAM LOSS ≈0.85 KG/HR
NO LOAD TEST

GRAPH G-1

SHT. 2 OF 2

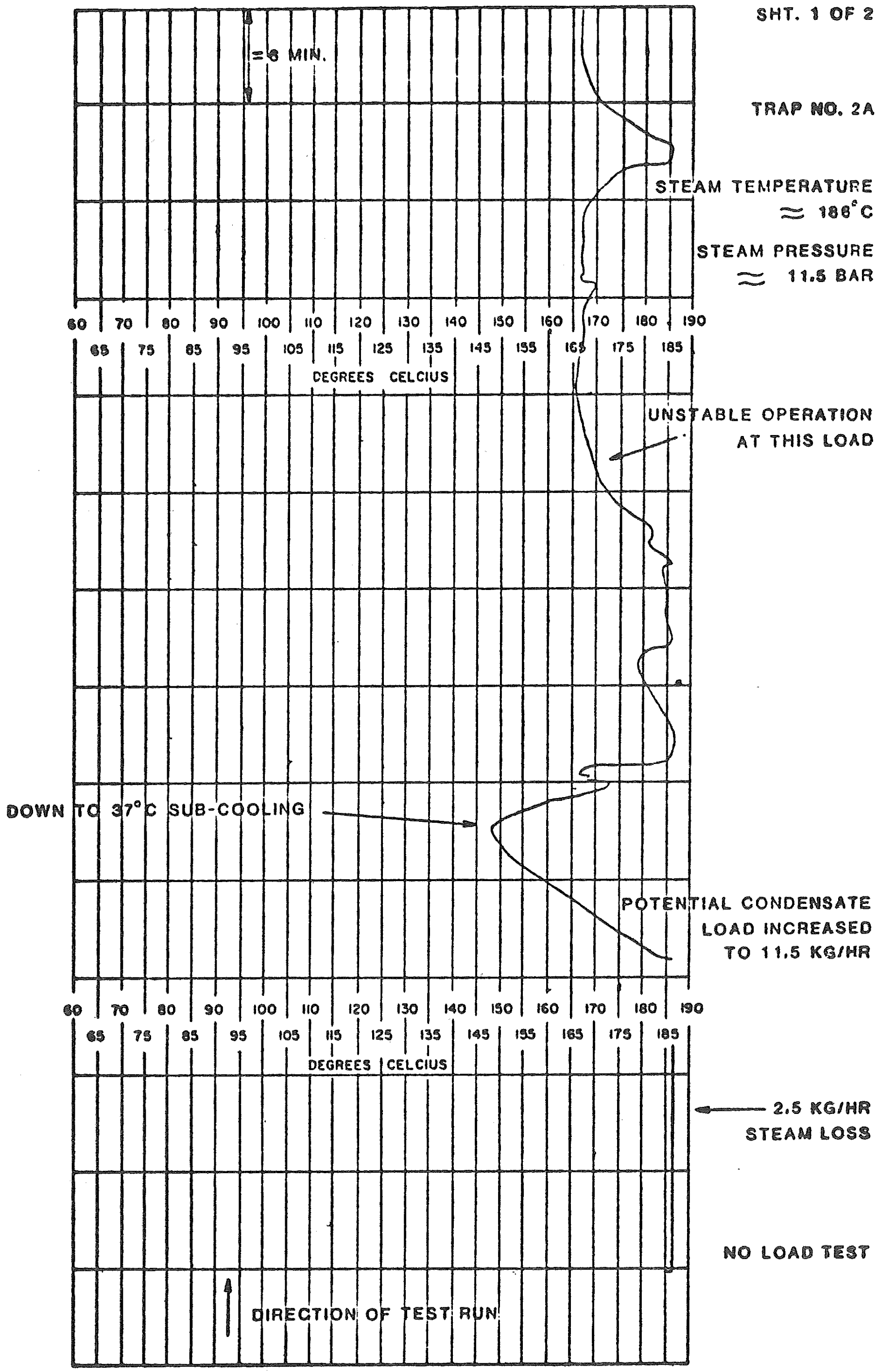
TRAP NO. 2A



ACTUAL DISCHARGE
52.5 KG/HR

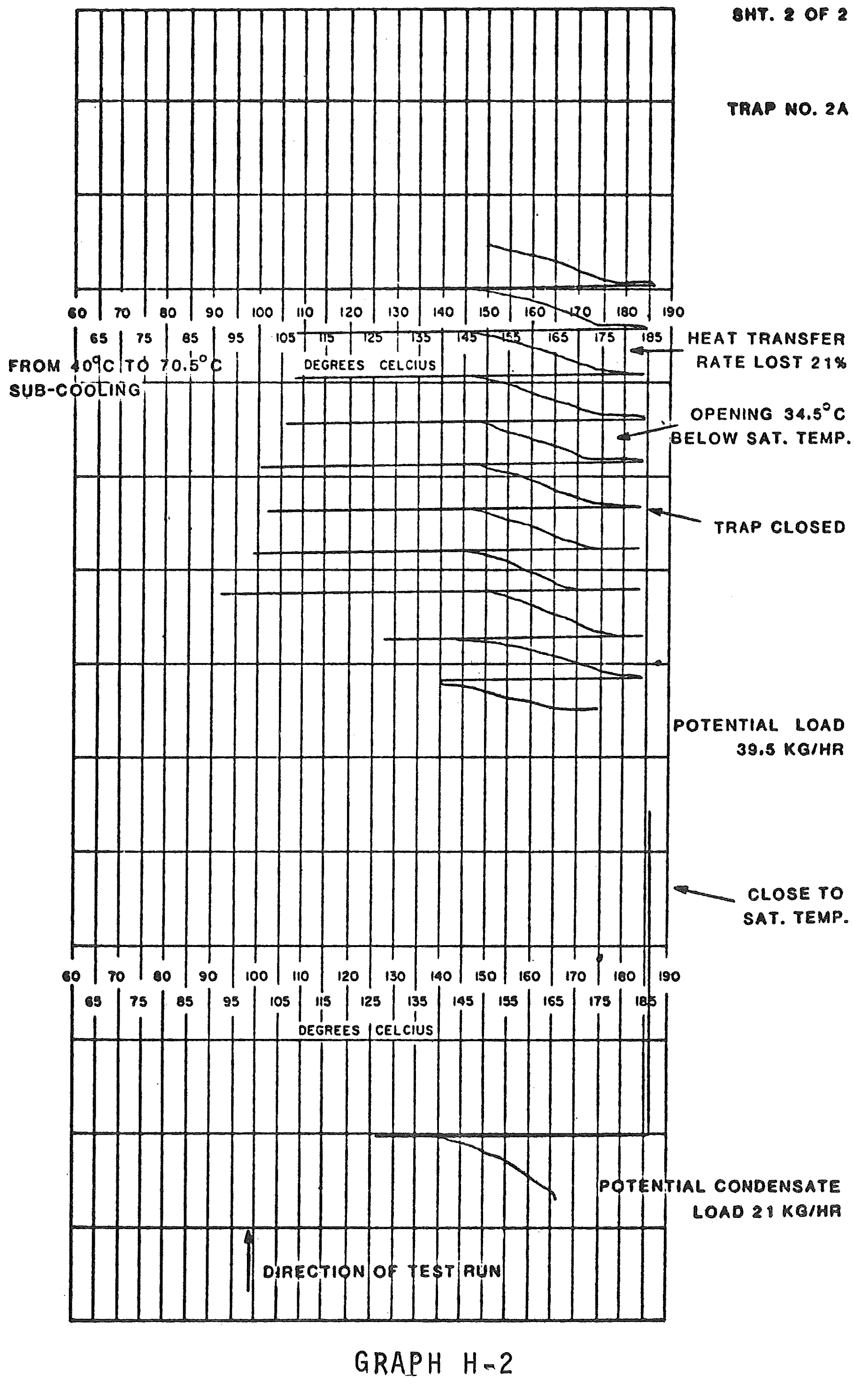
POTENTIAL CONDENSATE
LOAD 30 KG/HR

GRAPH G-2



SHT. 1 OF 2
= 6 MIN.
TRAP NO. 2A
STEAM TEMPERATURE ≈ 186°C
STEAM PRESSURE ≈ 11.5 BAR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
UNSTABLE OPERATION AT THIS LOAD
DOWN TO 37°C SUB-COOLING
POTENTIAL CONDENSATE LOAD INCREASED TO 11.5 KG/HR
2.5 KG/HR STEAM LOSS
NO LOAD TEST
DIRECTION OF TEST RUN

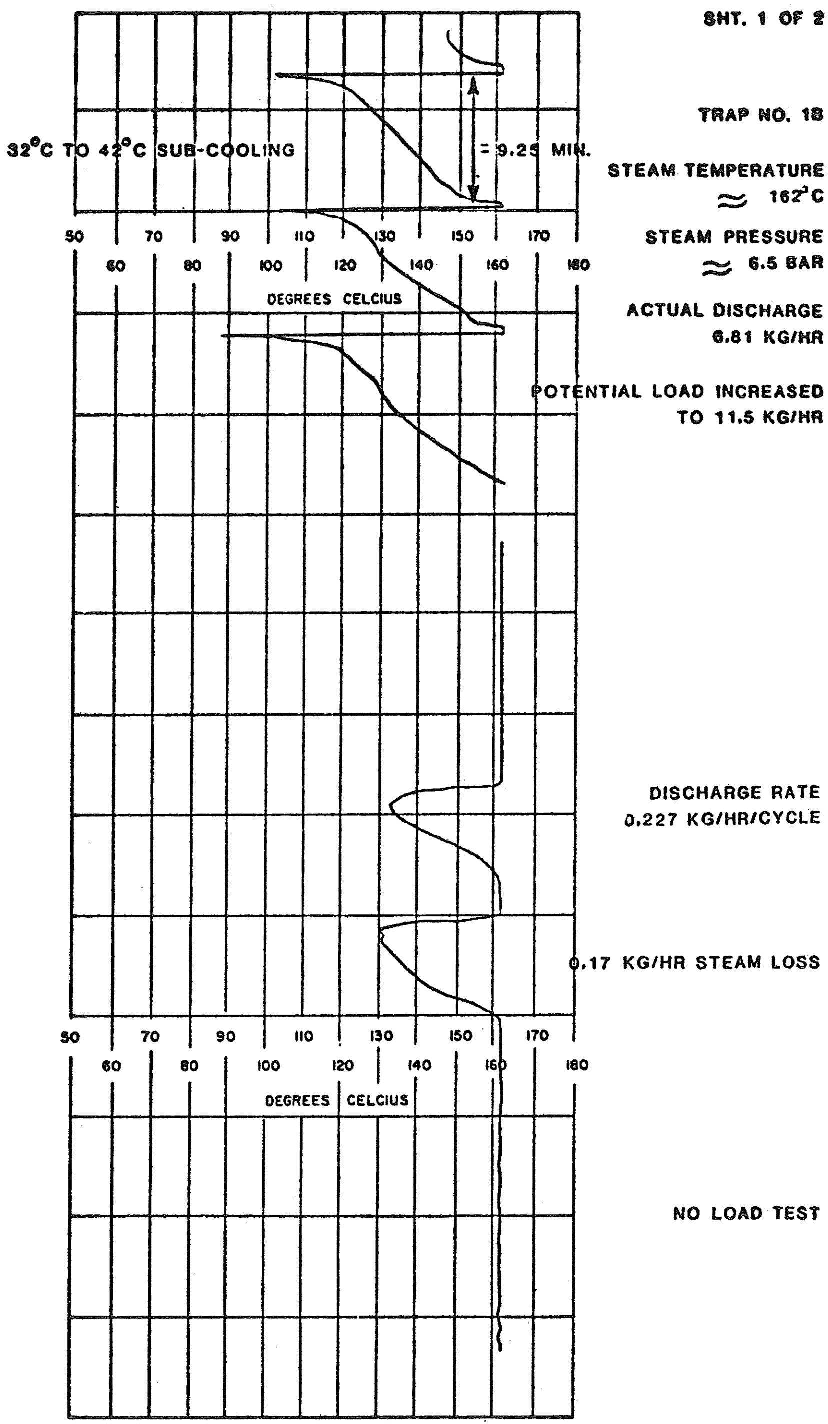
GRAPH H-1


SHT. 2 OF 2
TRAP NO. 2A
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
FROM 40°C TO 70.5°C
SUB-COOLING
HEAT TRANSFER
RATE LOST 21%
OPENING 34.5°C
BELOW SAT. TEMP.
TRAP CLOSED
POTENTIAL LOAD
39.5 KG/HR
CLOSE TO
SAT. TEMP.
POTENTIAL CONDENSATE
LOAD 21 KG/HR
DIRECTION OF TEST RUN

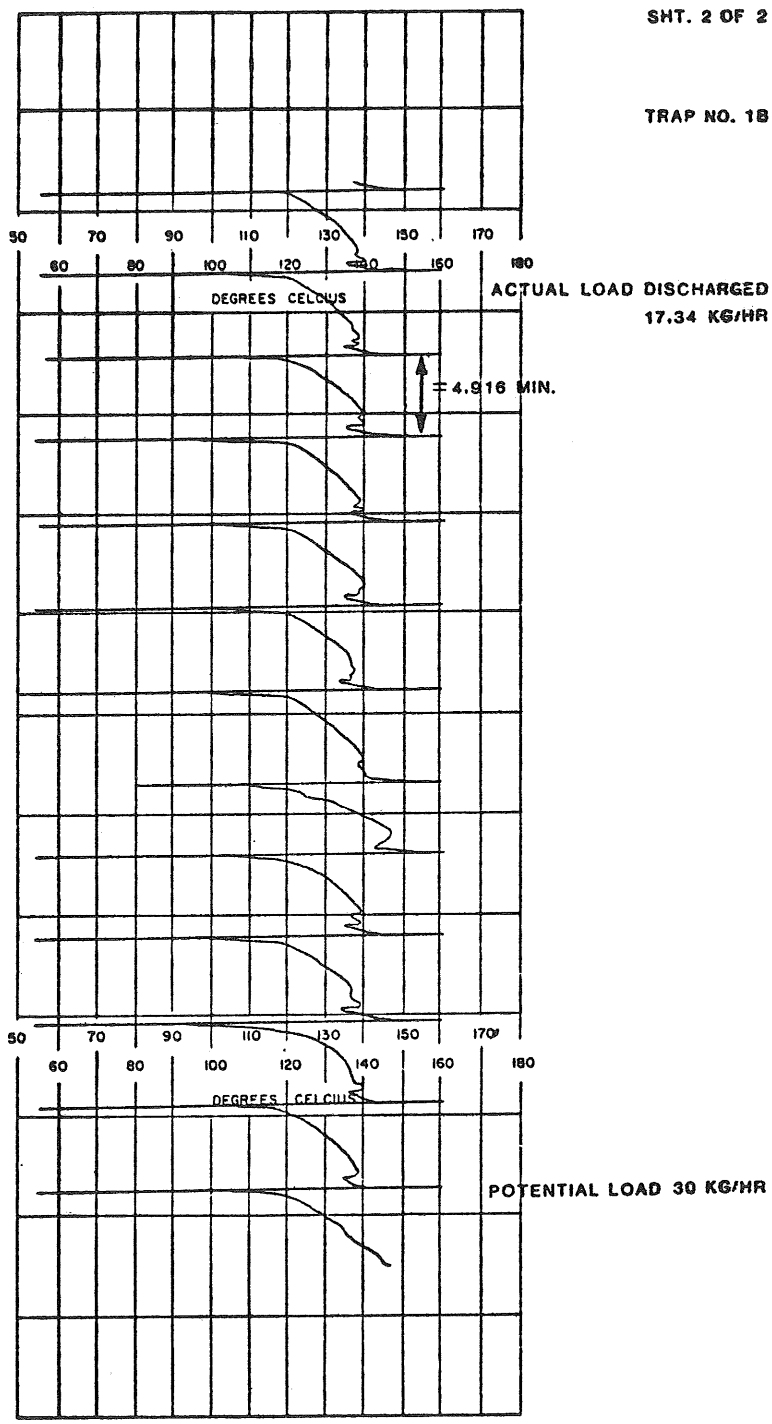
GRAPH H-2

The bimetallic trap 1B was tested at these two pressures and began with a no load test. (See Graphs I-1, I-2, J-1, J-2, J-3, and J-4.) Note that, by increasing the pressure, the steam leak or steam loss increased substantially. Also note that, at the low pressure, the cycle rate changed drastically from 9.25 to 4.916 minutes. At the higher pressure, the unit behaved quite erratically and, in many cases, would not come up to steam temperature. This was especially true with the condensate load of 11.5 kg/hr and halfway through the 21 kg/hr condensate load. Although the unit started cycling in a more typical way with a 39.5 kg/hr load, the opening below saturated temperature was approximately 39°C. The test recorded a heat transfer rate loss of 41%.

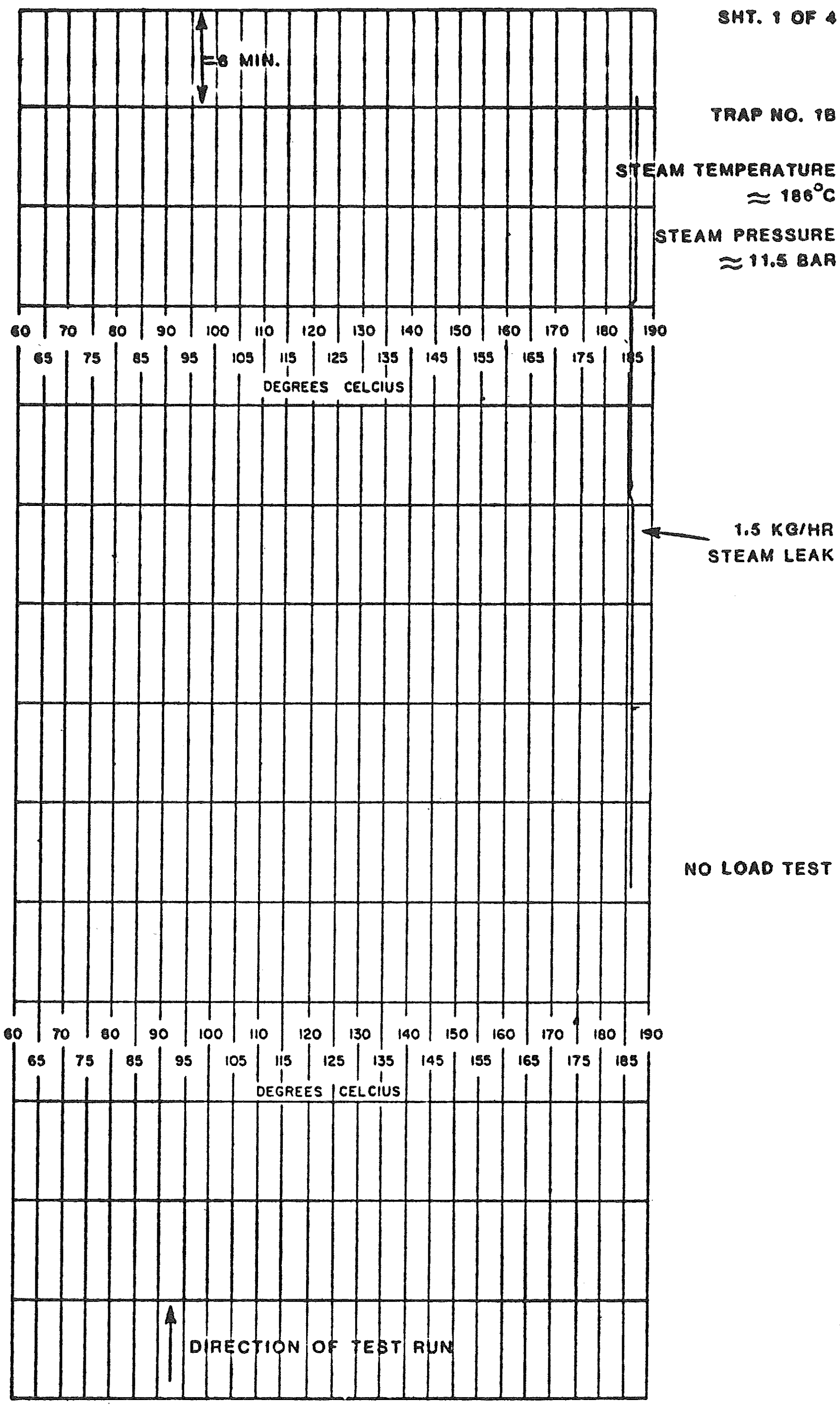
Bimetallic trap 2B, of the same manufacturer, model number and pressure as bimetallic trap 1B, lost 1.9 kg/hr on the no load test; but, at a higher pressure, lost only .36 kg/hr. (See Graphs K-1, K-2, L-1, L-2, L-3, and L-4.) This is exactly opposite to the reaction of Trap 1B. Although Trap 2B performed in the low pressure test approximately the same as Trap 1B, in the high pressure test Trap 2B took a fixed position and started cycling later in the 39.5 kg/hr condensate load.


SHT. 1 OF 2
TRAP NO. 18
32°C TO 42°C SUB-COOLING
= 9.25 MIN.
STEAM TEMPERATURE
≈ 162°C
STEAM PRESSURE
≈ 6.5 BAR
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
ACTUAL DISCHARGE
6.81 KG/HR
POTENTIAL LOAD INCREASED
TO 11.5 KG/HR
DISCHARGE RATE
0.227 KG/HR/CYCLE
0.17 KG/HR STEAM LOSS
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
NO LOAD TEST

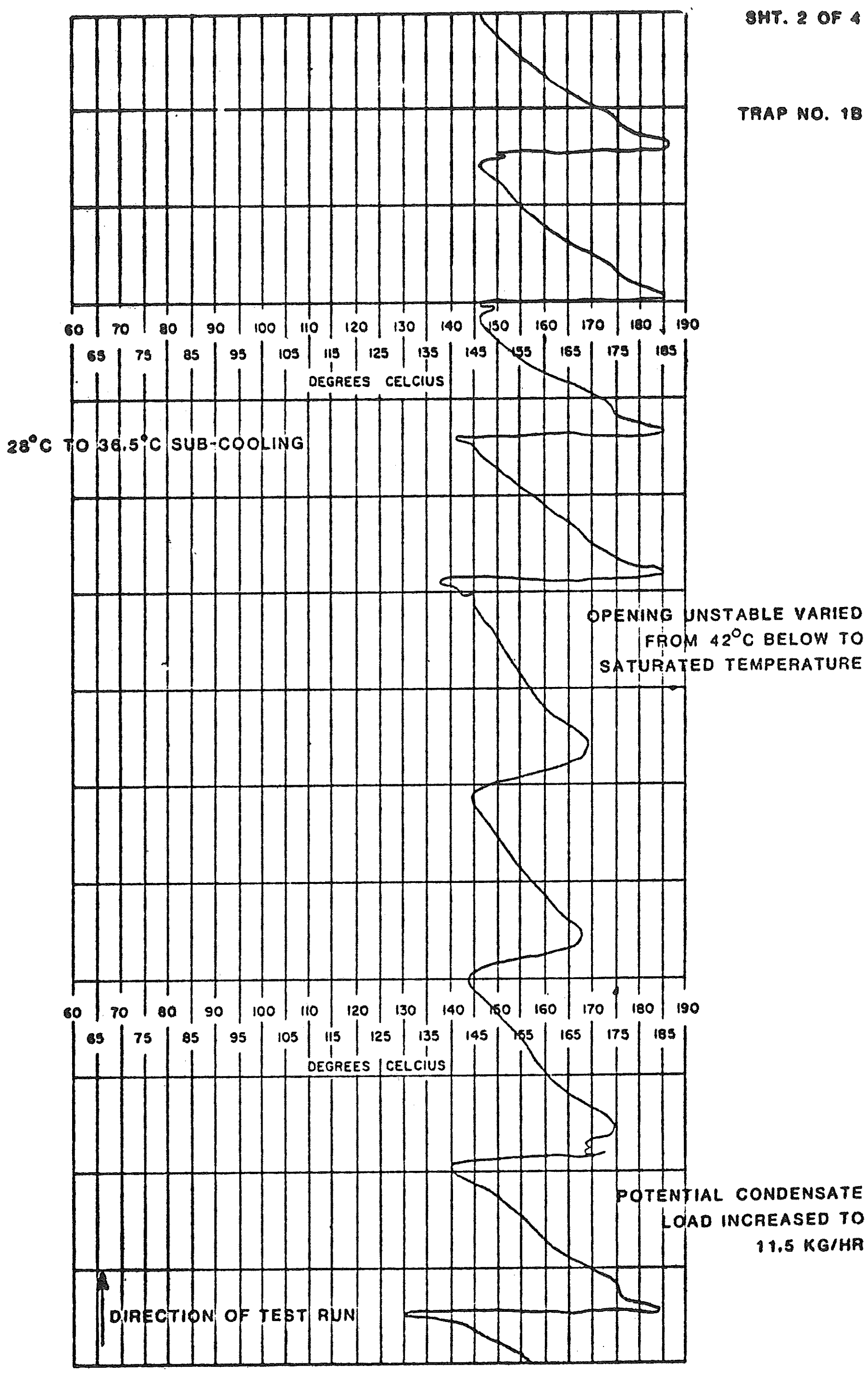
GRAPH I-1


SHT. 2 OF 2
TRAP NO. 1B
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
ACTUAL LOAD DISCHARGED
17.34 KG/HR
= 4.916 MIN.
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
POTENTIAL LOAD 30 KG/HR

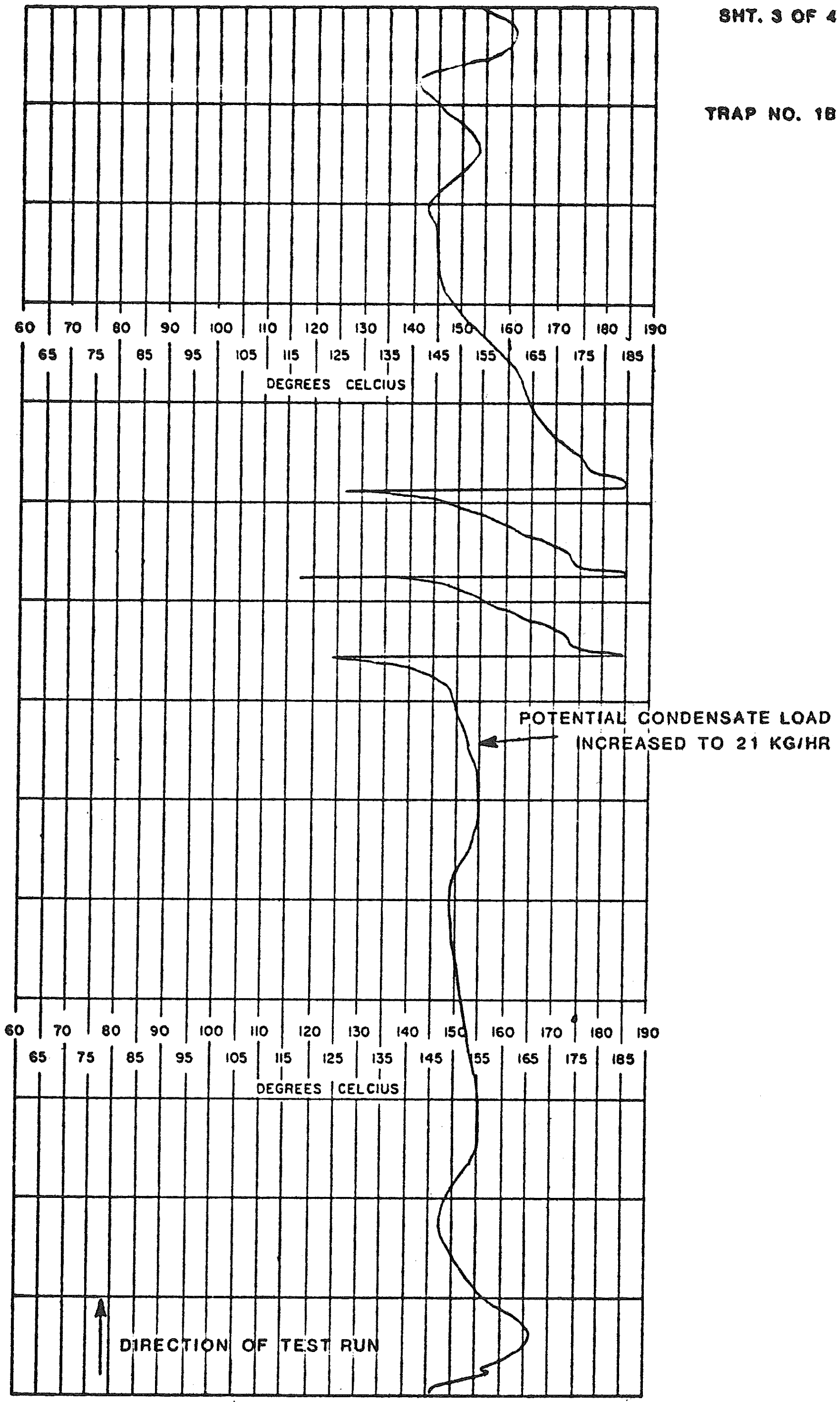
GRAPH I-2


SHT. 1 OF 4
6 MIN.
TRAP NO. 1B
STEAM TEMPERATURE ≈ 186°C
STEAM PRESSURE ≈ 11.5 BAR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
1.5 KG/HR STEAM LEAK
NO LOAD TEST
DIRECTION OF TEST RUN

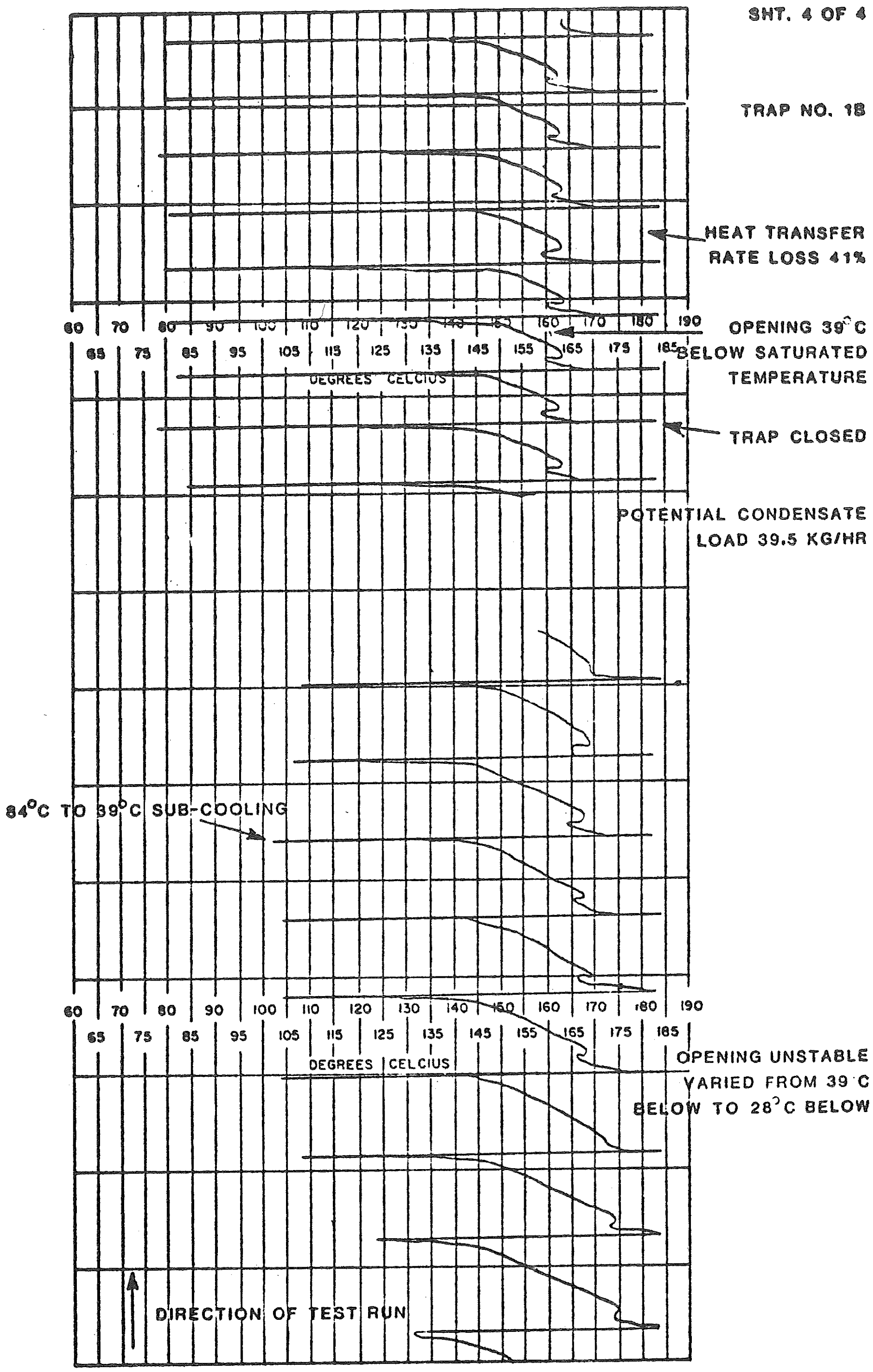
GRAPH J-1


SHT. 2 OF 4
TRAP NO. 1B
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
28°C TO 36.5°C SUB-COOLING
OPENING UNSTABLE VARIED
FROM 42°C BELOW TO
SATURATED TEMPERATURE
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE
LOAD INCREASED TO
11.5 KG/HR
DIRECTION OF TEST RUN

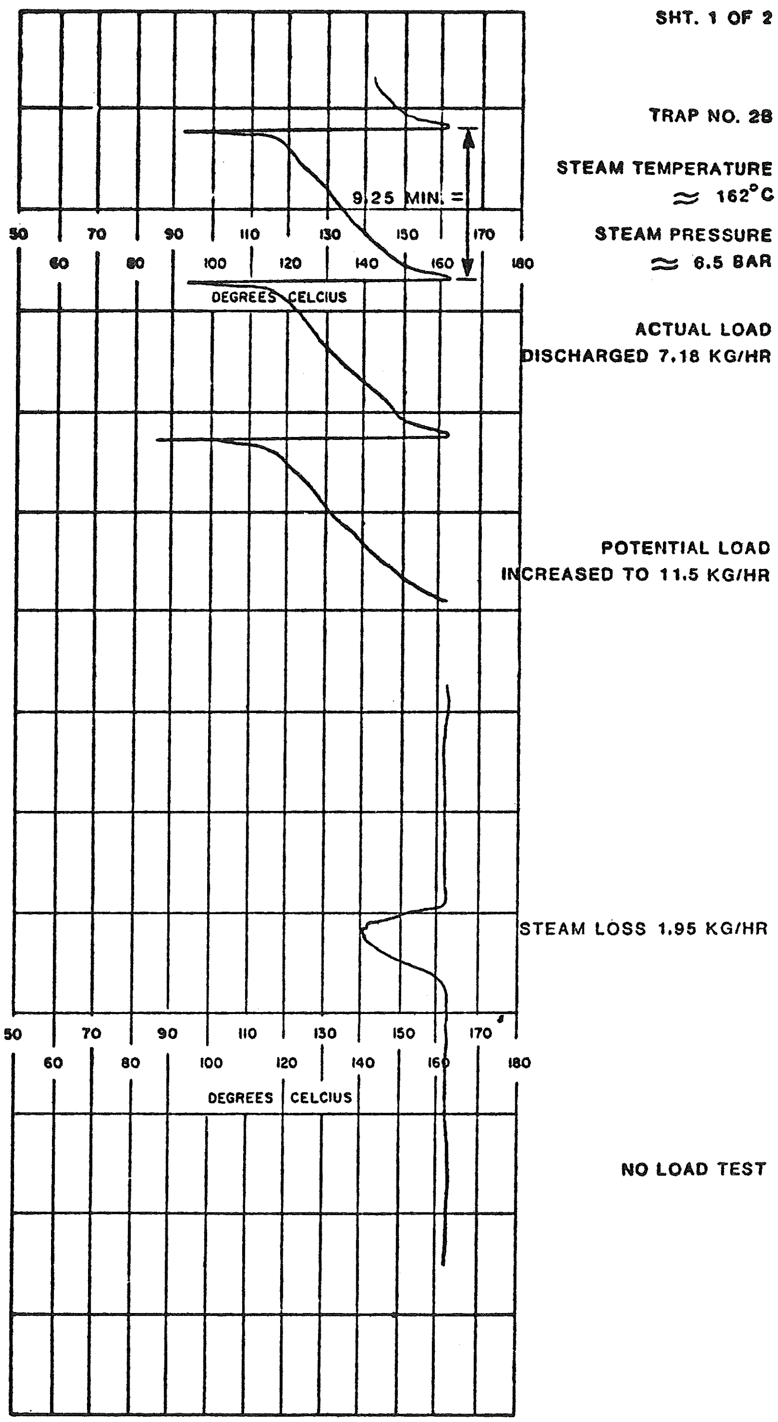
GRAPH J-2


SHT. 3 OF 4
TRAP NO. 1B
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE LOAD INCREASED TO 21 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
DIRECTION OF TEST RUN

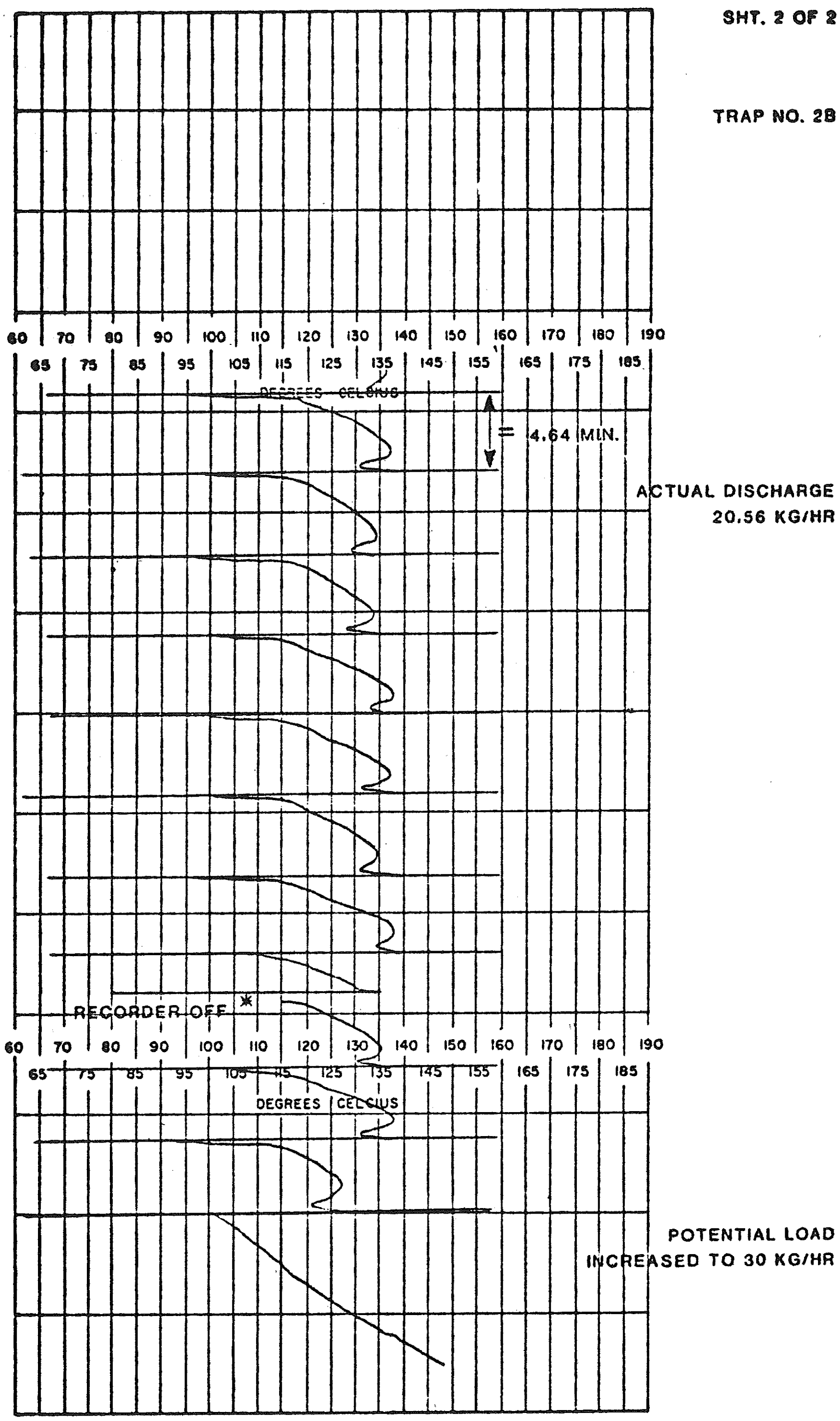
GRAPH J-3


SHT. 4 OF 4
TRAP NO. 18
HEAT TRANSFER RATE LOSS 41%
OPENING 39°C BELOW SATURATED TEMPERATURE
TRAP CLOSED
POTENTIAL CONDENSATE LOAD 39.5 KG/HR
84°C TO 39°C SUB-COOLING
OPENING UNSTABLE VARIED FROM 39°C BELOW TO 28°C BELOW
DIRECTION OF TEST RUN
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS

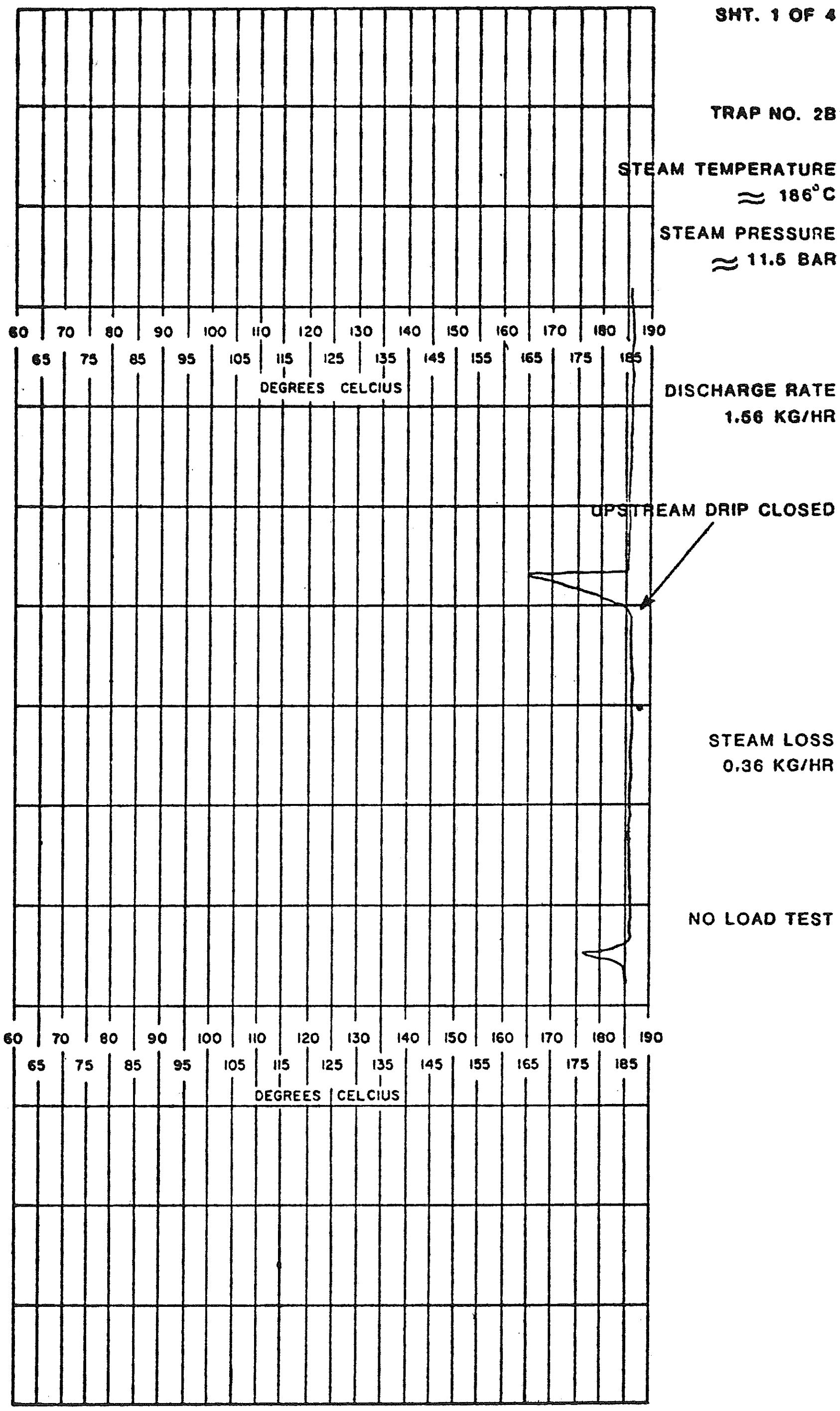
GRAPH J-4


SHT. 1 OF 2
TRAP NO. 28
STEAM TEMPERATURE
≈ 162°C
STEAM PRESSURE
≈ 6.5 BAR
9.25 MIN. =
50
60
70
80
90
100
110
120
130
140
150
160
170
180
DEGREES CELCIUS
ACTUAL LOAD
DISCHARGED 7.18 KG/HR
POTENTIAL LOAD
INCREASED TO 11.5 KG/HR
STEAM LOSS 1.95 KG/HR
NO LOAD TEST

GRAPH K-1


SHT. 2 OF 2
TRAP NO. 2B
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
= 4.64 MIN.
ACTUAL DISCHARGE
20.56 KG/HR
RECORDER OFF *
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL LOAD
INCREASED TO 30 KG/HR

GRAPH K-2


60
70
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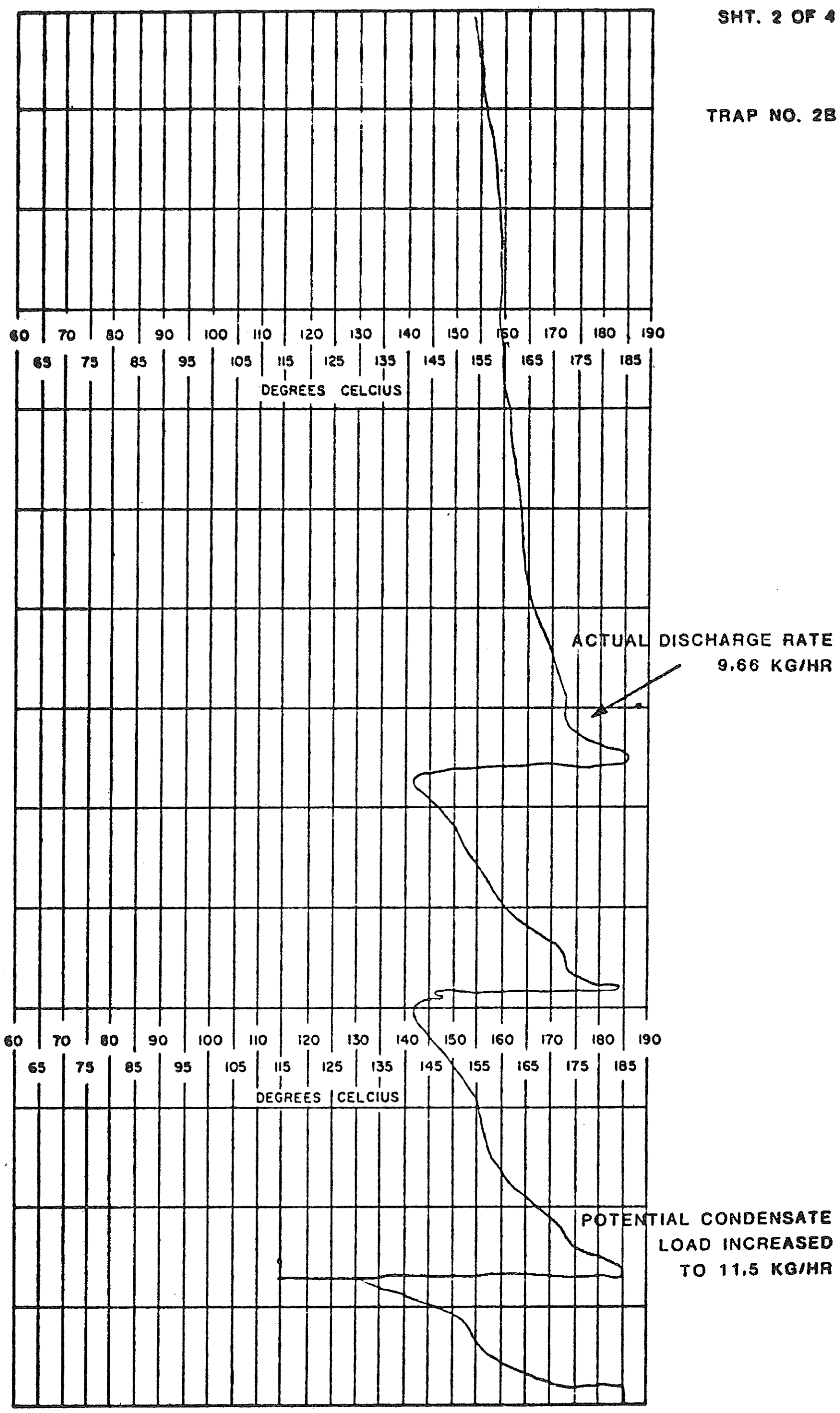
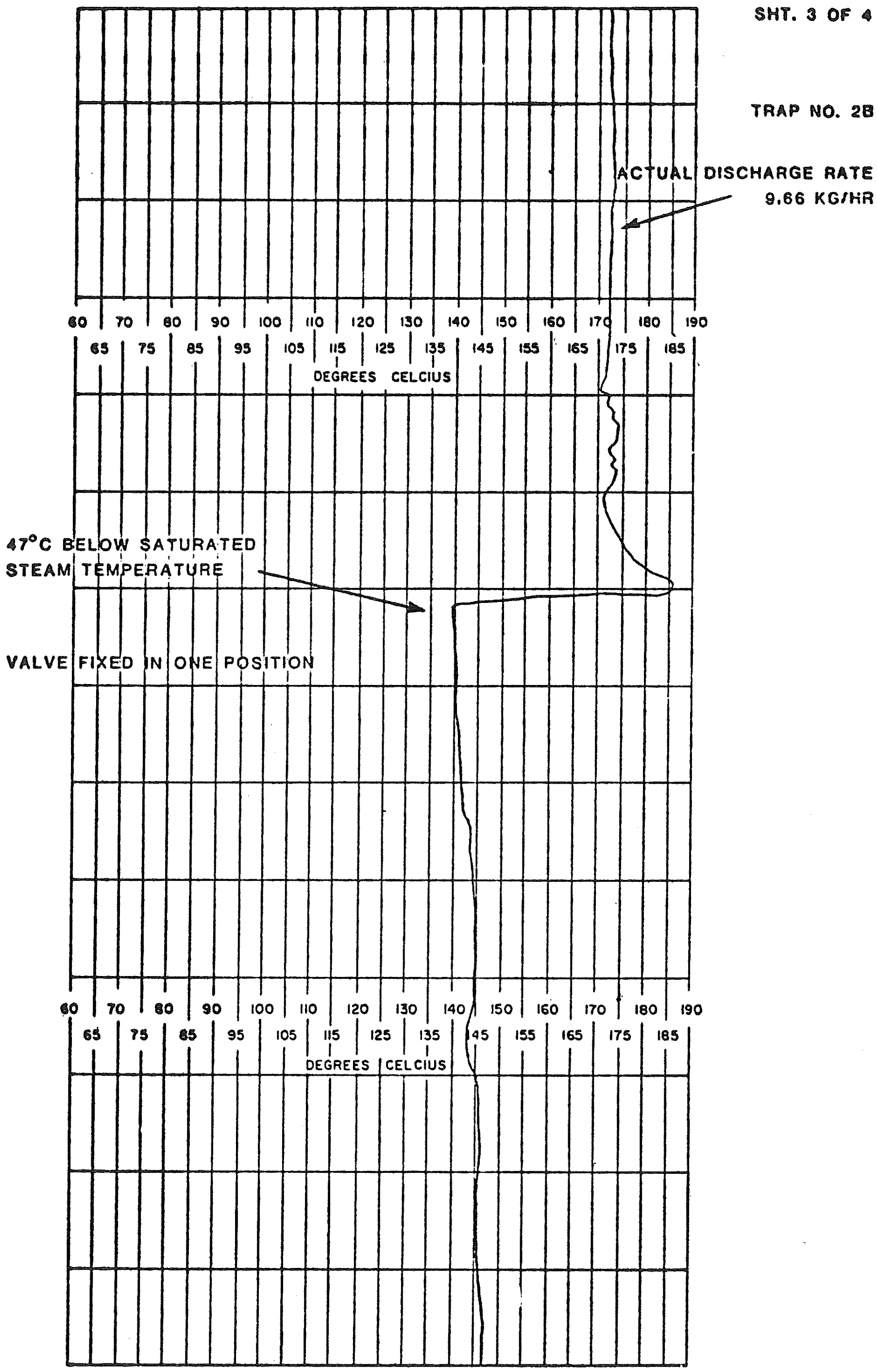
115
125
135
145
155
165
175
185
DEGREES CELCIUS
SHT. 1 OF 4
TRAP NO. 2B
STEAM TEMPERATURE
≈ 186°C
STEAM PRESSURE
≈ 11.5 BAR
DISCHARGE RATE
1.56 KG/HR
UPSTREAM DRIP CLOSED
STEAM LOSS
0.36 KG/HR
NO LOAD TEST

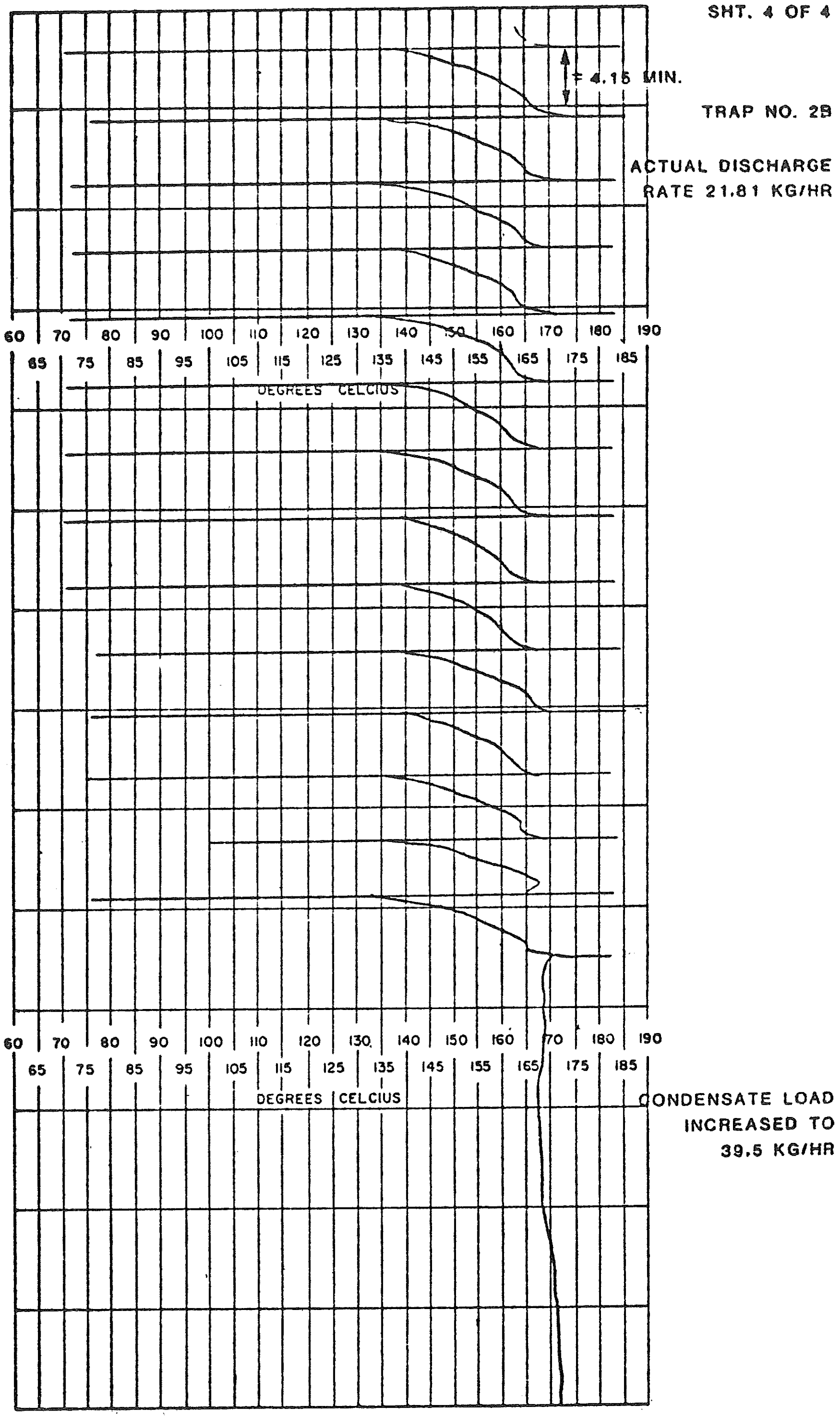
GRAPH L-1


SHT. 2 OF 4
TRAP NO. 2B
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ACTUAL DISCHARGE RATE
9.66 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE
LOAD INCREASED
TO 11.5 KG/HR

GRAPH L-2


SHT. 3 OF 4
TRAP NO. 28
ACTUAL DISCHARGE RATE
9.66 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
47°C BELOW SATURATED
STEAM TEMPERATURE
VALVE FIXED IN ONE POSITION
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS

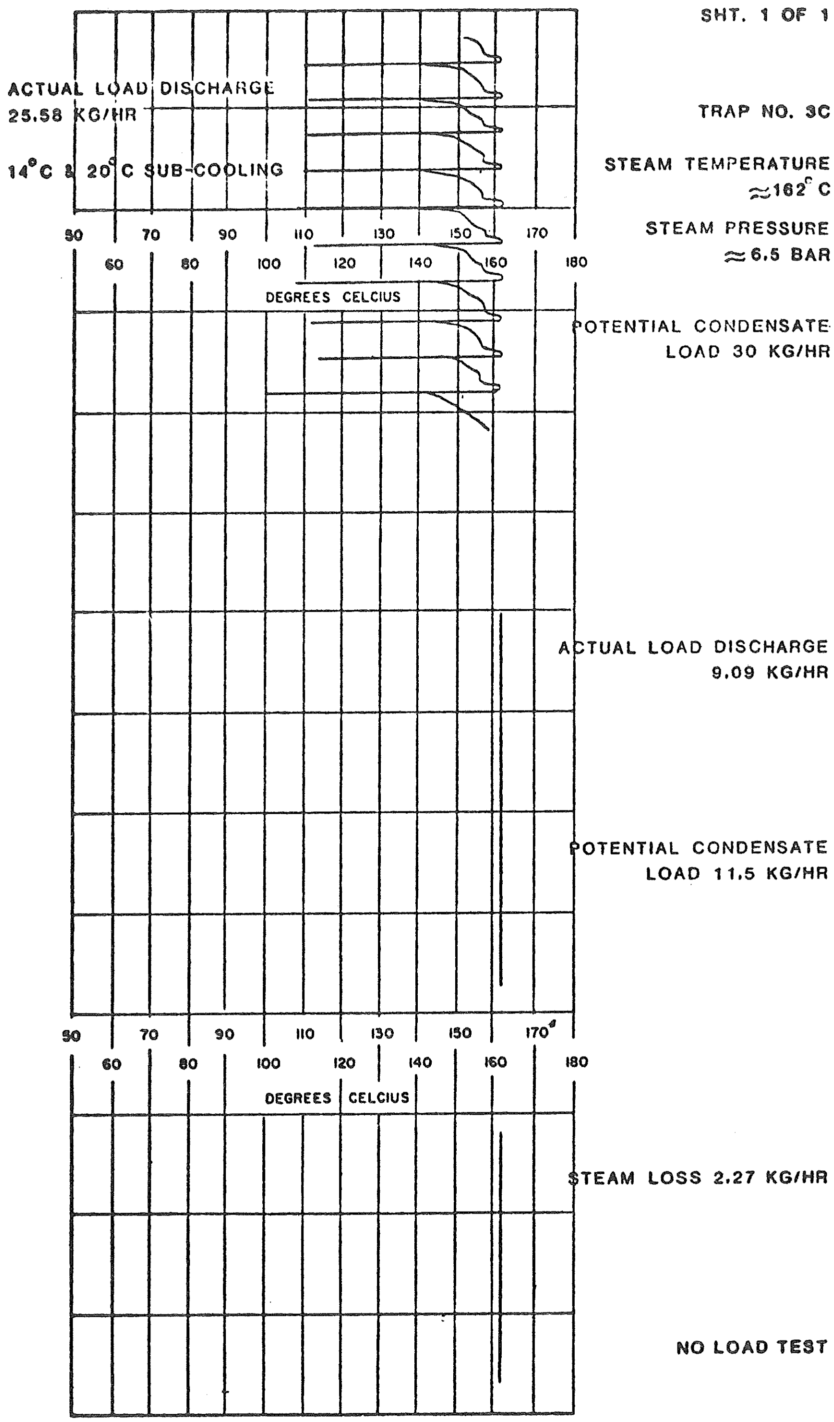
GRAPH L-3


SHT. 4 OF 4
4.15 MIN.
TRAP NO. 2B
ACTUAL DISCHARGE RATE 21.81 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
CONDENSATE LOAD INCREASED TO 39.5 KG/HR

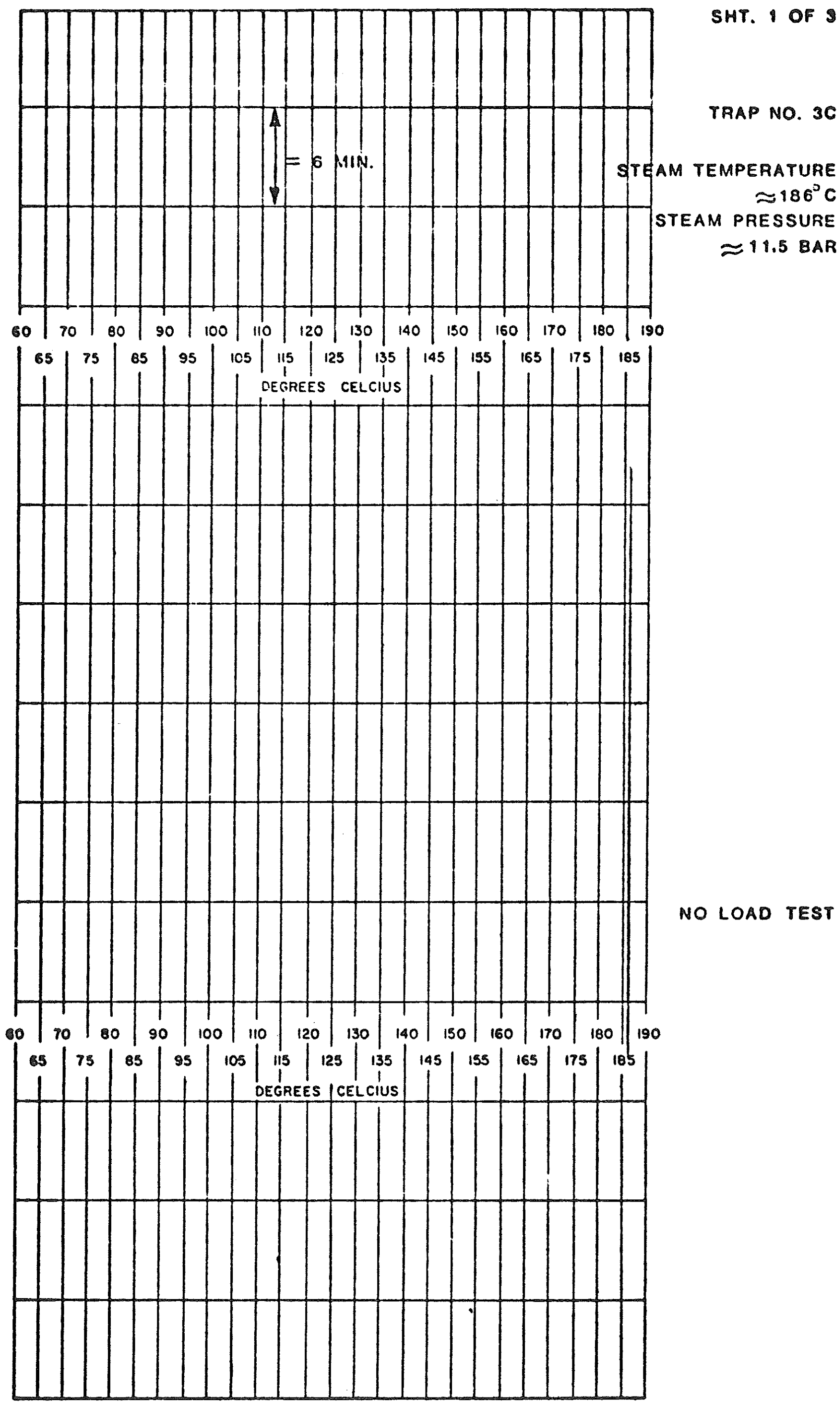
GRAPH L-4

Bimetallic trap 3C lost 2.27 kg/hr at the low pressure. At the higher pressure, no steam loss was noticeable on the no load test. (See Graphs M, N-1, N-2, and N-3.) At the low pressure, the unit took a fixed position and did not start cycling until the potential condensate load was increased to 30 kg/hr. The unit started cycling at the higher pressure when a potential load of 21.5 kg/hr was reached.

Bimetallic trap 4C lost steam at a rate of 1.13 kg/hr at low pressure, and 1.545 kg/hr at the high pressure. (See Graphs O, P-1, and P-2.) At both pressures, this unit started operating and cycling at 11.5 kg/hr. At the low pressure, the sub-cooling was -10°C; and at the high pressure it was $\approx$ 6°C. In comparing Traps 3C and 4C at the lower pressure, the actual sub-cooling had a difference between the two of approximately 5°C. At the high pressure, the difference in operation was even more obvious.


SHT. 1 OF 1
ACTUAL LOAD DISCHARGE
25.58 KG/HR
14°C & 20°C SUB-COOLING
TRAP NO. 3C
STEAM TEMPERATURE
≈162°C
STEAM PRESSURE
≈6.5 BAR
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
POTENTIAL CONDENSATE
LOAD 30 KG/HR
ACTUAL LOAD DISCHARGE
9.09 KG/HR
POTENTIAL CONDENSATE
LOAD 11.5 KG/HR
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
STEAM LOSS 2.27 KG/HR
NO LOAD TEST

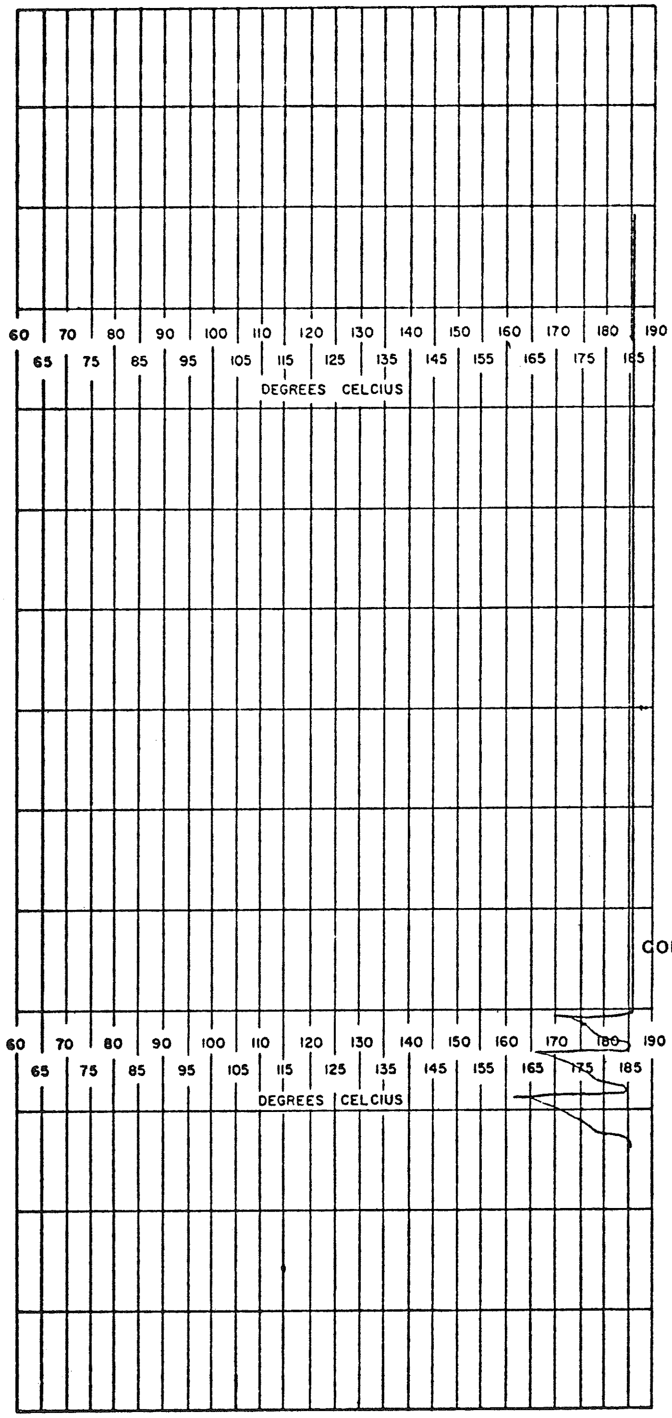
GRAPH M


SHT. 1 OF 3
TRAP NO. 3C
= 6 MIN.
STEAM TEMPERATURE ≈186°C
STEAM PRESSURE ≈11.5 BAR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
NO LOAD TEST
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS

GRAPH N-1

SHT. 2 OF 3

TRAP NO. 3C

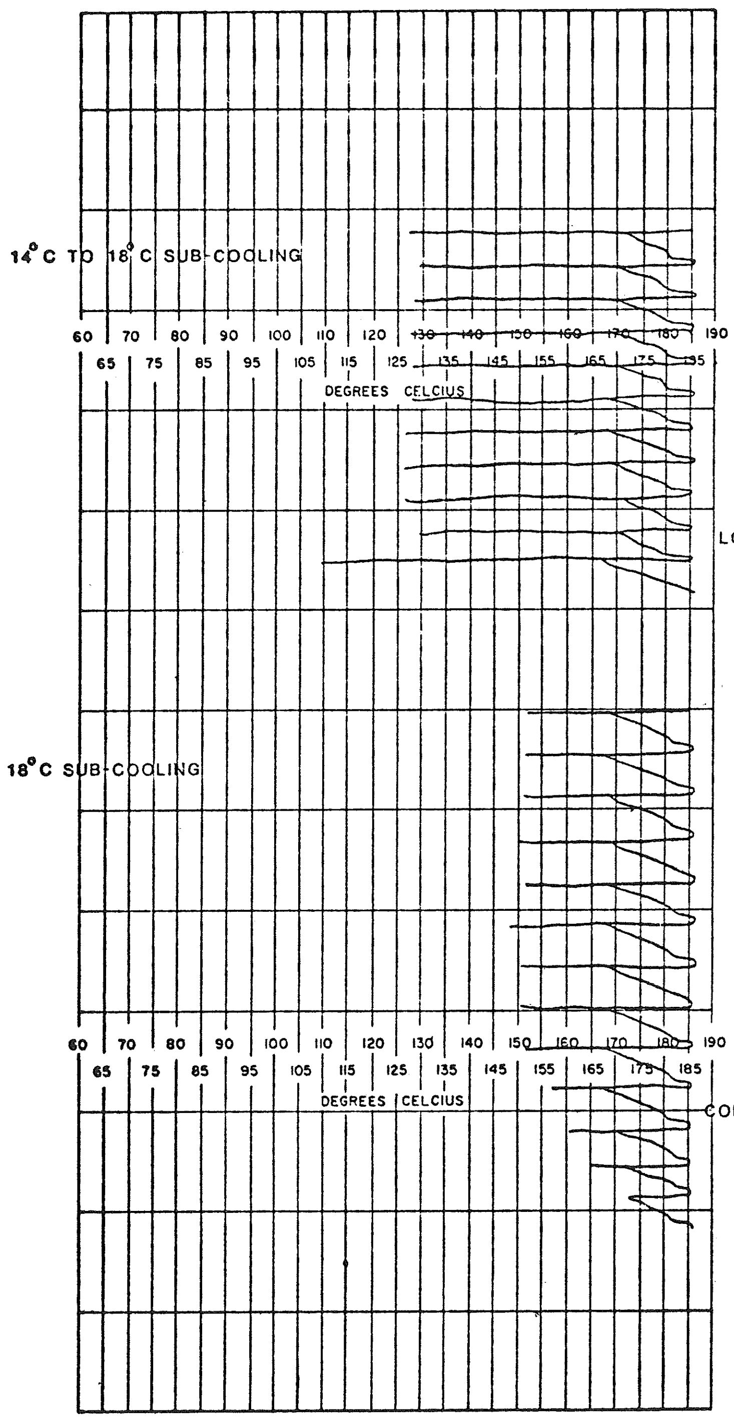


POTENTIAL
CONDENSATE LOAD
INCREASED TO
11.5 KG/HR

GRAPH N-2

SHT. 3 OF 3

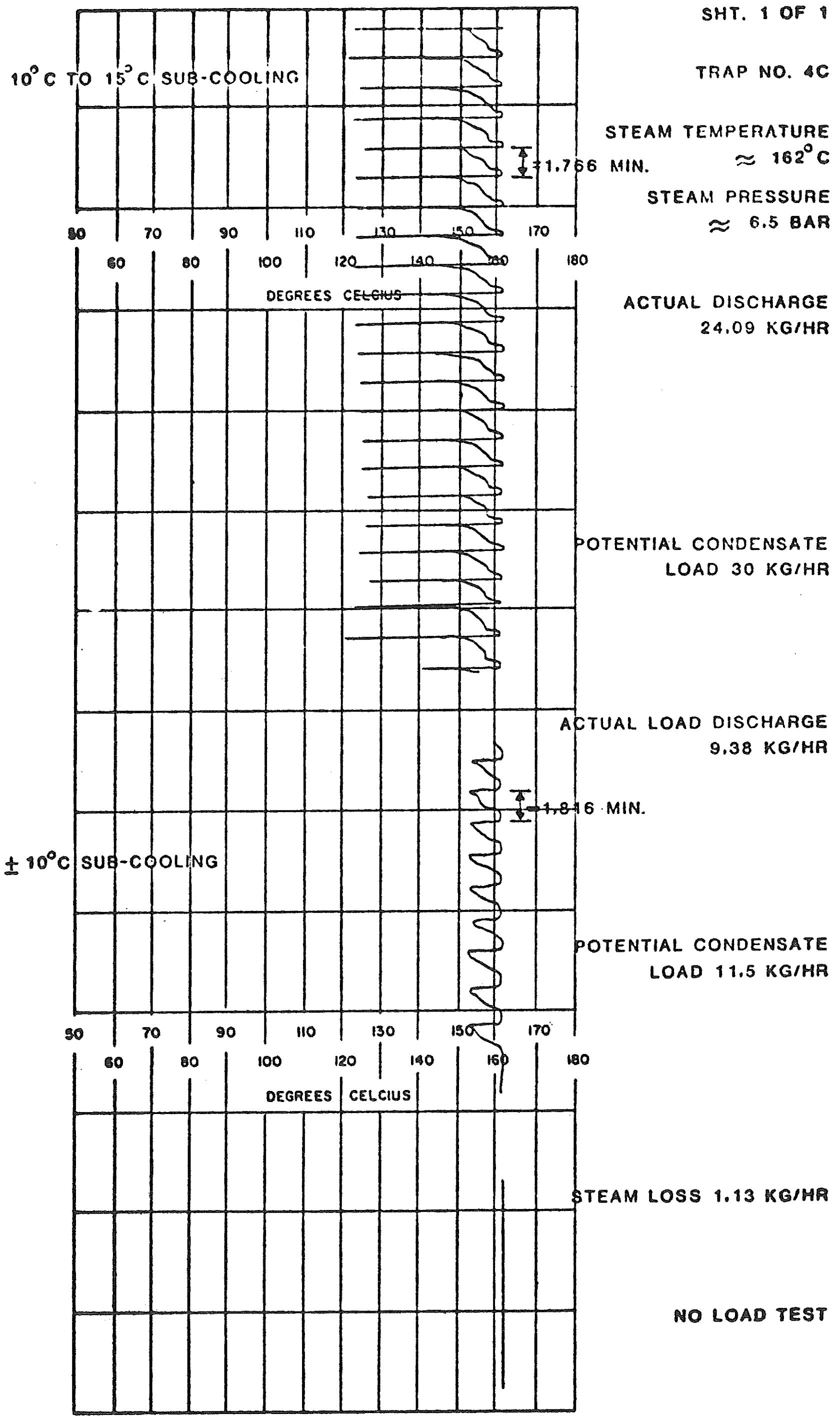
TRAP NO. 3C



POTENTIAL CONDENSATE LOAD 39.5 KG/HR

POTENTIAL CONDENSATE LOAD INCREASED TO 21.5 KG/HR

GRAPH N-3


SHT. 1 OF 1
TRAP NO. 4C
STEAM TEMPERATURE ≈ 162°C
STEAM PRESSURE ≈ 6.5 BAR
10°C TO 15°C SUB-COOLING
1.766 MIN.
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
ACTUAL DISCHARGE 24.09 KG/HR
POTENTIAL CONDENSATE LOAD 30 KG/HR
ACTUAL LOAD DISCHARGE 9.38 KG/HR
1.816 MIN.
± 10°C SUB-COOLING
POTENTIAL CONDENSATE LOAD 11.5 KG/HR
50 70 90 110 130 150 170
60 80 100 120 140 160 180
DEGREES CELCIUS
STEAM LOSS 1.13 KG/HR
NO LOAD TEST

GRAPH O

SHT. 1 OF 2

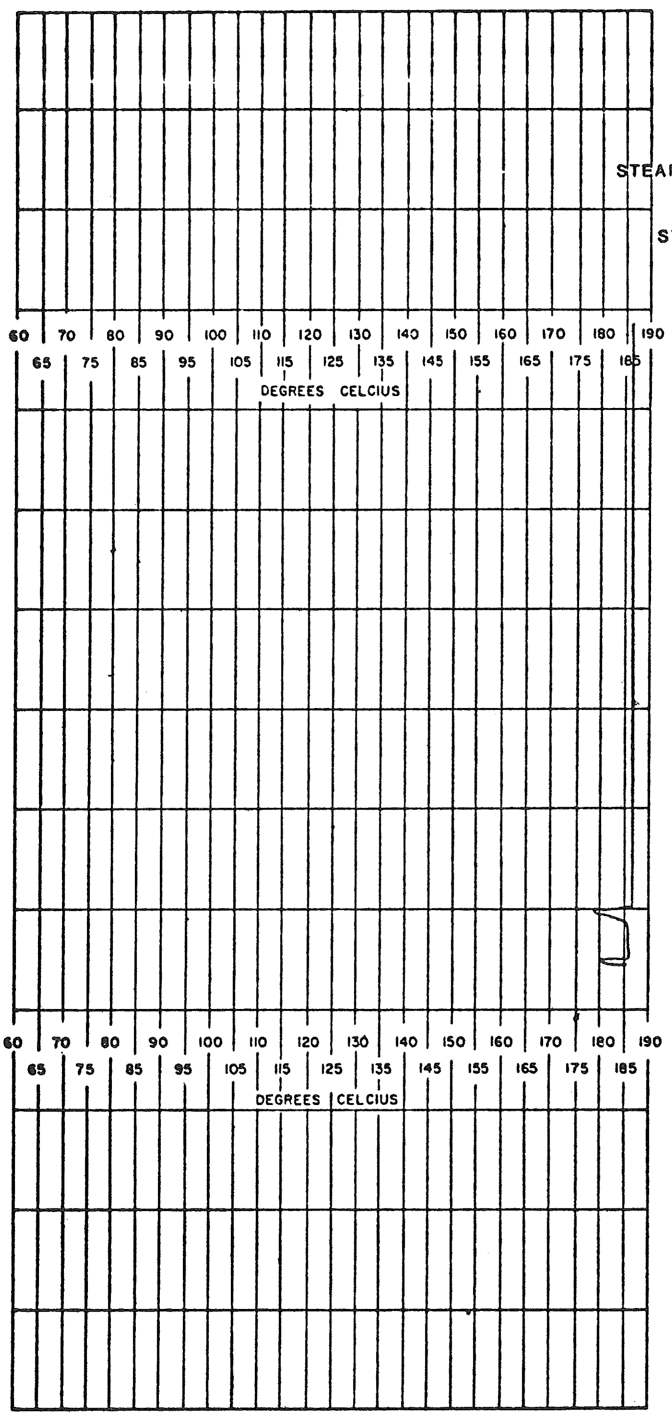
TRAP NO. 4C

STEAM TEMPERATURE
≈ 186°C

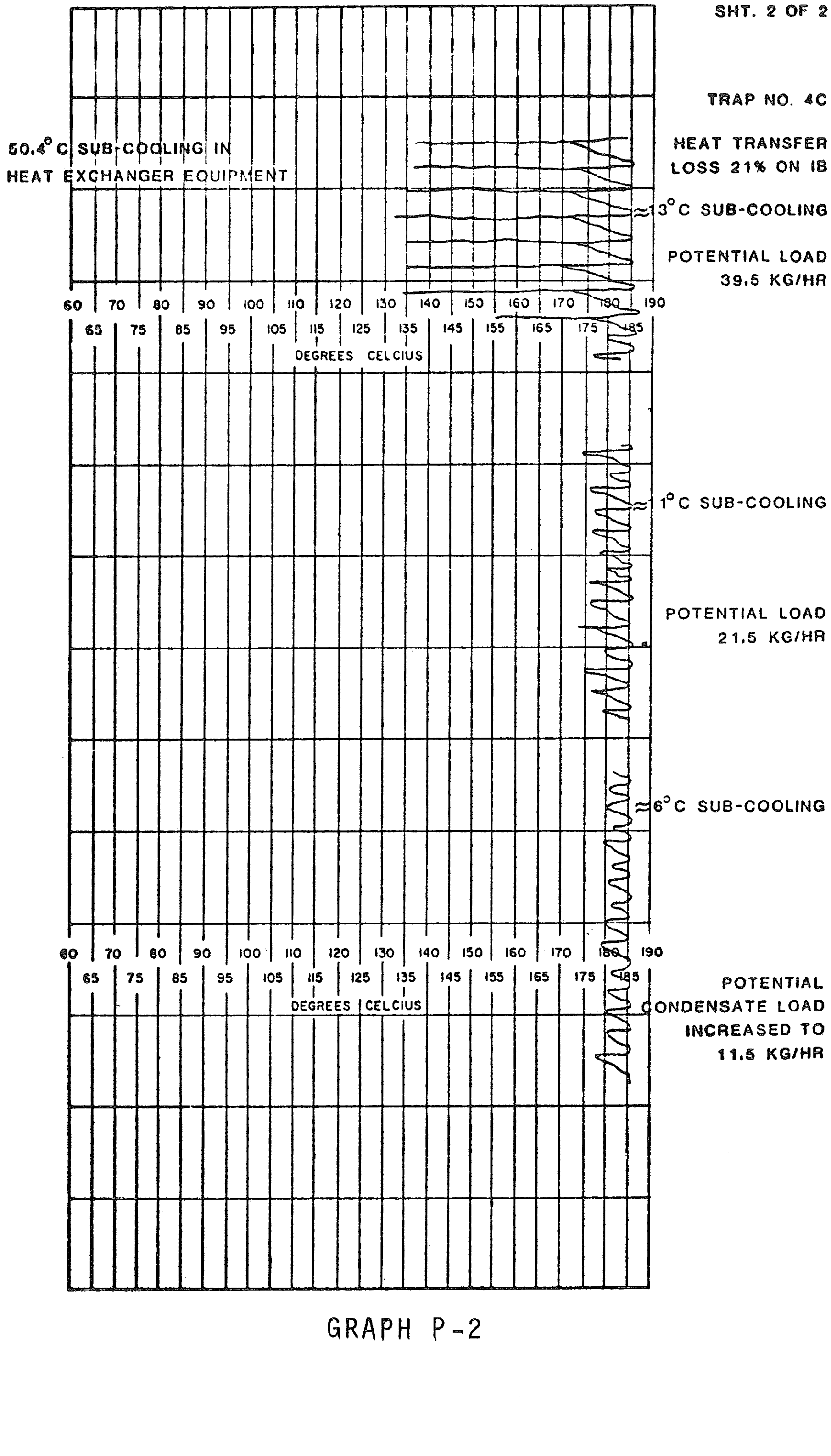
STEAM PRESSURE
≈ 11.5 BAR

STEAM LOSS
1.545 KG/HR

NO LOAD TEST



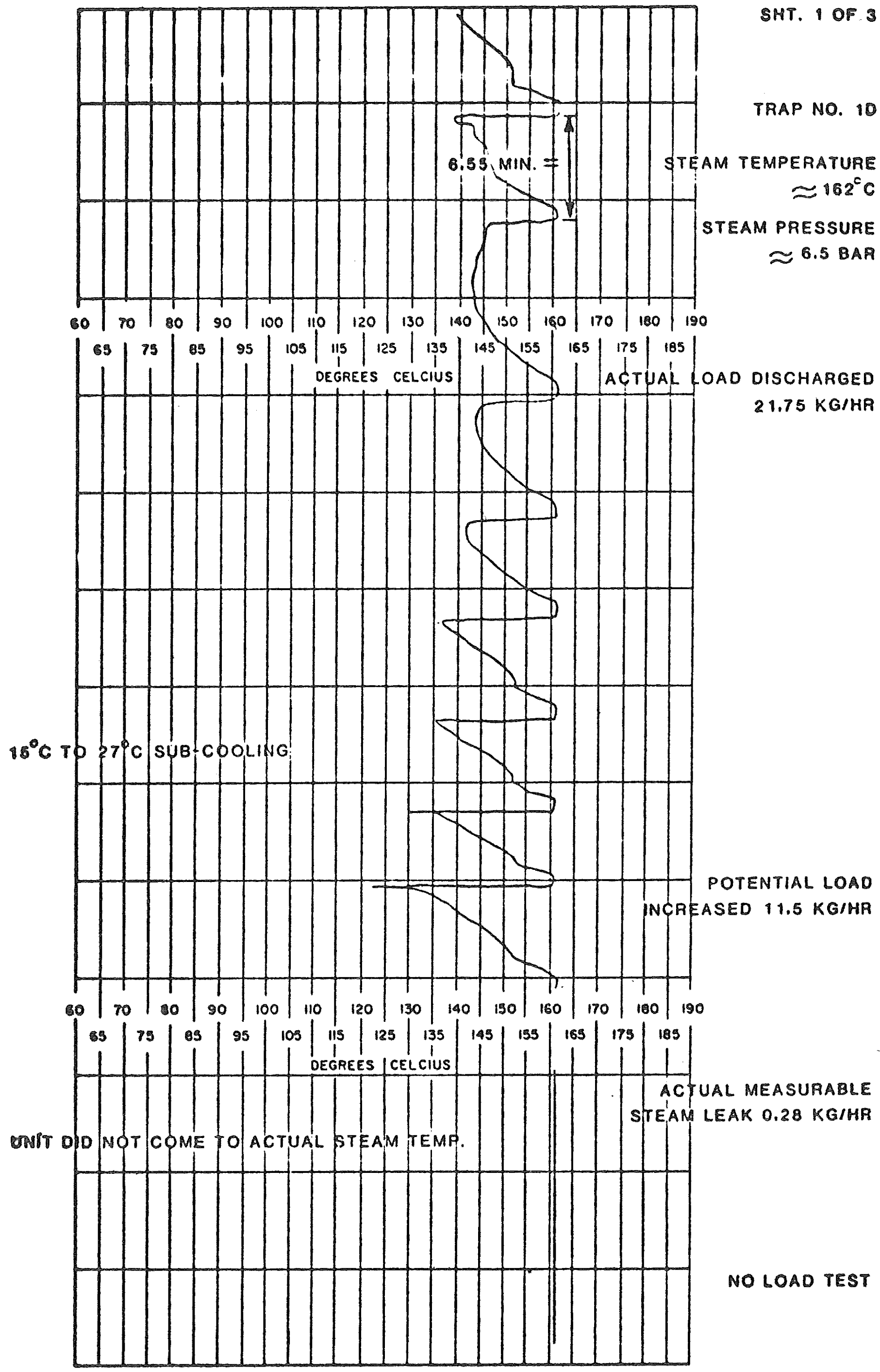
GRAPH P-1


SHT. 2 OF 2
TRAP NO. 4C
50.4°C SUB-COOLING IN
HEAT EXCHANGER EQUIPMENT
HEAT TRANSFER
LOSS 21% ON IB
≈13°C SUB-COOLING
POTENTIAL LOAD
39.5 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
≈11°C SUB-COOLING
POTENTIAL LOAD
21.5 KG/HR
≈6°C SUB-COOLING
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL
CONDENSATE LOAD
INCREASED TO
11.5 KG/HR

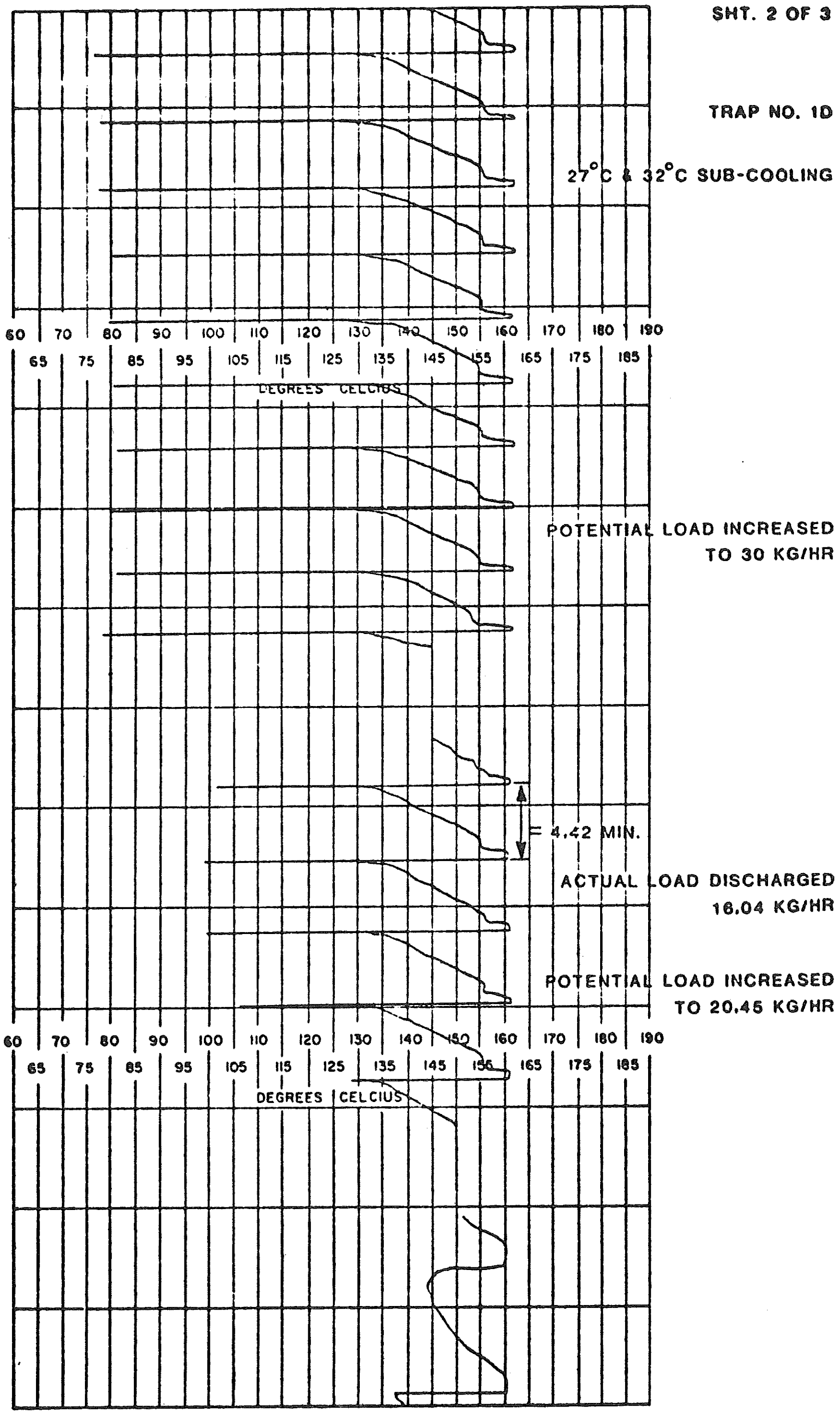
GRAPH P-2

Bimetallic trap 1D had a .28 kg/hr steam loss at low pressure during the no load test, and dropped to .25 kg/hr steam loss at the higher pressure. (See Graphs Q-1, Q-2, Q-3, R-1, and R-2.) By increasing the condensate load in both cases to 11.5 kg/hr, sub-coolings of 15 to 27°C at the low pressure and 28°C at the higher pressure occurred. At the potential condensate load of 21 kg/hr, the total sub-cooling at the high pressure was 50.9°C and 62°C at the low pressure. At the higher potential loads, sub-cooling totaled over 80°C at the high pressure.

In conclusion, we found that no bimetallic trap, when exposed to different pressures, performed consistently under the same condensate load. Also, the potential condensate load was not achieved by any of the bimetallic traps. Although the capacities were well within the units' range, the opening curves of the units were too far below steam temperature to react to the conditions of the system. Because of this, the settings of the units were adjusted according to the manufacturer's instructions. By doing so, the capacities of the units should have increased or the units should have operated closer to steam temperature.


SHT. 1 OF 3
TRAP NO. 1D
6.55 MIN. =
STEAM TEMPERATURE
≈ 162°C
STEAM PRESSURE
≈ 6.5 BAR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ACTUAL LOAD DISCHARGED
21.75 KG/HR
15°C TO 27°C SUB-COOLING
POTENTIAL LOAD
INCREASED 11.5 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ACTUAL MEASURABLE
STEAM LEAK 0.28 KG/HR
UNIT DID NOT COME TO ACTUAL STEAM TEMP.
NO LOAD TEST

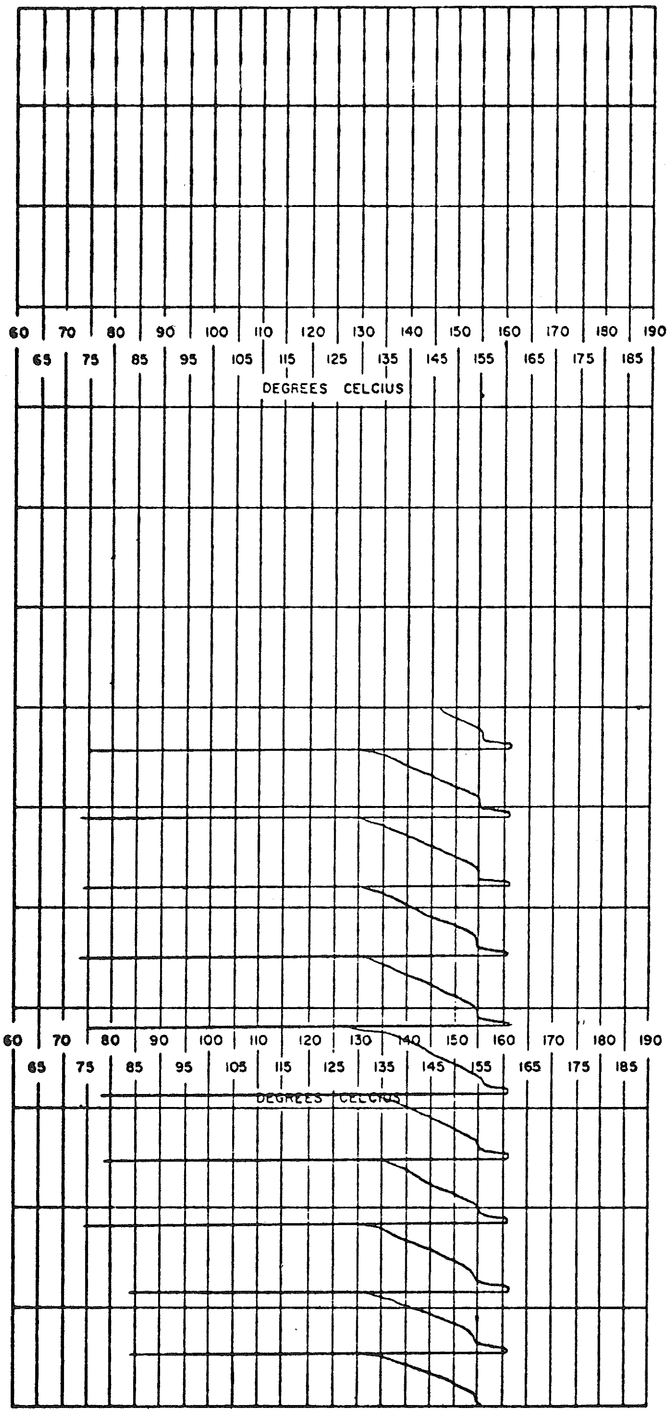
GRAPH Q-1


SHT. 2 OF 3
TRAP NO. 1D
27°C & 32°C SUB-COOLING
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL LOAD INCREASED TO 30 KG/HR
= 4.42 MIN.
ACTUAL LOAD DISCHARGED 16.04 KG/HR
POTENTIAL LOAD INCREASED TO 20.45 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS

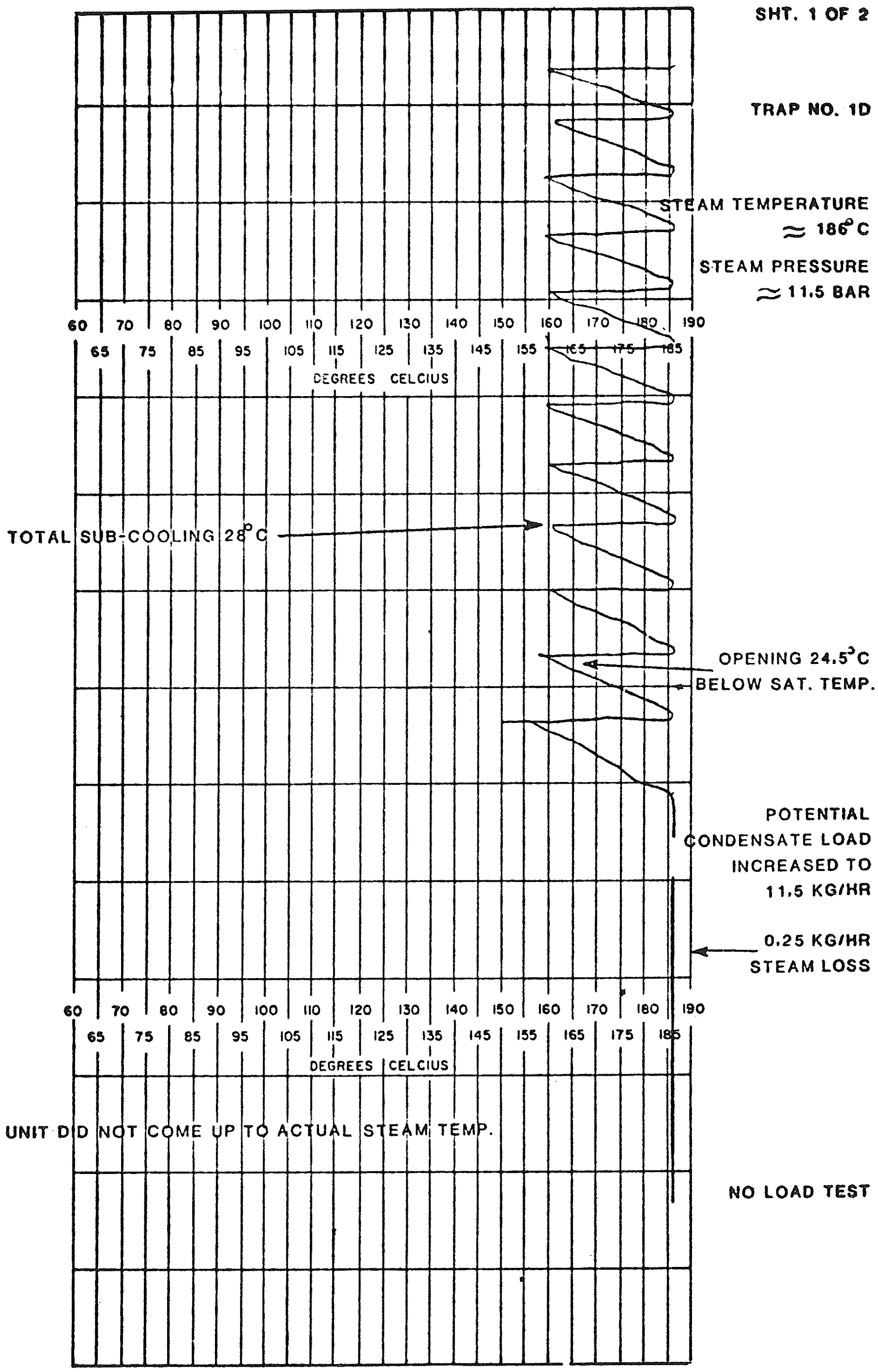
GRAPH Q-2

SHT. 3 OF 3

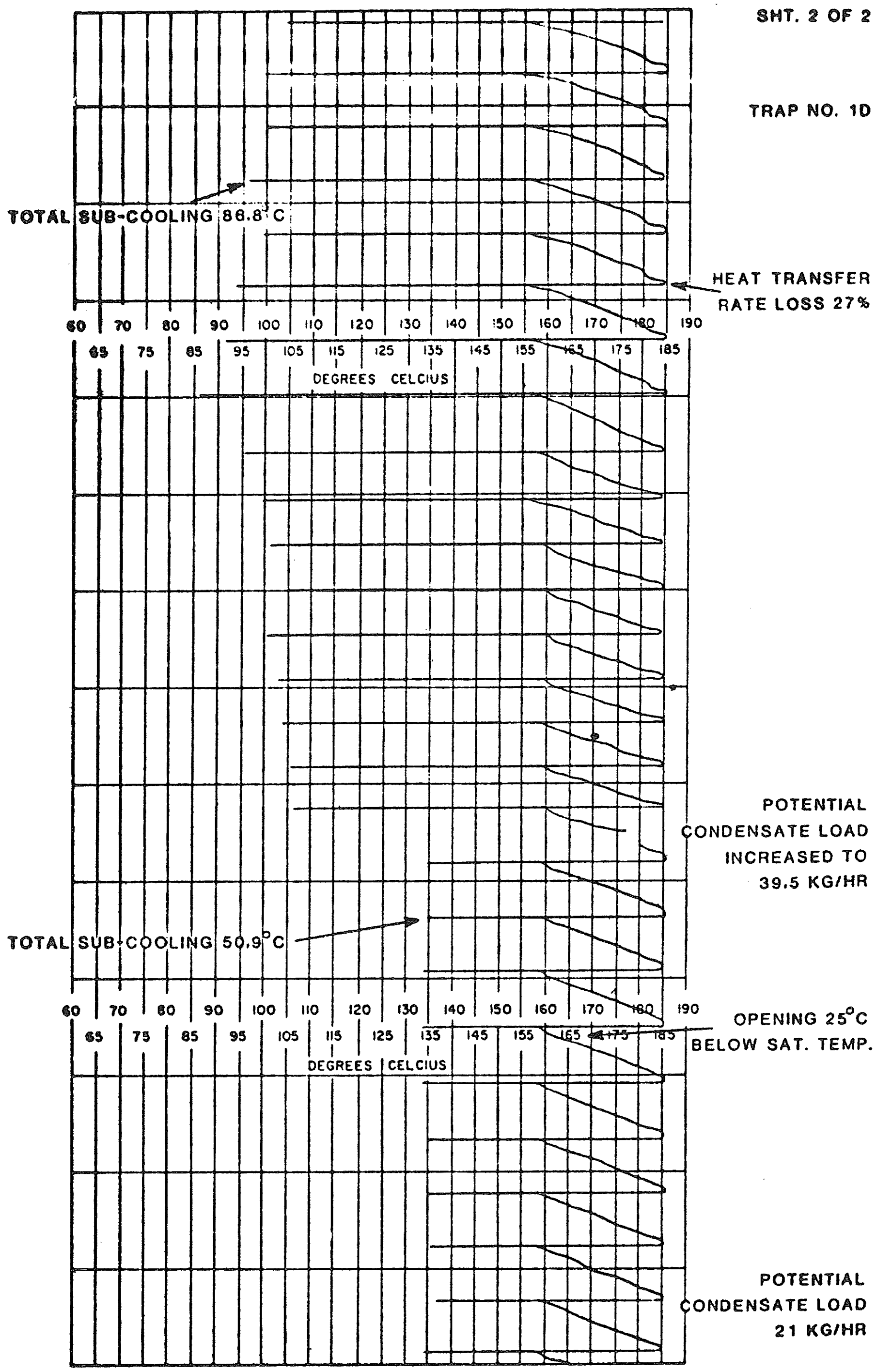
TRAP NO. 1D



GRAPH Q-3


SHT. 1 OF 2
TRAP NO. 1D
STEAM TEMPERATURE
≈ 186°C
STEAM PRESSURE
≈ 11.5 BAR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
TOTAL SUB-COOLING 28°C
OPENING 24.5°C
BELOW SAT. TEMP.
POTENTIAL
CONDENSATE LOAD
INCREASED TO
11.5 KG/HR
0.25 KG/HR
STEAM LOSS
UNIT DID NOT COME UP TO ACTUAL STEAM TEMP.
NO LOAD TEST

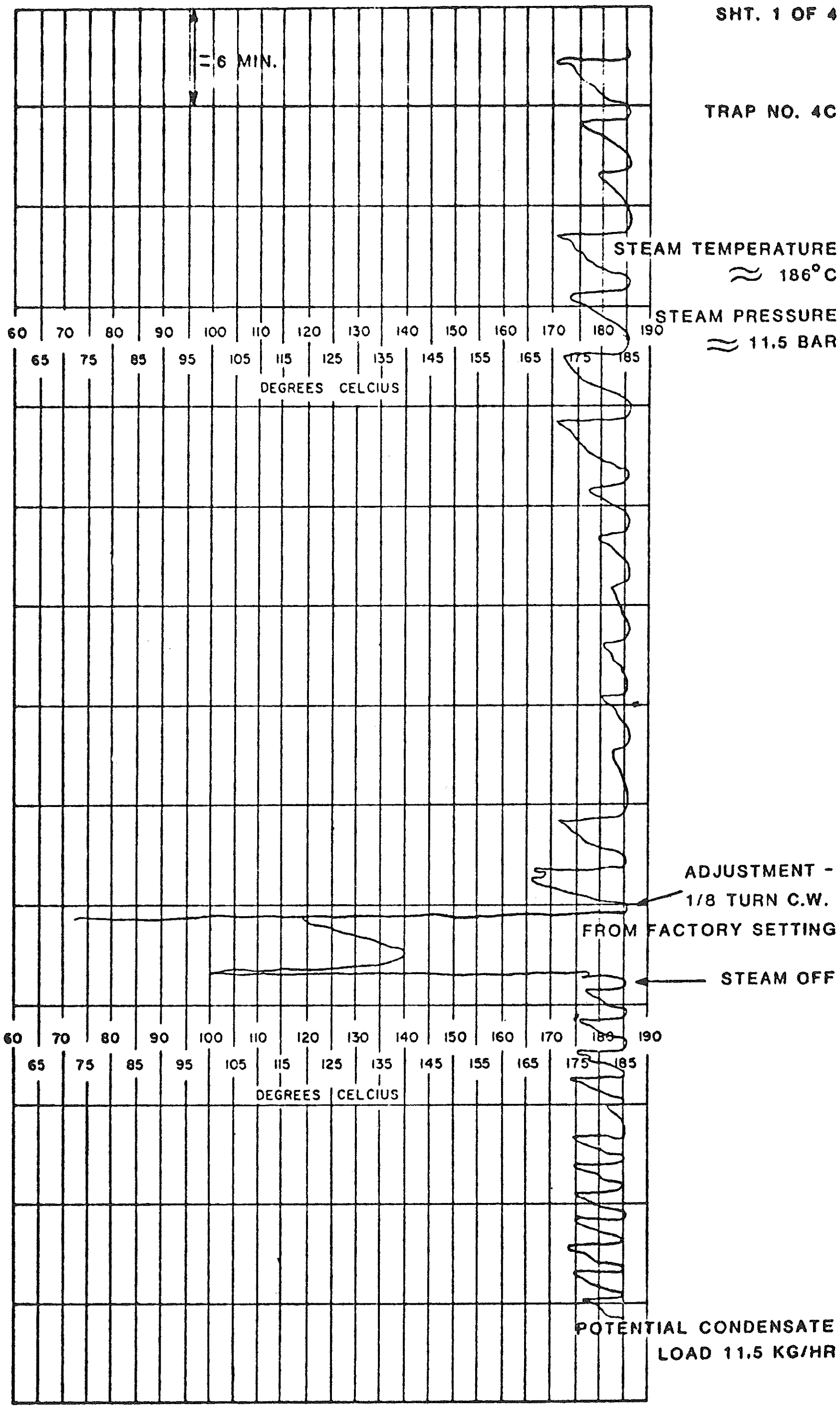
GRAPH R-1


SHT. 2 OF 2
TRAP NO. 1D
TOTAL SUB-COOLING 86.8°C
HEAT TRANSFER RATE LOSS 27%
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE LOAD INCREASED TO 39.5 KG/HR
TOTAL SUB-COOLING 50.9°C
OPENING 25°C BELOW SAT. TEMP.
POTENTIAL CONDENSATE LOAD 21 KG/HR

GRAPH R-2

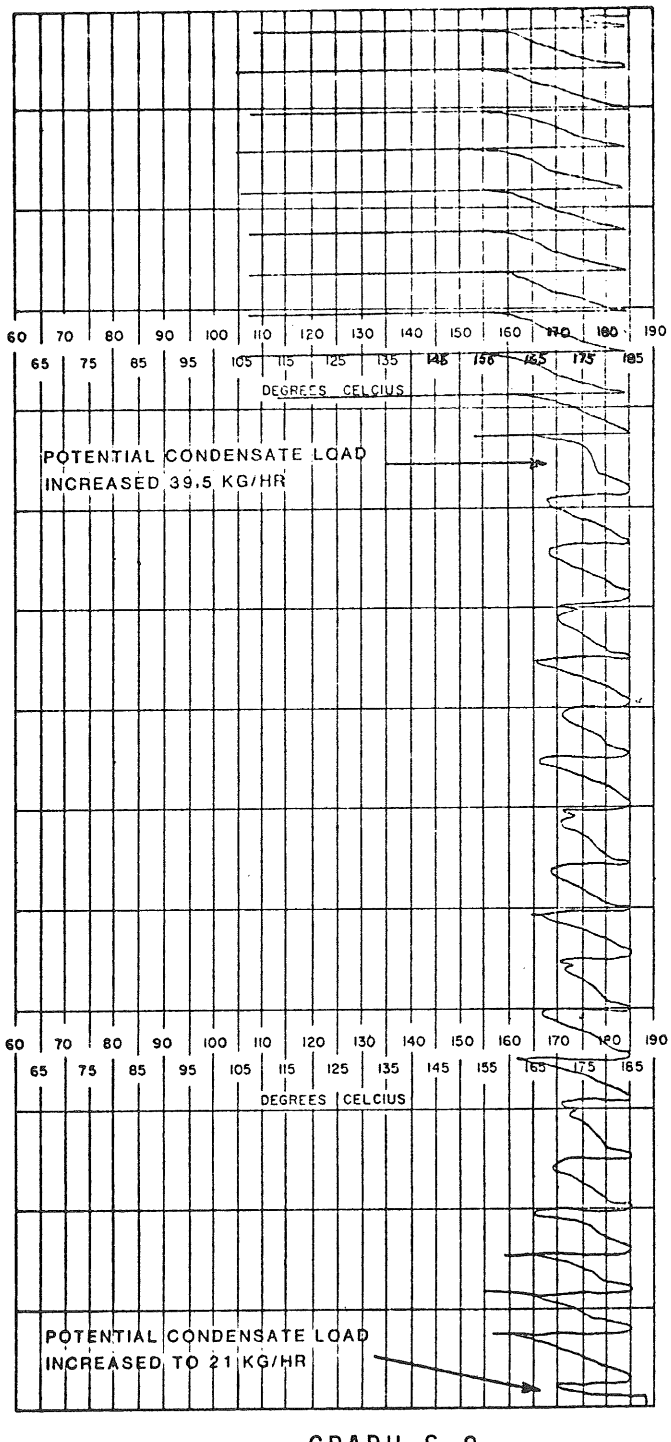
Graphs S-1, S-2, S-3, S-4, T-1, T-2, T-3, T-4, T-5, T-6, and T-7 show the results of changing the temperature settings closer to the saturated steam temperature curve. These two traps were previously used on our performance test. Please note the difference in operation of the two units from the very beginning. Trap 4C, which was easier to adjust than any of the other bimetallic traps, was still irregular, erratic, and extremely difficult to control. Changing the temperature setting either closer to or farther from the steam curve was quite difficult and was, in many cases, not attainable. Therefore, retesting the adjusted units was not possible.

Because of the irregular behavior of the bimetallic trap, the problem was studied further and the test setup changed by installing the traps at different lengths from the heat exchanger. The results in Graph U show that the length of uninsulated pipe and the condensate load have a direct bearing upon the trap performance. The first set of curves show an inverted bucket trap under the same conditions as Trap 3C, a bimetallic trap. The temperature shown were measured directly ahead of those traps.

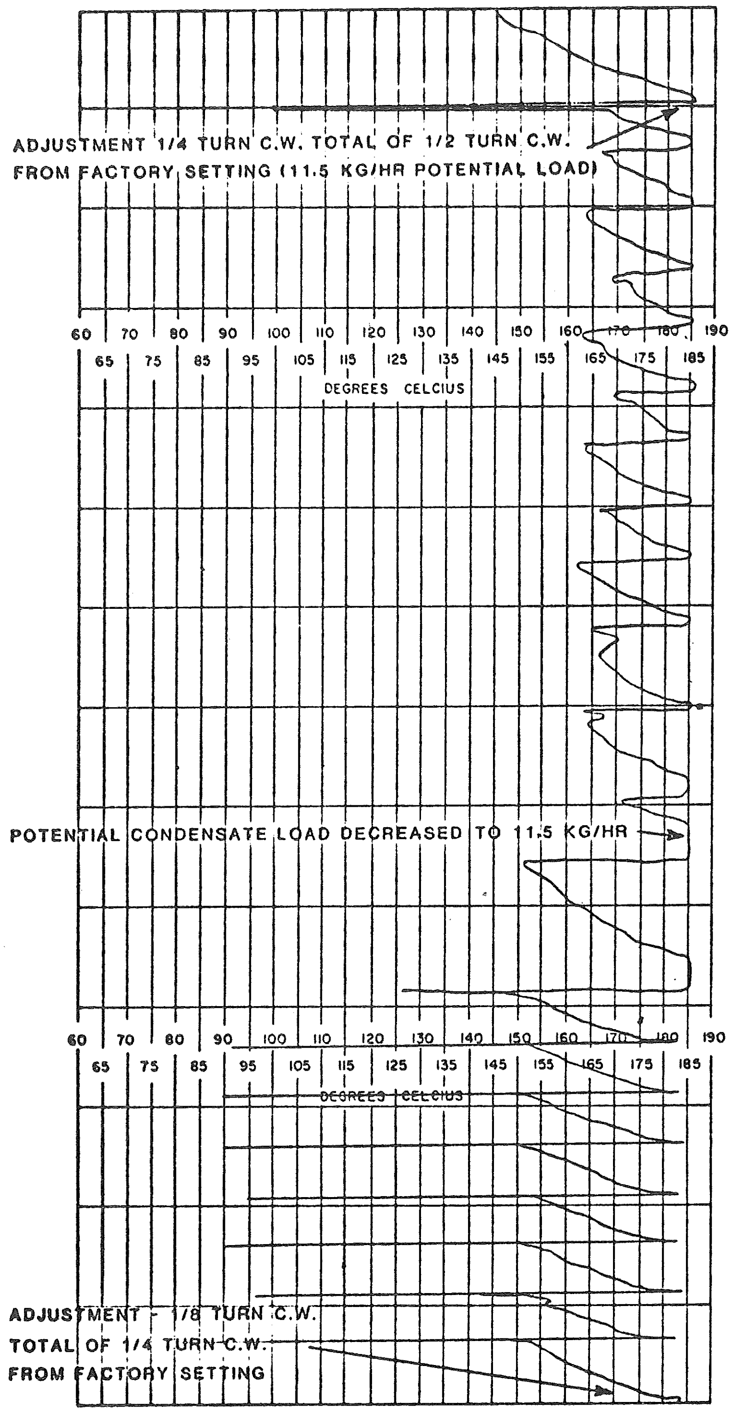

SHT. 1 OF 4
= 6 MIN.
TRAP NO. 4C
STEAM TEMPERATURE
≈ 186°C
STEAM PRESSURE
≈ 11.5 BAR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ADJUSTMENT -
1/8 TURN C.W.
FROM FACTORY SETTING
STEAM OFF
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE
LOAD 11.5 KG/HR

GRAPH S-1

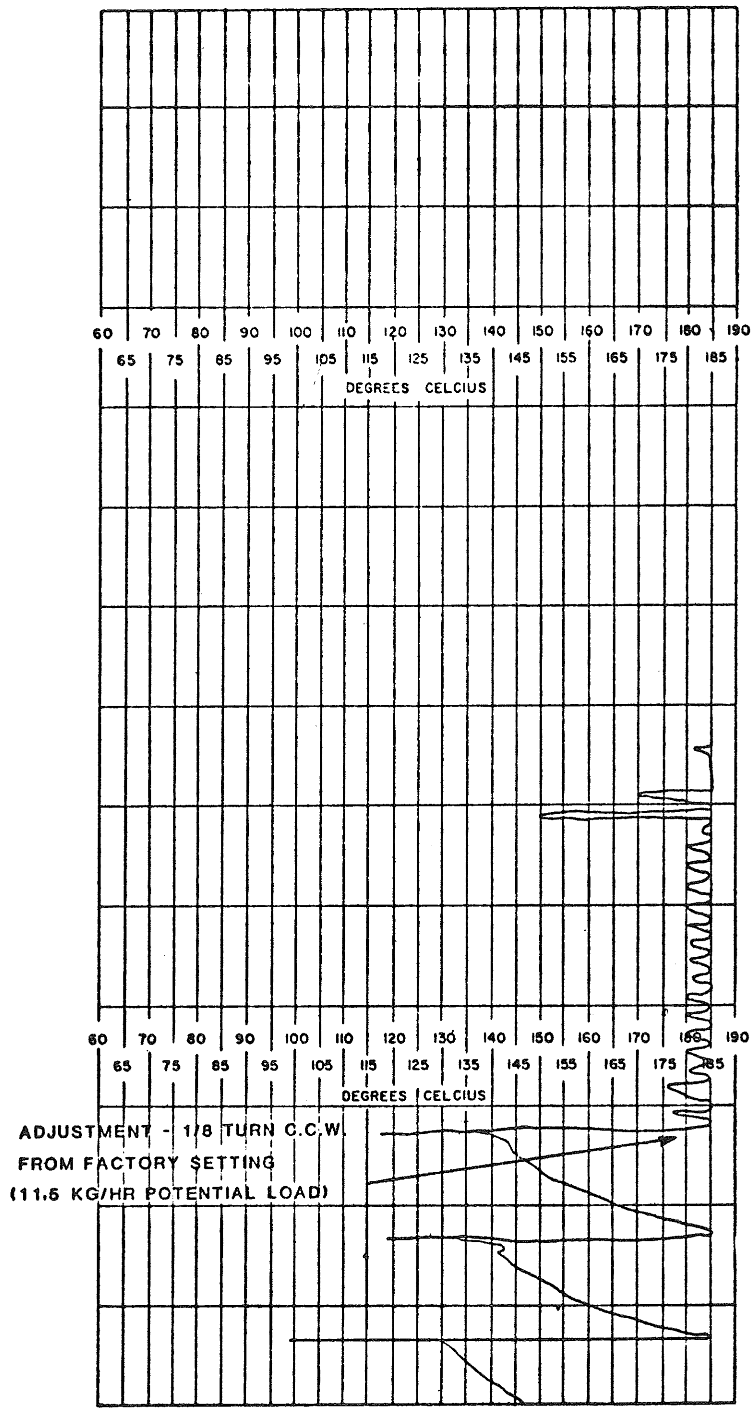
SHT. 2 OF 4



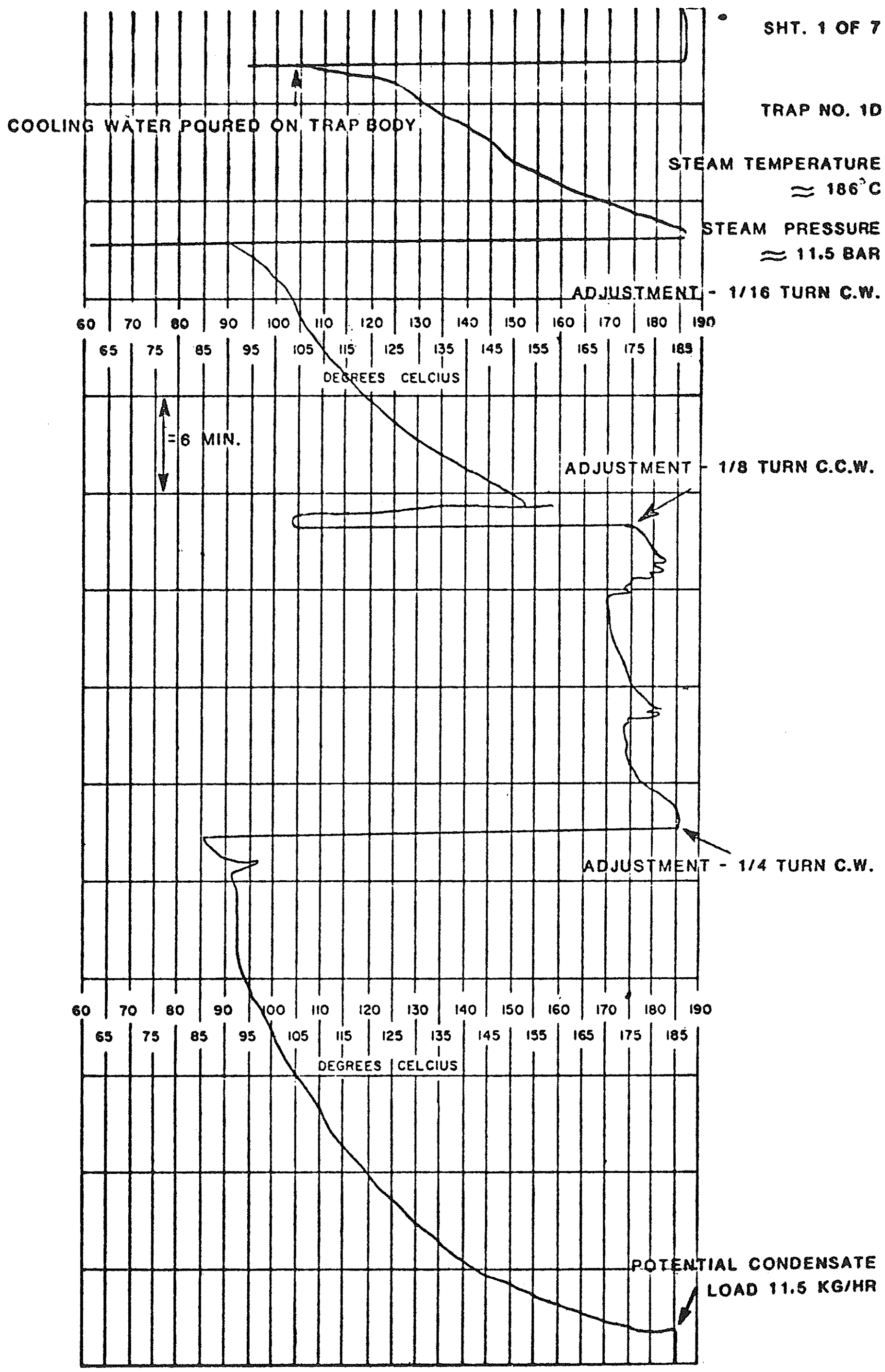
GRAPH S-2


ADJUSTMENT 1/4 TURN C.W. TOTAL OF 1/2 TURN C.W. FROM FACTORY SETTING (11.5 KG/HR POTENTIAL LOAD)
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE LOAD DECREASED TO 11.5 KG/HR
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ADJUSTMENT - 1/8 TURN C.W. TOTAL OF 1/4 TURN C.W. FROM FACTORY SETTING

GRAPH S-3

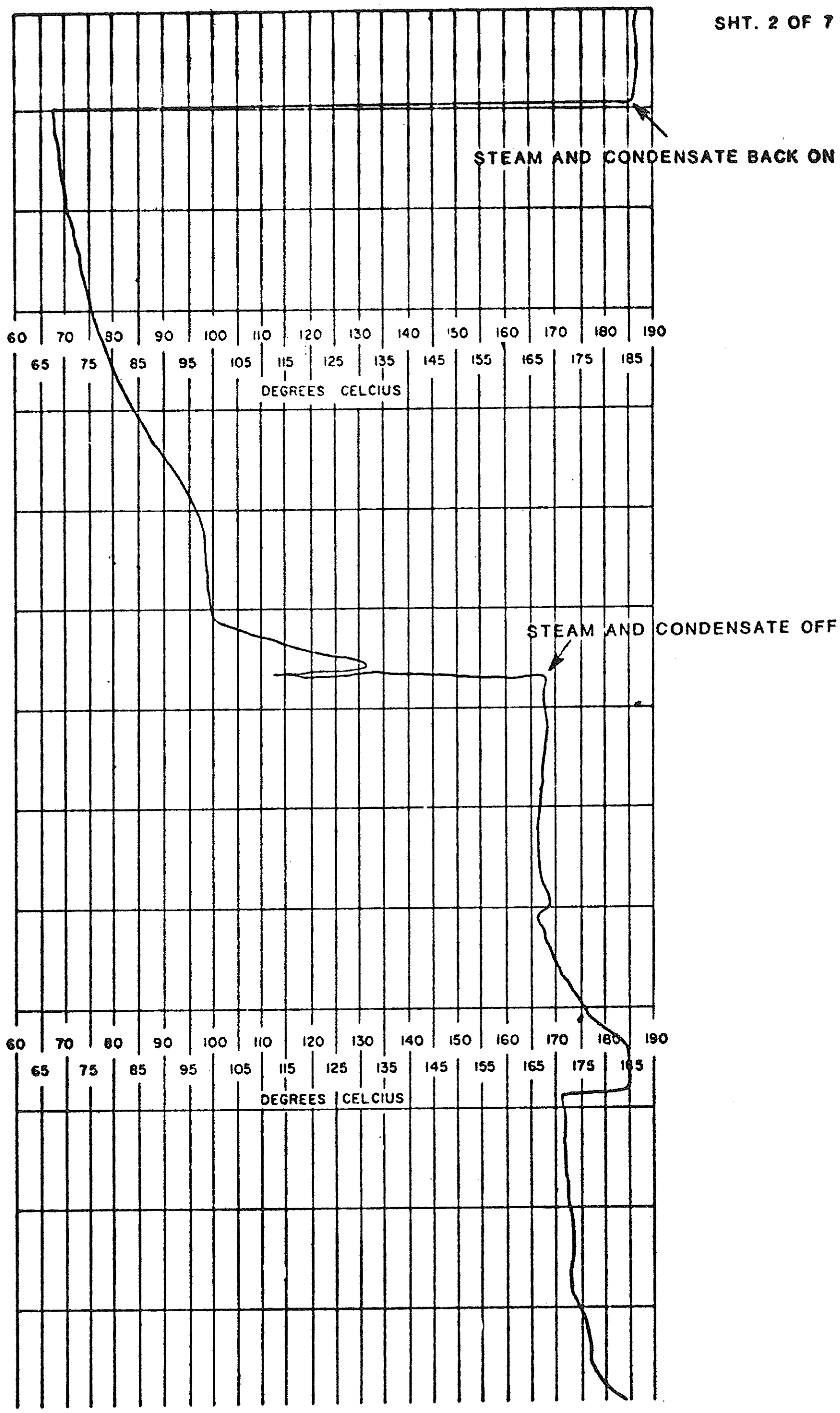

60
70
80
90
100
110
120
130
140
150
160
170
180
190
65
75
85
95
105
115
125
135
145
155
165
175
185
DEGREES CELCIUS
ADJUSTMENT - 1/8 TURN C.C.W.
FROM FACTORY SETTING
(11.5 KG/HR POTENTIAL LOAD)

GRAPH S-4

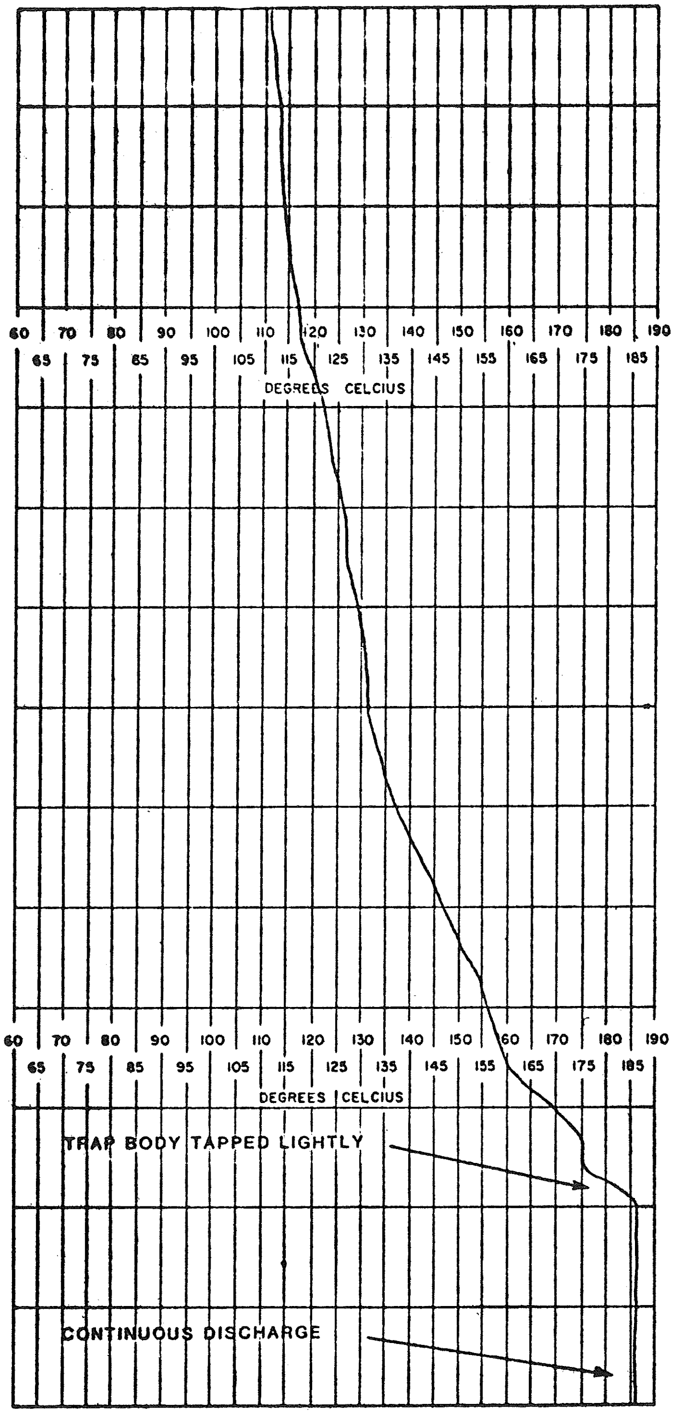

SHT. 1 OF 7
TRAP NO. 1D
COOLING WATER POURED ON TRAP BODY
STEAM TEMPERATURE ≈ 186°C
STEAM PRESSURE ≈ 11.5 BAR
ADJUSTMENT - 1/16 TURN C.W.
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
=6 MIN.
ADJUSTMENT - 1/8 TURN C.C.W.
ADJUSTMENT - 1/4 TURN C.W.
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE LOAD 11.5 KG/HR

GRAPH T-1

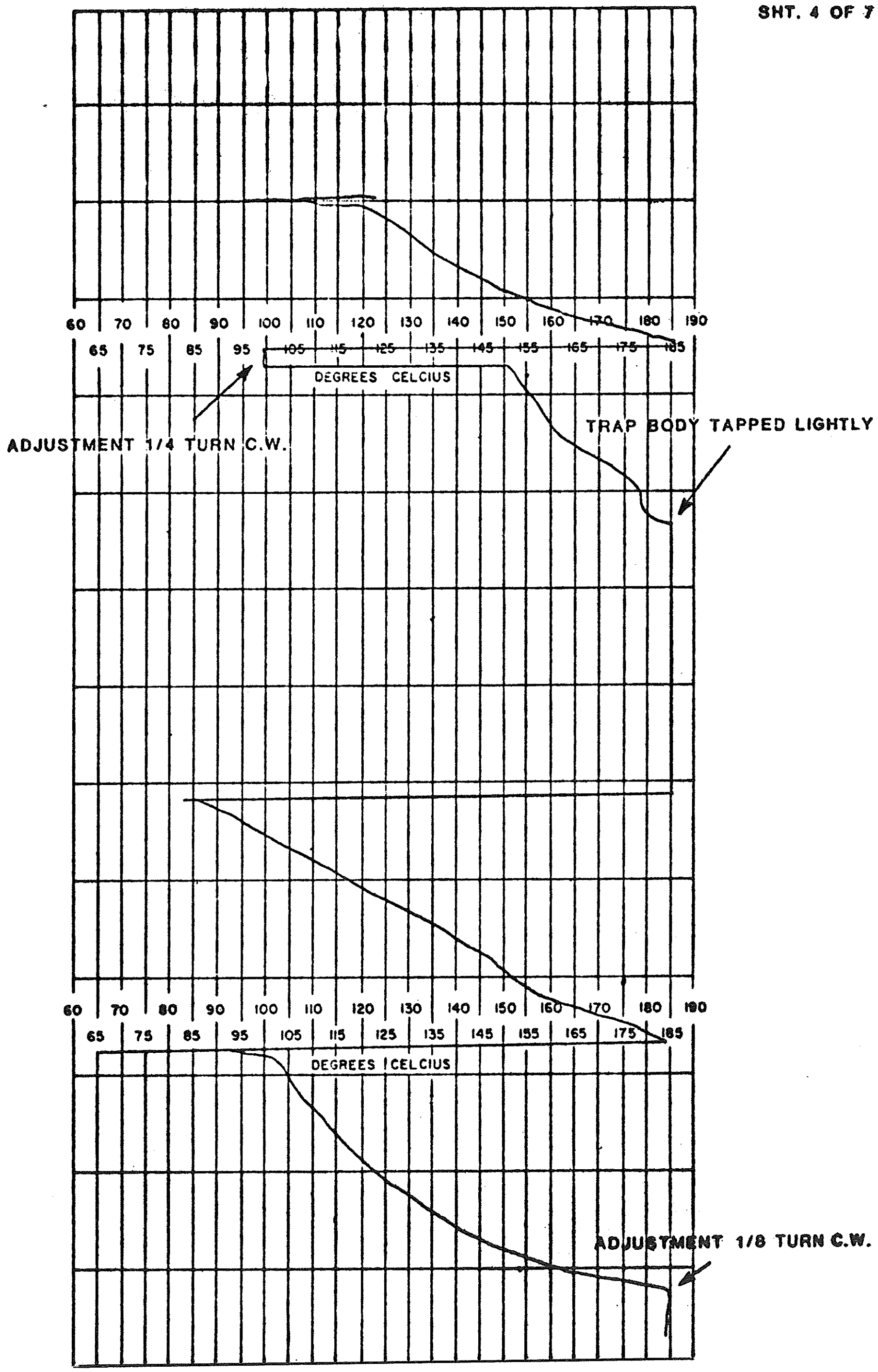
SHT. 2 OF 7



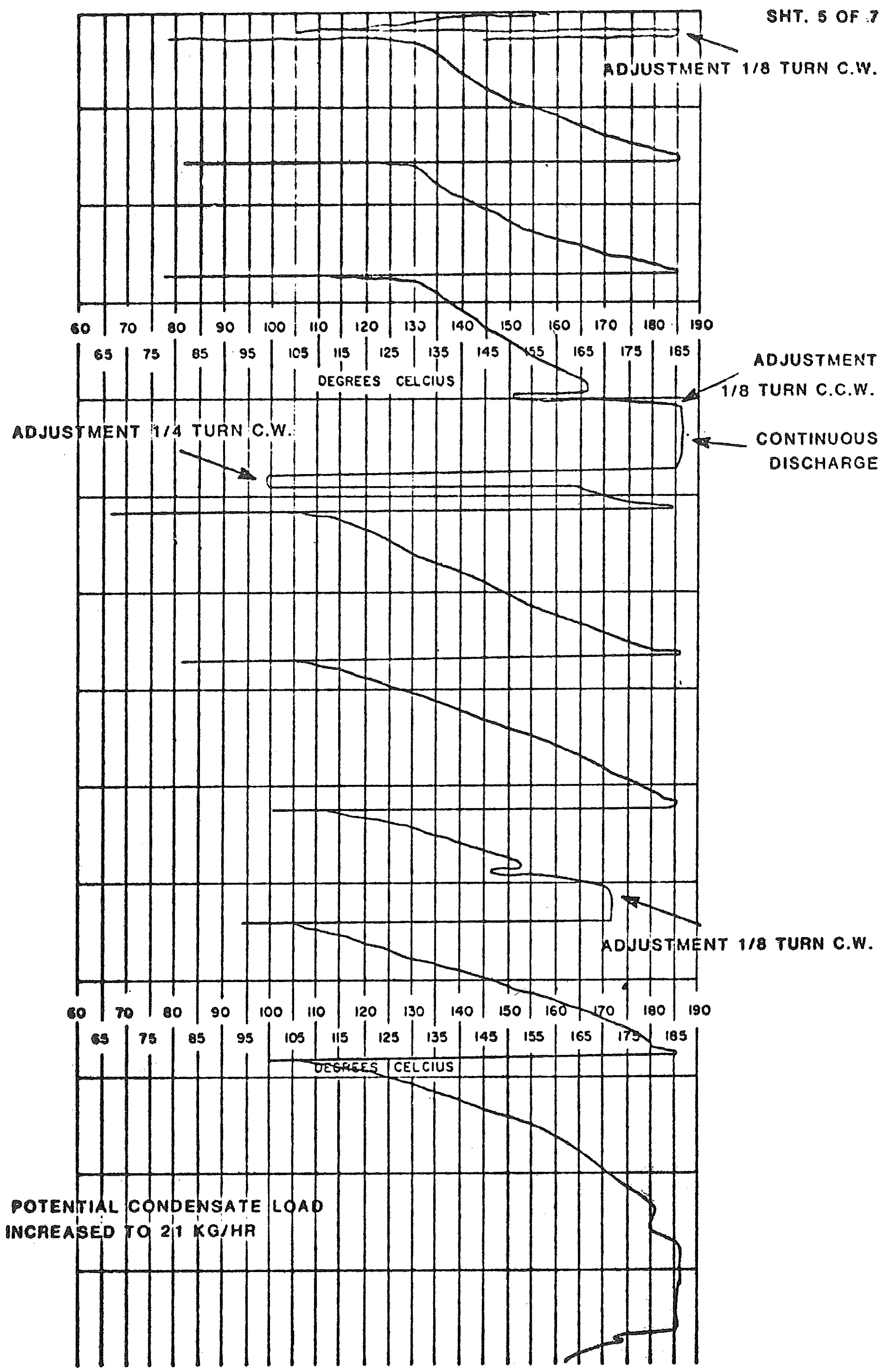
GRAPH T-2


60
70
80
90
100
110
120
130
140
150
160
170
180
190
65
75
85
95
105
115
125
135
145
155
165
175
185
DEGREES CELCIUS
TRAP BODY TAPPED LIGHTLY
CONTINUOUS DISCHARGE

GRAPH T-3

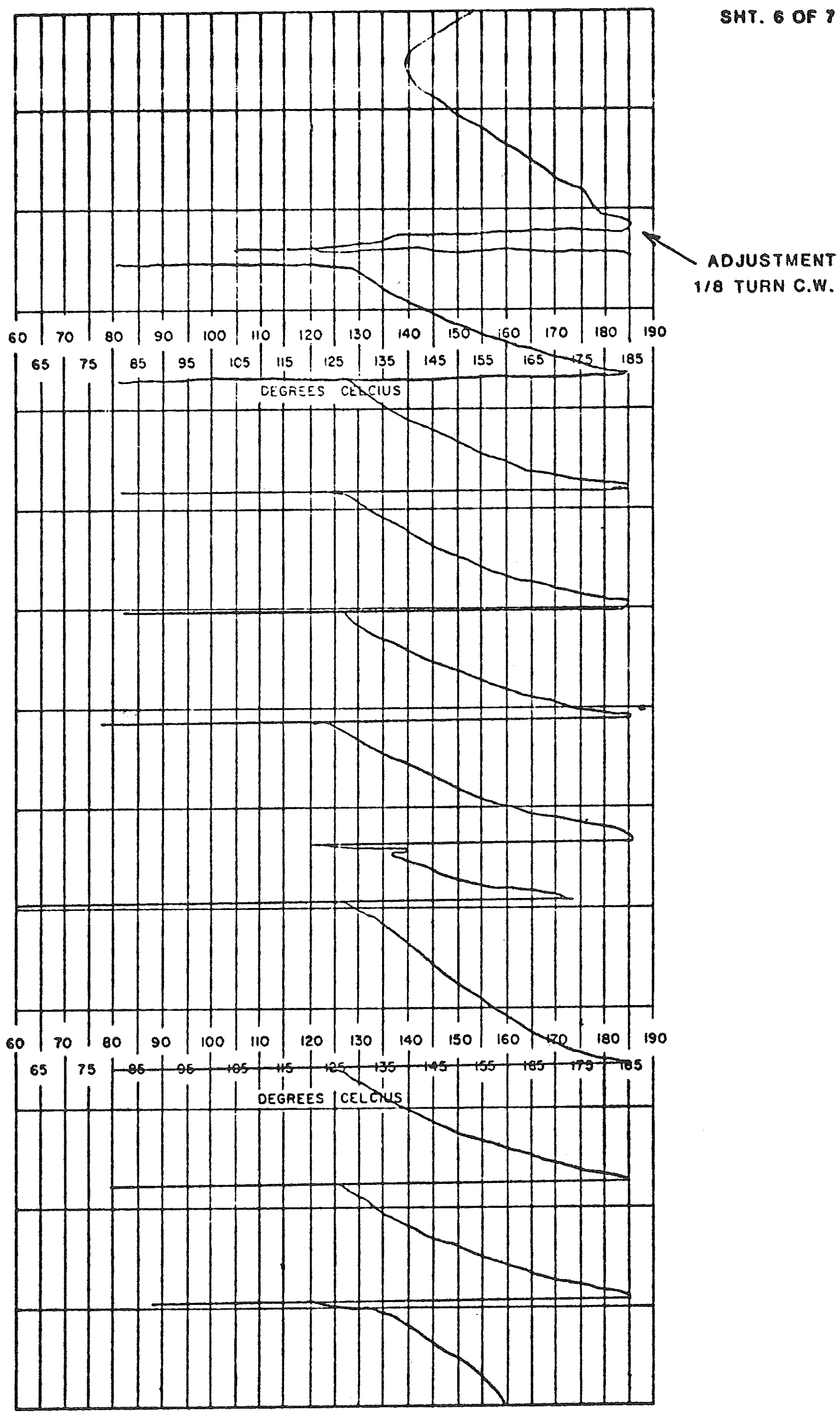

60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ADJUSTMENT 1/4 TURN C.W.
TRAP BODY TAPPED LIGHTLY
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ADJUSTMENT 1/8 TURN C.W.

GRAPH T-4

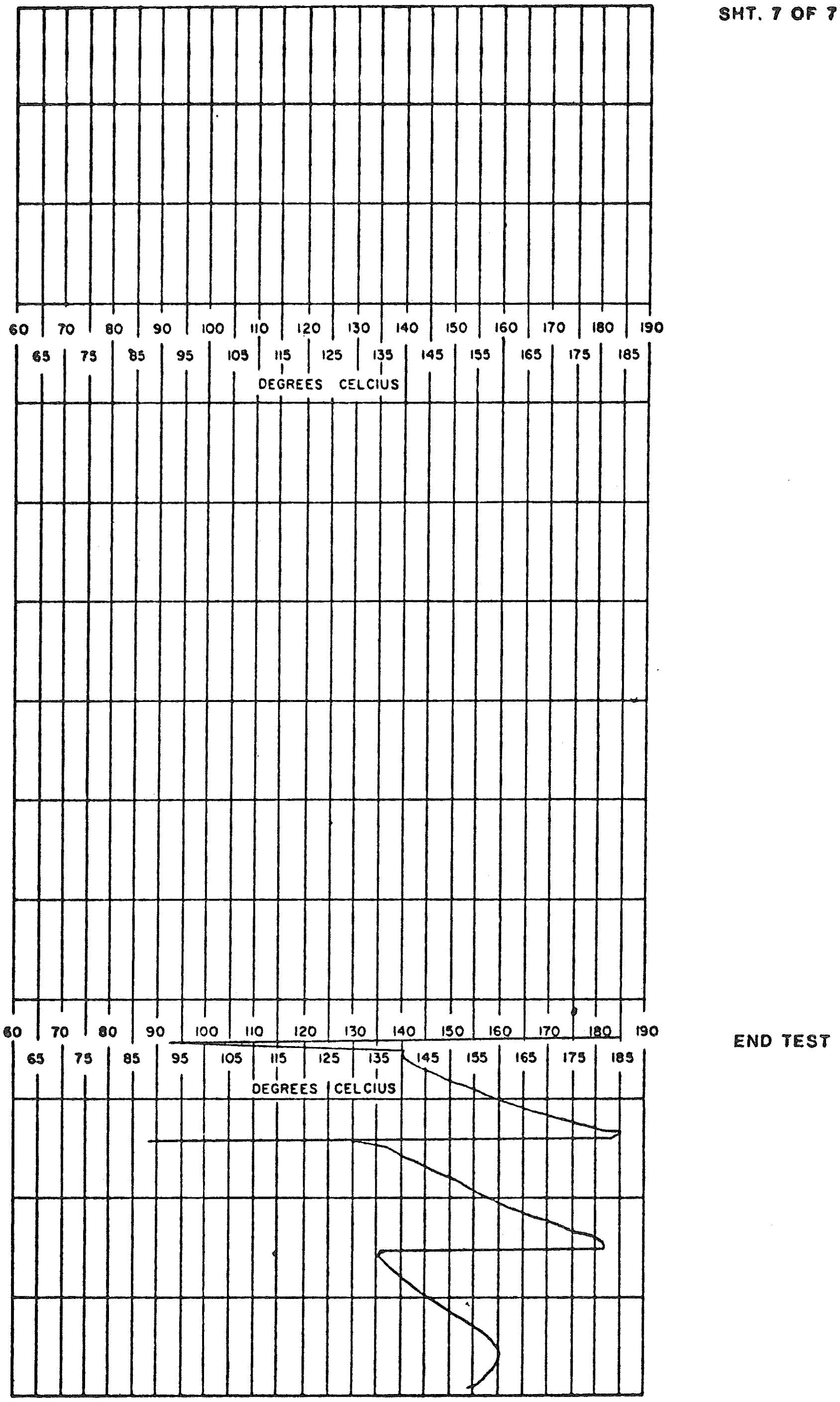

SHT. 5 OF 7
ADJUSTMENT 1/8 TURN C.W.
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
ADJUSTMENT 1/8 TURN C.C.W.
ADJUSTMENT 1/4 TURN C.W.
CONTINUOUS DISCHARGE
ADJUSTMENT 1/8 TURN C.W.
60 70 80 90 100 110 120 130 140 150 160 170 180 190
65 75 85 95 105 115 125 135 145 155 165 175 185
DEGREES CELCIUS
POTENTIAL CONDENSATE LOAD
INCREASED TO 21 KG/HR

GRAPH T-5

SHT. 6 OF 7



GRAPH T-6


60
70
80
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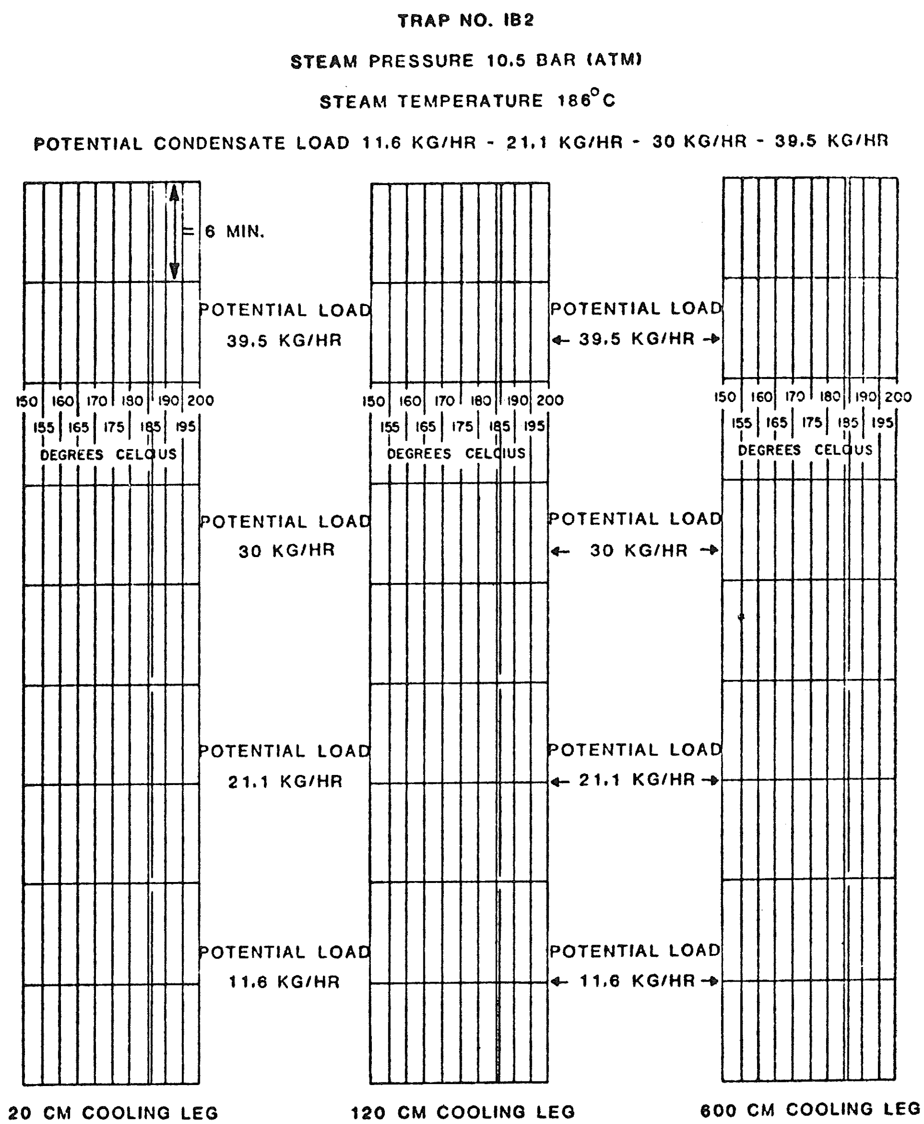
100
110
120
130
140
150
160
170
180
190
65
75
85
95
105
115
125
135
145
155
165
175
185
DEGREES CELCIUS
END TEST

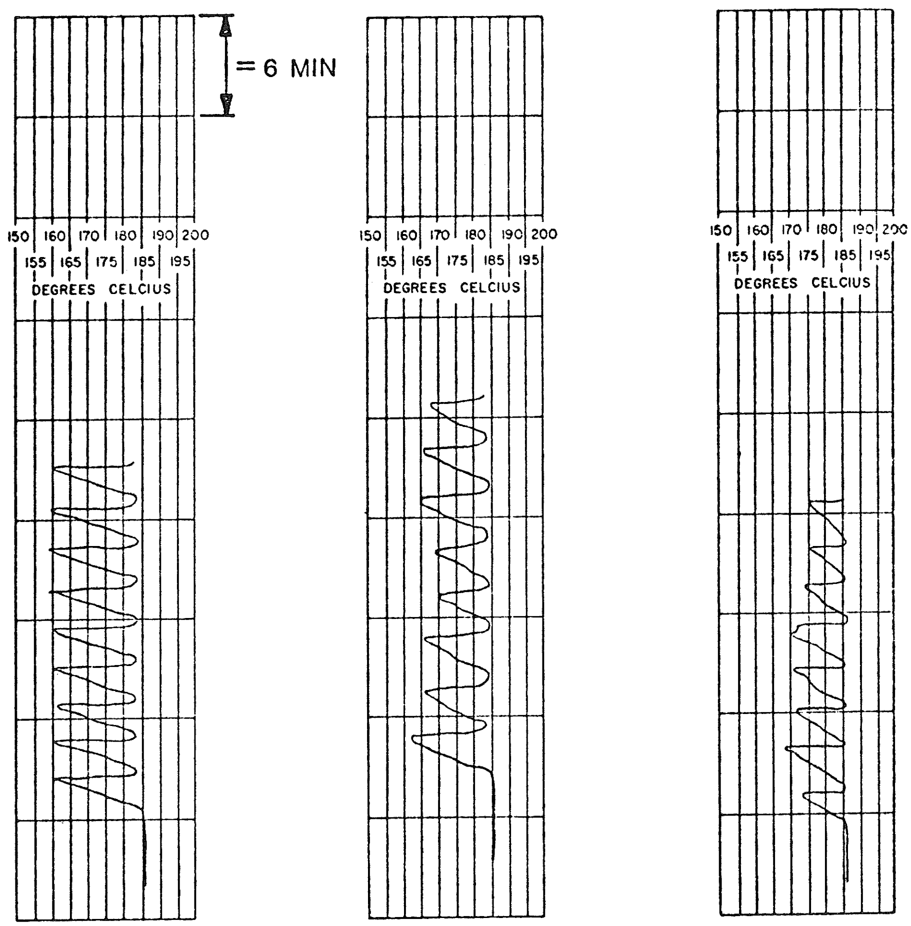
GRAPH T-7


TRAP NO. IB2
STEAM PRESSURE 10.5 BAR (ATM)
STEAM TEMPERATURE 186°C
POTENTIAL CONDENSATE LOAD 11.6 KG/HR - 21.1 KG/HR - 30 KG/HR - 39.5 KG/HR
= 6 MIN.
POTENTIAL LOAD 39.5 KG/HR
POTENTIAL LOAD ← 39.5 KG/HR →
150 160 170 180 190 200
155 165 175 185 195
DEGREES CELCIUS
POTENTIAL LOAD 30 KG/HR
POTENTIAL LOAD ← 30 KG/HR →
POTENTIAL LOAD 21.1 KG/HR
POTENTIAL LOAD ← 21.1 KG/HR →
POTENTIAL LOAD 11.6 KG/HR
POTENTIAL LOAD ← 11.6 KG/HR →
20 CM COOLING LEG
120 CM COOLING LEG
600 CM COOLING LEG

GRAPH U

Please note that the inverted bucket trap remained at steam temperature and discharged all the condensate independently of the condensate load and the length of pipe between the heat exchanger and the trap itself. Bimetallic trap 3C, however, reacted much differently to these conditions. (See Graphs V-1, V-2, V-3, and V-4.) At the lowest capacity of 11.6 kg/hr and with the shortest cooling leg, the unit opened at approximately 25.7°C below the saturated temperature and closed 3-4° below saturation. The unit was fairly constant, however. Temperature change, both with cooling legs of 120 cm and 600 cm, became gradually smaller and the closing point began to approach saturated steam temperature, especially with the 600 cm cooling leg. When the load was increased to 21.1 kg/hr, the same phenomenon occurred. Although the sub-cooling with the shortest cooling leg increased, it remained nearly the same in the large cooling leg. In all three cases, the unit stayed at one fixed position after it had cycled several times. At a capacity of 30 kg/hr, the unit cycled with the two longer cooling legs only. With the 120 cm cooling leg, it started to cycle and then took a quasi-fixed position, opening and closing just slightly which brought the condensate temperature far below the actual saturated steam temperature.

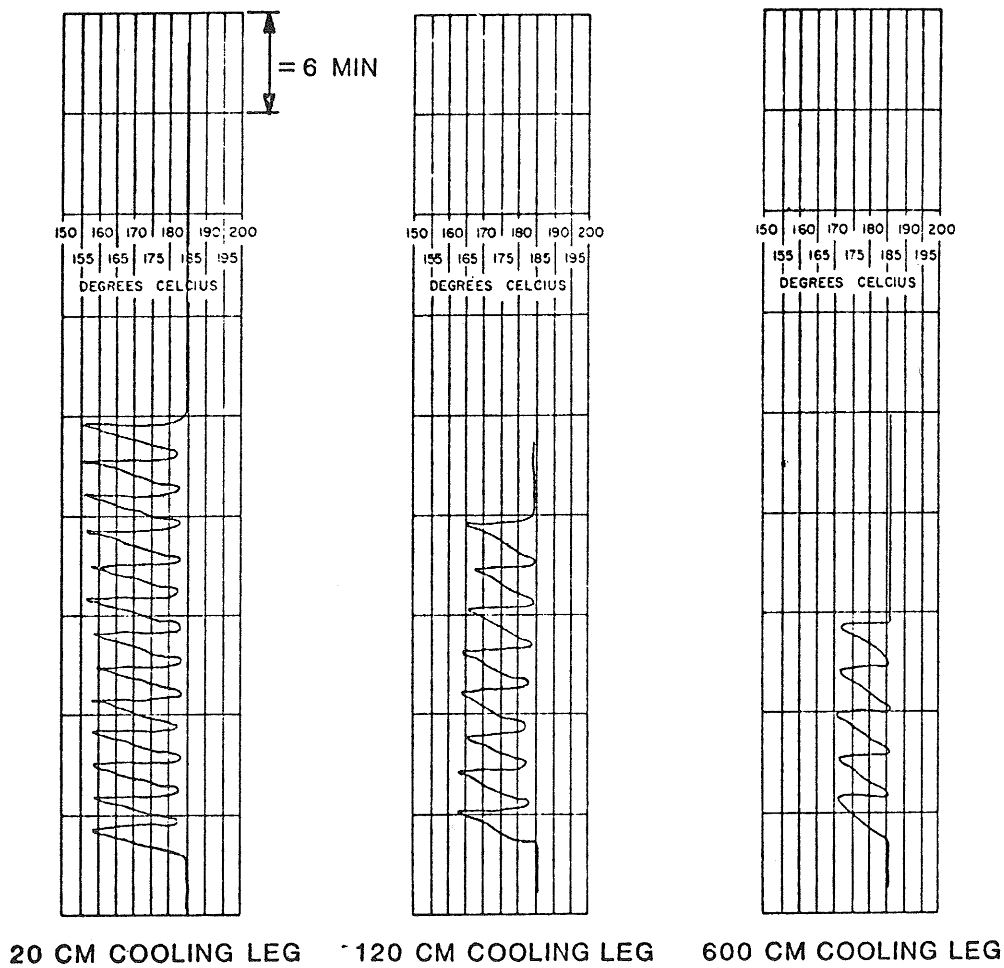
TRAP NO. 3C
STEAM PRESSURE 10.5 BAR (ATM)
STEAM TEMPERATURE 186 °C
POTENTIAL CONDENSATE
LOAD 11.6 KG/HR



20 CM COOLING LEG 120 CM COOLING LEG 600 CM COOLING LEG

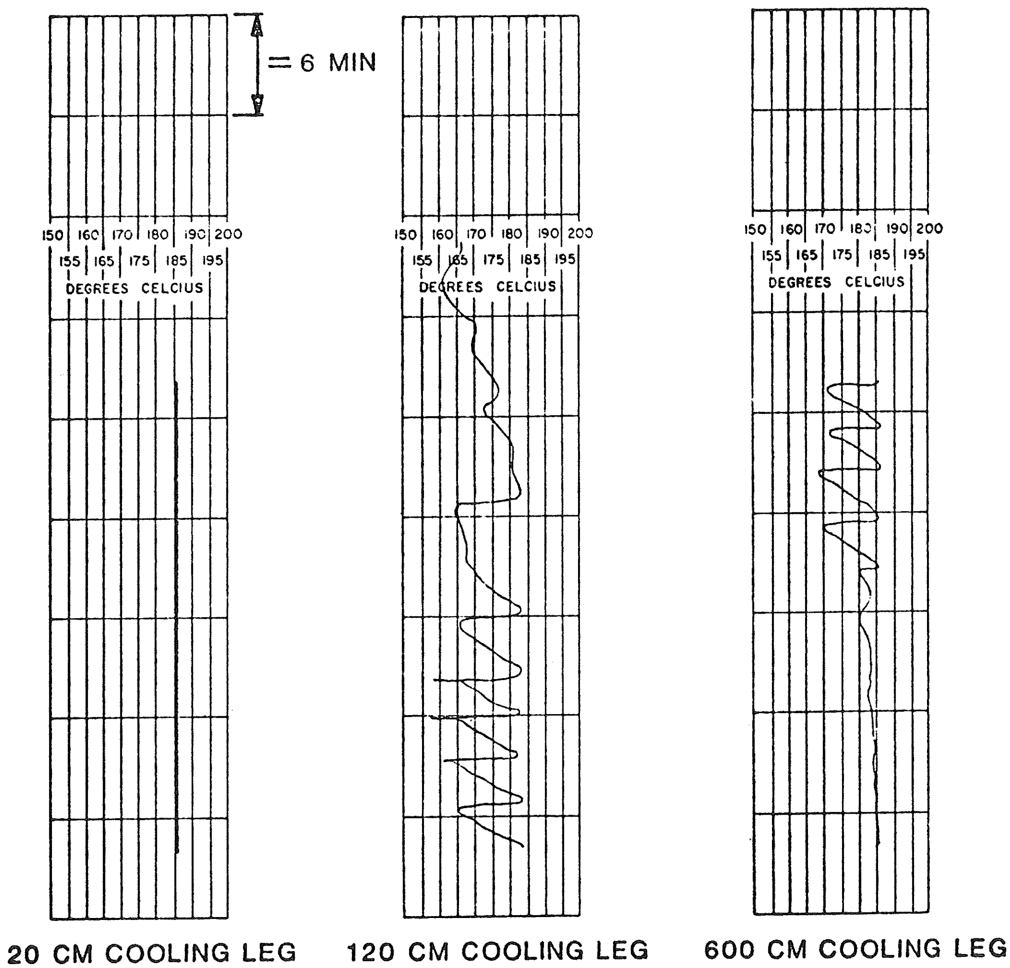
GRAPH V-1

TRAP NO. 3C
STEAM PRESSURE 10.5 BAR (ATM)
STEAM TEMPERATURE 186 °C
POTENTIAL CONDENSATE
LOAD 21.1 KG/HR



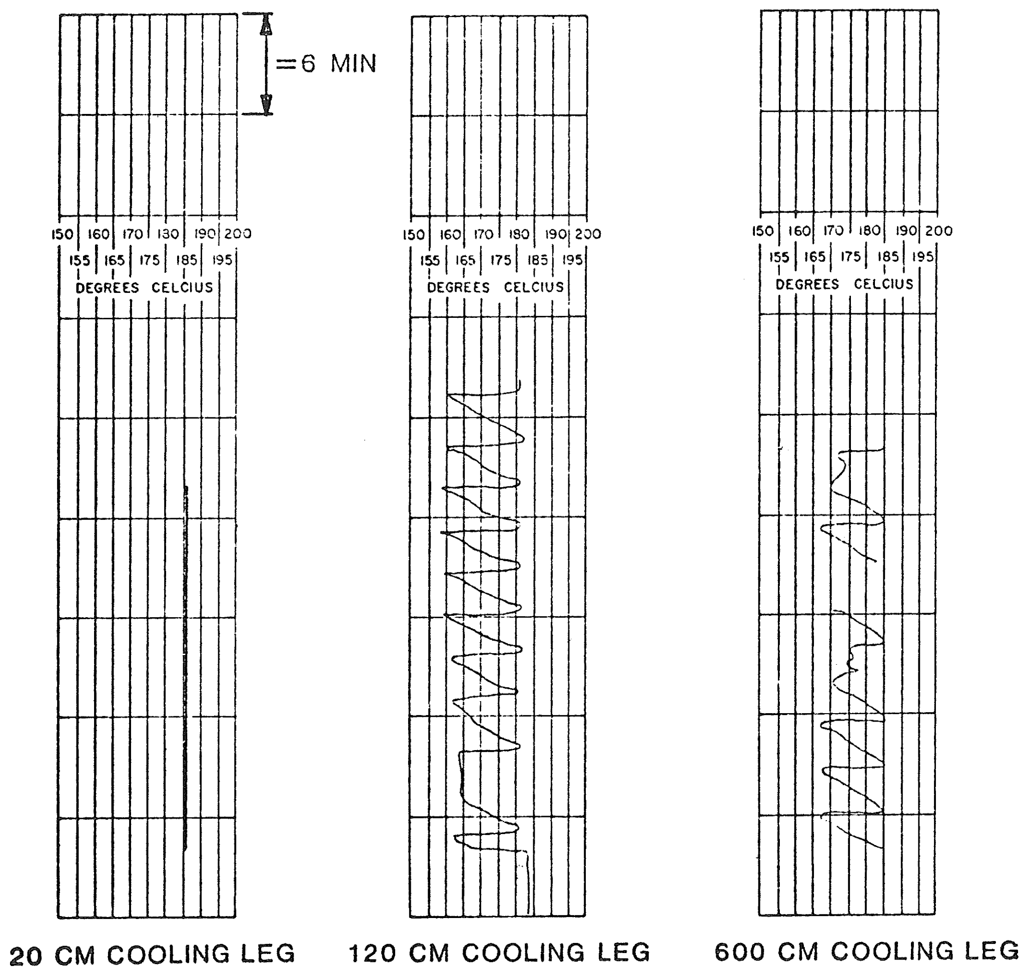
GRAPH V-2

TRAP NO. 3C
STEAM PRESSURE 10.5 BAR (ATM)
STEAM TEMPERATURE 186 °C
POTENTIAL CONDENSATE
LOAD 30 KG/HR



GRAPH V-3

TRAP NO. 3C
STEAM PRESSURE 10.5 BAR (ATM)
STEAM TEMPERATURE 186 °C
POTENTIAL CONDENSATE
LOAD 39.5 KG/HR



GRAPH V-4

The reverse was seen in the 600 cm cooling leg where the unit started in a fixed position and then opened and closed at a later point. When the load was increased to 39.5 kg/hr, the unit cycled only with the 120 cm cooling leg, opening at $\approx$ 37° below saturation and closing $\approx$ 5° below saturation temperature. The 600 cm leg was totally unpredictable, and the 20 cm cooling leg took a fixed position.

After establishing the general performance of the bimetallic traps and determining that sub-cooling was quite substantial in all units, we examined what sub-cooling actually does to or for the system. There is no advantage in having the condensate sub-cooled before it is returned to the boiler house, since the kilocalories left behind in the system must then be added to bring the boiler feed water up to temperature. Also, there is more danger of corrosion when the condensate is sub-cooled. In cases where the condensate is not returned, a system better than sub-cooling the condensate in a piece of heat exchanging equipment should be used. Whether this is a tracer line or another type of heat exchanger does not alter that conclusion. In many cases, steam is available at a certain pressure because it is taken from the secondary side of a turbine or brought in from an outside source. In both cases, the steam pressure might be quite high. It is often believed that, by sub-cooling the condensate, less steam can be used to perform the same work. This, of course, has some disadvantages. In the first place, the heat exchanging equipment must be much larger than is actually necessary. Some manufacturers even go so far as to calculate a system for high temperature water rather than steam. In cases where the condensate is not returned, a better system is to use the high pressure steam and then utilize the heat available in the condensate to make low pressure steam for feeding a second and, possibly, a third tracer line. This arrangement utilizes all the available heat in the most efficient way.

In looking at a system that discharges sub-cooled condensate rather than saturated condensate from a steam trap, we know that a certain amount of heat must be transferred from a tracer line for the steam using equipment, q_{REQ}. We also know the amount the boiler is taxed (H_{BOILER}) to satisfy this requirement, assuming that no condensate is returned. Therefore, the formula to determine H_{BOILER} is as follows.

$$H_{BOILER} = \left(\frac{q_{REQ}}{i''_{AVAIL}}\right) \cdot \left(i''_{STEAM}\right) - \left(\frac{q_{REQ}}{i''_{AVAIL}}\right) \cdot \left(i'_{MAKEUP}\right)$$

Where:

i''_{AVAIL} = heat available in kilograms of steam

i''_{STEAM} = specific enthalpy of saturated steam

i'_{MAKEUP} = specific enthalpy of the makeup water

Since:

$$\frac{q_{REQ}}{i''_{AVAIL}} = L$$

Where:

L = load in kg/hr

Thus:

$$H_{BOILER} = \left(\frac{q_{REQ}}{i''_{AVAIL}}\right) \cdot \left(i''_{STEAM}\right) - \left(L \cdot i'_{MAKEUP}\right)$$

We must assume that there are no transmission losses and that saturated steam is generated with no condensate return. If we have a system in which condensate is sub-cooled to gain the sensible heat from the condensate, we can determine the steam pressure at which a trap that does not sub-cool the condensate can be used and yet taxes the boiler the same amount. We have System 1 discharging saturated condensate and System 2 discharging sub-cooled condensate.

$$q^1_{REQ} = q^2_{REQ}$$

And:

$$H^1_{BOILER} = H^2_{BOILER}$$

Therefore:

$$\left(\frac{q^1_{REQ}}{i''^1_{AVAIL}}\right) \cdot \left(i''^1_{STEAM}\right) - \left(L^1 \cdot i'^1_{MAKEUP}\right) = \left(\frac{q^2_{REQ}}{i''^2_{AVAIL}}\right) \cdot \left(i''^2_{STEAM}\right) - \left(L^2 \cdot i'^2_{MAKEUP}\right)$$

Further, let us assume that the heat content of the makeup water is small and the same for both systems.

$$\left(\frac{q^1_{REQ}}{i''^1_{AVAIL}}\right) \cdot \left(i''^1_{STEAM}\right) = \left(\frac{q^2_{REQ}}{i''^2_{AVAIL}}\right) \cdot \left(i''^2_{STEAM}\right)$$

Since:

$$q^1_{REQ} = q^2_{REQ}$$

Therefore:

$$\frac{i''^{1}_{STEAM}}{i''^{1}_{AVAIL}} = \frac{i''^{2}_{STEAM}}{i''^{2}_{AVAIL}}$$

In System 1 with saturated condensate discharge:

$$i''^{1}_{AVAIL} = r^{1}$$

Where:

r = latent heat

In System 2, with sub-cooled condensate discharge:

$$i''^{2}_{AVAIL} = r^{2} + i'^{2} - i'^{2}_{SUB}$$

$$= i''^{2}_{STEAM} - i'^{2}_{SUB}$$

Where:

i' = specific enthalpy of saturated condensate

i'_{SUB} = specific enthalpy of sub-cooled condensate

So:

$$\frac{i''^{1}_{STEAM}}{r^{1}} = \frac{i''^{2}_{STEAM}}{i''^{2}_{STEAM} - i'^{2}_{SUB}}$$

And we arrive at the following final equation:

$$i'^{2}_{SUB} = i''^{2}_{STEAM} \cdot \left(1 - \frac{r^{1}}{i''^{1}_{STEAM}}\right)$$

With this equation we can set a steam pressure for System 1 and System 2 to determine the amount of sub-cooling required in System 2 to tax the boiler the same number of kilocalories.

$$H^1_{BOILER} = H^2_{BOILER}$$

Example:

System 2 at 12 bar, 188°C

$$i''^2_{STEAM} = 664.6 \text{ kcal/kg}$$

System 1 at 6 bar, 158.5°C

$$r^1 = 498 \text{ kcal/kg}$$

$$i'^1 = 658.1 \text{ kcal/kg}$$

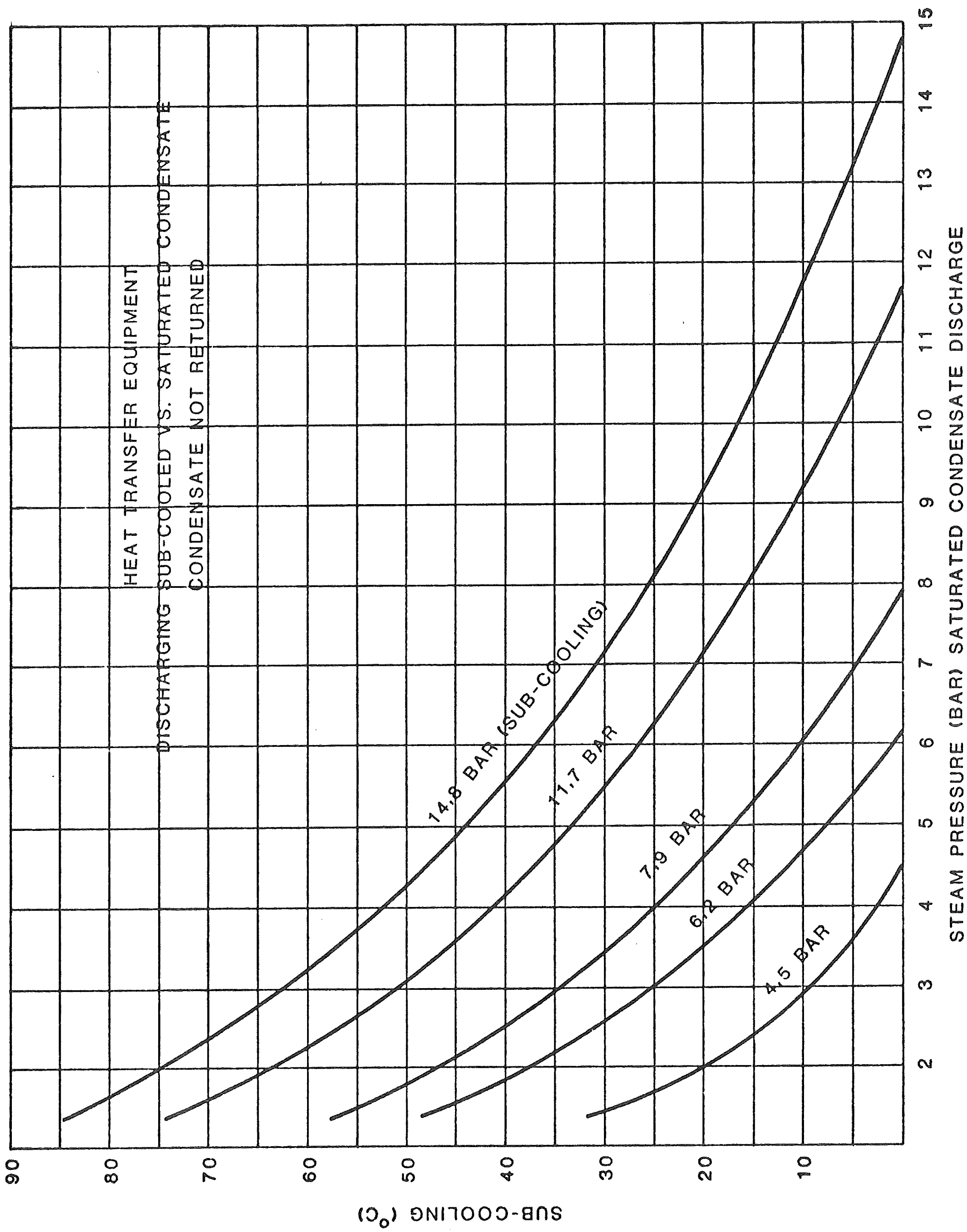
Therefore:

$$i'^2_{SUB} = i''^2_{STEAM} \cdot \left(1 - \frac{r^1}{i'^1}\right)$$

Or:

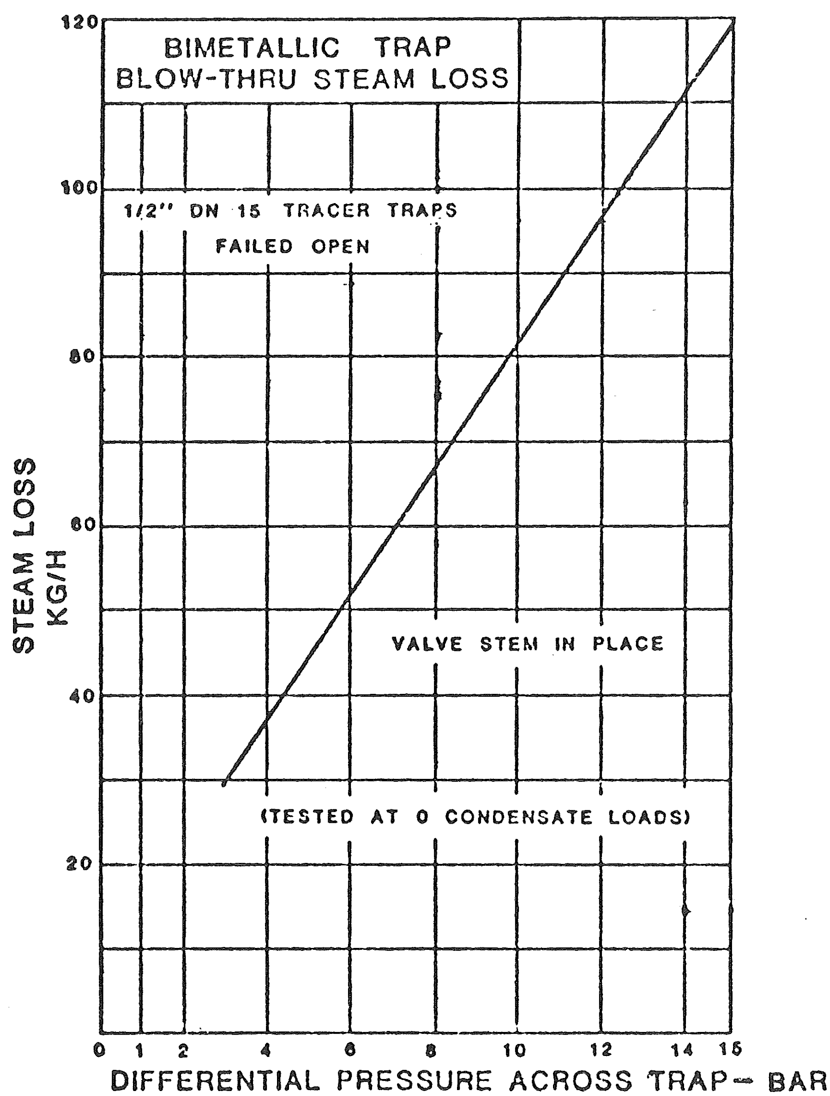
$$664.6 \cdot \left(1 - \frac{498}{658.1}\right) = 161.7 \text{ kcal/kg}$$

This corresponds to a temperature of 160.3°C or 27.7°C sub-cooling (188°C - 160.3°C). Looking at it from another point of view, the boiler would be taxed the same amount if the system was at 6 bar and discharged saturated condensate. The above example is plotted on Graph W, along with a series of these calculations.

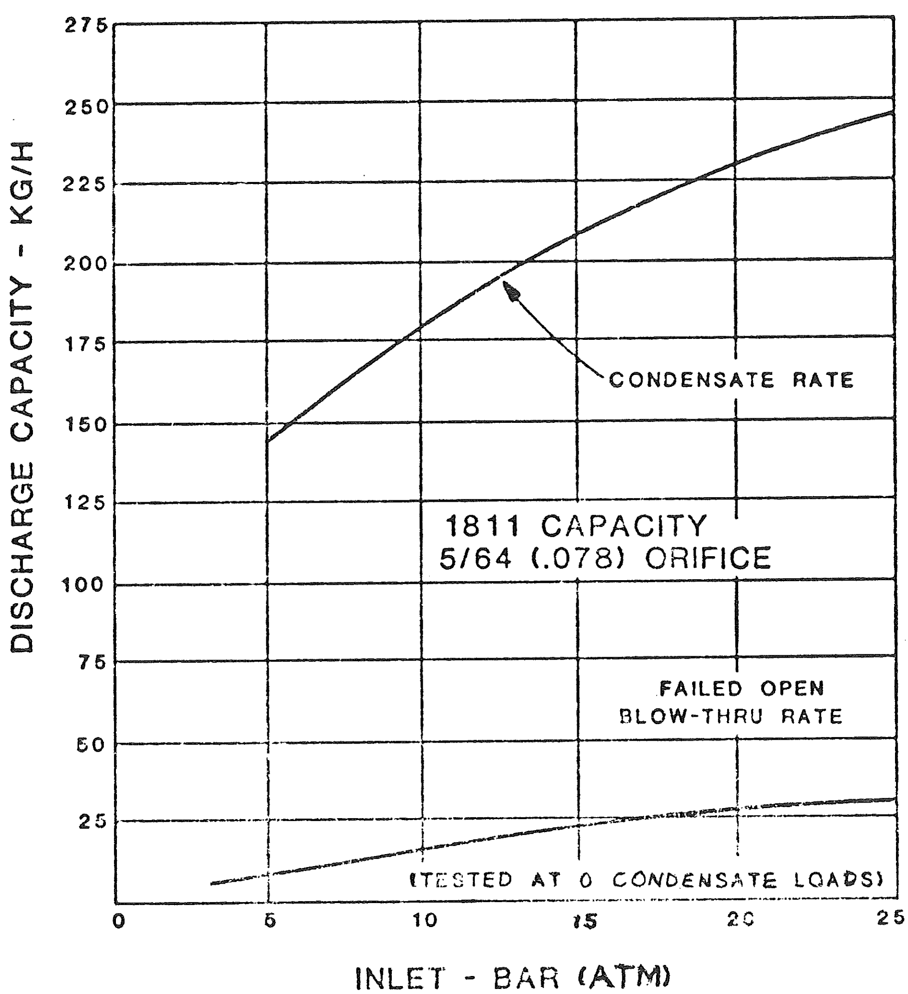

HEAT TRANSFER EQUIPMENT
DISCHARGING SUB-COOLED VS. SATURATED CONDENSATE
CONDENSATE NOT RETURNED
14,8 BAR (SUB-COOLING)
11,7 BAR
7,9 BAR
6,2 BAR
4,5 BAR
STEAM PRESSURE (BAR) SATURATED CONDENSATE DISCHARGE
2 3 4 5 6 7 8 9 10 11 12 13 14 15
SUB-COOLING (°C)
90 80 70 60 50 40 30 20 10

GRAPH W

This shows that traps with sub-cooling will tax the boiler the same as traps with no sub-cooling at a corresponding lower pressure. Another important point which we would like to bring out is when a bimetallic trap fails to open, the steam loss can be extremely large. The graph below shows a composite of blow-through with full valve opening of various types of bimetallic traps. Please note that all the traps which were used were 1/2" tracer line traps. The valve stem was kept in place, but we have seen several failures where the valves actually fell out. We did not take this into consideration in this graph, but in case the valve would fall out of place, the total steam loss of a blow-through trap would be considerably higher.



If we could compare this with the 1811 trap with a 5/64" orifice, you will see that our trap with its small orifice, which by the way, handles more condensate than most of the drip and tracer lines deliver, has a much lower blow-through rate than the bimetallic trap when it fails to open.



As a last point, we would like to look at a freezing problem which can exist with steam traps. Although the bimetallic trap can be installed in different positions, the condensing steam will create a vacuum in the piping which could close the valve, thereby stopping the flow of condensate and trapping condensate between the control valve and bimetallic valve. With an inverted bucket trap, the unit's valve would not close off and in case of a vacuum, air in an open system and/or condensate in a close return system, would break such a vacuum and the condensate would flow back towards the trap.

Although the bimetallic trap might freeze and break the bolting at the cap, the stainless steel inverted bucket trap would give a high resistance to the damage from freezing.

In summary, the following conclusions could be drawn from the testing:

1. The best heat exchange can be obtained by using a trap which discharges condensate close to steam temperature. Although it may be appealing from a simple theoretical approach, in reality substituting the sensible heat of hot water for the latent heat of steam results, in most cases, in an energy waste. The drop of heat transfer rate of steam using equipment will result in a reduced and uneven system.

2. As we have seen on the charts, a bimetallic thermostatic trap is unpredictable and nonrepeatable, and to field adjust a bimetallic trap is almost impossible. An inverted bucket trap, due to its method of operation, does not require adjustment.

3. In cases where the bimetallic trap backs up condensate or discharges back-up condensate which is below steam temperature, thermal and hydraulic shocks could damage the system while it increases at the same time the pipe corrosion problem.

4. Due to the large orifice present in a bimetallic trap, in comparison to an inverted bucket trap, a horrendous amount of energy is wasted in a very short time when they fail in the open position.

5. While an inverted bucket trap remains open during shutdown, a bimetallic trap might close due to the fact that its valve works as a natural check valve. Since this vacuum can not be broken, a possible freezing problem can occur because the condensate is not allowed to drain itself.