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Engineering

AN EXPERIMENTAL SINGLE-TUBE EVAPORATOR

by

G. S. Nichols and E. S. Occhipinti
Separations Technology Division

January 1956

E. I. du Pont de Nemours & Co.
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ABSTRACT

An experimental single-tube evaporator was evaluated for low-holdup evaporation of uranyl nitrate solutions. Control of evaporation was adequate at high percentage evaporation, but entrainment was excessive.

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TABLE OF CONTENTS

LIST OF FIGURES AND TABLES

AN EXPERIMENTAL SINGLE-TUBE EVAPORATOR

INTRODUCTION

An evaporator consisting of a single run of steam-jacketed pipe was selected for study on the basis that such an evaporator would contain a minimum amount of hot liquid and that it would have greater mechanical strength than any other type of evaporator. In the pipe type of evaporator, the solution is fed into one end of the pipe and a mixture of liquid and vapor is discharged from the other end to a separator.

The present study was undertaken on a laboratory scale to determine how well such a pipe evaporator can be controlled and how much of the solute is entrained in the effluent vapor.

SUMMARY

A laboratory-scale pipe evaporator was used to concentrate dilute solutions of uranyl nitrate. The over-all heat transfer coefficient was approximately 760 Btu/hr-ft^2-$^\circ$F.

At 98 per cent evaporation the uranium entrainment was 7 per cent of the total uranium fed to the evaporator. This entrainment is excessive, and will require a satisfactory solution of separator details before application to plant problems will be successful.

A unit capable of handling 1000 pounds of solution per hour would require approximately eighty feet of 1-1/2 inch stainless steel pipe.

DISCUSSION

DESCRIPTION

Equipment

The experimental system that was installed at the TNX Laboratory consisted of a feed tank, a pipe evaporator, a separator, and a condenser. The design of the evaporator was based on an analysis of mixed phase flow through a duct.(1) The main evaporator unit was a coil of 14.5 feet of 3/8-inch Type 304 stainless steel tubing. The coil was enclosed in a steam jacket made of a 5-foot length of 4-inch steel pipe (Figure 6). The auxiliary equipment included an 8 x 36-inch stainless steel feed tank, a 4 x 8-inch stainless steel separator, and a condenser with 60 feet of 3/8-inch stainless tubing in a 4-inch x 5-foot shell. (Figures 6 through 9).

Experimental Procedure

The operating procedure consisted of pressurizing the feed tank to 25 psig with air and feeding the liquid to the pipe evaporator through a restricting orifice. This procedure partially divorced the feed rate from fluctuations in evaporator pressure. Steam pressure on the jacket was controlled either manually or by instruments to maintain a constant temperature of the mixture of liquid and vapor product. The liquid was removed from the mixture by a cyclone separator, and was collected in the concentrate receiver. The vapor was condensed and the condensate was collected in a 55-gallon drum. The flowsheet is shown on Figure 9.

Instrumentation

Automatic control was accomplished by the pressure - temperature cascade control shown in Figure 5. This system consisted of a temperature controller, a temperature transmitter with an iron-constantan thermocouple as the measuring element for the temperature of the exit stream, a pressure recorder, a pressure transmitter, and an air-to-open motor valve in the steam supply line.

OPERATION WITH MANUAL CONTROL

When the fraction of feed that was evaporated was less than 90 per cent, the pipe evaporator was controlled manually, i.e. with a fixed steam pressure, without difficulty. Initial runs were performed using process water as feed, and the degree of vaporization was varied by changing the steam pressure in the jacket. The results of the initial runs are presented in Table I and the composite data from each run are shown in Figure 1. These early results are in agreement with the later results except where the fraction evaporated approached 1.0.

OPERATION WITH INSTRUMENT CONTROL

Control

The boiling point elevation that occurred while uranyl nitrate solutions were being concentrated was sufficient to provide a control function. This control function was transmitted to a variable set point of the steam pressure controller; the output from the steam pressure controller operated the steam supply valve. Temperature deviations of ± 1°C at 104°C and steam pressure deviations of ± 0.5 psi were observed.

The long connection between the evaporator discharge and the separator, plus the small scale of operation, resulted in large heat losses. The product concentration was therefore always considerably less than the concentration that would be indicated by the boiling point of the solution. Holdup in the piping probably contributed to a poor material balance for uranyl nitrate.

Correlation of Data

The results of the evaporator tests while operating on instruments are tabulated in Table II. Each run lasted about thirty minutes and used about a half-tank of feed. The steam pressure and the feed pressure were averaged over each run (Columns 2 and 3). The product flows corresponding to these runs are tabulated in Table III.

The mean value for U_o, the over-all heat transfer coefficient for the evaporator, was 761 Btu/hr-ft²-°F. The calculated values of U_o for each run are listed in Table II, and cover a comparatively narrow range between 707 and 838 Btu/hr-ft²-°F.

The flows and pressures in the pipe evaporator are correlated by Equation 1, derivation for which is given in the Appendix.

$$(P_sP_o - 1/2\ P_o^2)(1/W^n) = Kx^n + C \tag{1}$$

Where W = feed rate, lb/hr

x = fraction of feed evaporated

P_s = steam pressure, psig

P_o = feed pressure, psig

K = proportionality constant

Equation 1 is plotted in Figure 1, using a value of $n = 3$. The resulting correlation is linear, with $K = 0.0147$ and $C = 0.0009$. Evaporators for higher capacities were estimated with the aid of this relationship (Figure 4).

Entrainment

The exit streams showed an undesirably high entrainment of uranium in the overhead vapor. Table III contains analyses of the exit streams and losses of the uranium to the condensate. The uranium loss to the condensate is shown in Figure 2 as a function of the percentage of evaporation. In Figure 3, this uranium loss is shown as a straight line function of the log of the fraction not evaporated (i.e. fraction of concentrate). Extrapolation of the curve indicates that for required vaporizations (99.2 per cent) an excessive entrainment of 8 to 10 per cent of the uranium can be expected with the present equipment.

The excessive entrainment problem could be improved by the methods outlined in the following table, but the magnitude of the improvements is not known except for series evaporation. If two evaporators were operated in series with concentrate recycled from the second cycle to the first, losses would be reduced to 0.1 per cent or DF = 1000 at 85 per cent vaporization. The other proposals require experimental development.

METHODS OF REDUCING LOSSES

Description	Design Data Required
Series operation	Present information satisfactory
Improved entrainment separators	No successful solution known except to use fine-mesh filters where the pressure drops will be limiting
Partial condensation	If condensation occurs on existing drops this method would be helpful, but condensation probably occurs at the condenser wall.

CALCULATIONS*

1. Evaluation of

$$\frac{P_sP_o - 1/2\, P_o^2}{W^3} \text{ and } x^3$$

Sample data are taken from Run 31, Tables II and III.

$P_s = 29$ psig

$P_o = 16$ psig

Feed Rate = 274 cc/min

Concentrate Rate = 76.3 cc/min

$$W = \frac{(274)(60)}{454} = 36.3 \text{ lb/hr}$$

$$\frac{P_sP_o - 1/2\, P_o^2}{W^3} = \frac{(29)(16) - 1/2\,(16)(16)}{(36.3)^3} = 7.05$$

$$x = \frac{274-76.3}{274} = 0.722$$

$$x^3 = 0.376$$

2.. Losses - also calculated for Run 31, Table III

% U Loss in overhead condensate

$$= \frac{(\text{condensate concentration})(\text{fraction evaporated})(100)}{(\text{feed concentration})}$$

$$= \frac{(0.068)(0.722)(100)}{2.11} = 2.3\%$$

---

* Nomenclature is listed in the Appendix.

3. Calculation of U_o

Data from Run 45, Table II

P_s = 32 psig = 46.7 psia

P_o = 16.5 psig = 31.2 psia

W = 34.7 lb/hr

x = 0.83 lb evap./lb feed

Assumptions

(1) Vapor produced is saturated steam (no boiling point rise).

(2) Feed enters at 70°F.

(3) For purposes of calculating Δt_m, the feed inlet temperature is assumed to be the saturation temperature corresponding to the feed pressure, P_o.

From the capacity equation,

$$q = U_o A \Delta t_m$$

Where

q = heat transferred, Btu/hr

U_o = over-all heat transfer coefficient, Btu/hr-ft^2-°F

A = heat transfer area, ft^2

Δt_m = log mean temperature difference, °F

For 83 per cent vaporization,

$$q = W(1-x)(H_1 - H_2) + W(x)(H_v - H_2)$$

Where

W = liquid throughput, 34.7 lb/hr

x = fraction of liquid evaporated, 0.83

H_1 = enthalpy of saturated liquid (water) at 212°F, 180 Btu/lb (The correction of boiling point elevation by the salts has been neglected.)

H_2 = enthalpy of inlet feed at 70°F, 38 Btu/lb

H_v = enthalpy of exit steam vapor at atmospheric pressure, 1150 Btu/lb

$$q = 34.7(0.17)(180-38) + 34.7(0.83)(1150-38)$$
$$= (5.9)(142) + 28.8\ (1112) = 32{,}900 \text{ Btu/hr}$$

t_s = temperature of saturated steam at P_s of 46.7 psia = 267.7°F

t_i = temperature of saturated liquid at P_o of 31.2 psia = 252.5°F

t_f = temperature of exit mixture of liquid and vapor at 14.7 psia = 212°F

$$\Delta t_m = \frac{(267.7 - 212) - (267.7 - 252.5)}{\ln \frac{267.7 - 212}{267.7 - 252.5}} = 41.2^\circ F$$

$$A = \pi DL$$

Where

D = 0.0231 ft for 3/8-inch tube

L = 14.5 ft

$$A = (3.14)(0.0231)(14.5) = 1.05 \text{ ft}^2$$

$$U_o = \frac{q}{A\Delta t_m} = \frac{32{,}900}{(1.05)(41.2)}$$

$$U_o = 760 \text{ Btu/hr-ft}^2\text{-}^\circ F$$

4. Design Calculations

Equation 6 in the Appendix was derived by treating the pipe diameter as a variable. Since only one pipe diameter was investigated, the pipe diameter was treated as a constant in the calculations and included with the constant coefficient. Design calculations will have less uncertainty if the mass velocity used is near those of the experimental conditions.

A pipe evaporator designed to evaporate 2000 lb/hr at 85 per cent evaporation requires a feed supply pressure of 20 psig to obtain sufficient agitation. The mass velocity of experimental conditions is used.

$$\text{Total feed rate, } W = \frac{2000}{0.85} = 2350 \text{ lb/hr}$$

From Table II the feed rate, W, for 85 per cent evaporation is about 35 lb/hr. At equivalent mass velocity the diameter of the proposed evaporator is:

$$\left(\frac{D}{D_o}\right)^2 = \frac{W}{W_o}$$

$$\frac{D}{D_o} = \left(\frac{2350}{35}\right)^{1/2} = 8.20$$

D_o = 0.0231 ft (3/8-inch tube)

D = (0.0231)(8.20) = 0.189 ft or 2.28 inches

The steam pressure is computed from Figure 1.

$x = 0.85$

$x^3 = 0.62$

$$\frac{P_s P_o - 1/2\ P_o^2}{W^3} = 0.0099 \text{ from Figure 1}$$

$$(P_s)(20) - 1/2\ (20)(20) = (0.0099)(35)^3$$

$$P_s = \frac{425 + 200}{20} = 31.3 \text{ psig}$$

The experimental flow rate was used to compute the steam pressure because Figure 1 was computed from the experimental condition by assuming that the pipe diameter was constant and equal to 3/8-inch diameter tube.

The length of pipe is computed from the heat transfer requirement.

$$q = W(1-x)(H_1 - H_2) + W(x)(H_V - H_2)$$

Where

W = liquid throughput, 2350 lb/hr

x = fraction of liquid evaporated, 0.85

H_1 = enthalpy of saturated liquid (water at 212°F, 180 Btu/lb

H_2 = enthalpy of inlet feed at 70°F, 38 Btu/lb

H_V = enthalpy of exit steam vapor at atmospheric pressure, 1150 Btu/lb

$$q = (2350)(0.15)(180-38) + (2350)(0.85)(1150-38)$$

$$= (2350)(967) = 2.27 \times 10^6 \text{ Btu/hr}$$

t_s = temperature of saturated steam at P_s = 31.3 psig = 274.4°F

t_i = temperature of saturated liquid at P_o = 20 psig = 258.8°F

t_f = temperature of exit mixture at atmospheric pressure = 212°F

$$\Delta t_m = \frac{(274.4 - 212) - (274.4 - 258.8)}{\ln \frac{(274.4 - 212)}{(274.4 - 258.8)}} = 33.7^\circ F$$

From the capacity equation,

$$q = U_o A \Delta t_m$$

g = heat transferred = 2.27×10^6 Btu/hr

A = pipe area, ft^2

Δt_m = log mean temperature difference, ^oF

U_o = over-all heat transfer coefficient, 760 Btu/hr-ft^2-^oF

$$A = \frac{2.27 \times 10^6}{(33.7)(760)} = 88.7 \text{ ft}^2$$

$$A = \pi DL$$

$$L = \frac{(88.7)}{(\pi)(0.189)} = 149 \text{ ft}$$

The limits of reliability of this design were not established. Extension of Equation 1 to flow in the viscous region would be questionable since all the data were taken with the Reynold's numbers well into the turbulent region. Equation 1 would also be inaccurate at high steam temperatures where a different mode of boiling is expected to occur. The Reynold's number for this design is 4.47×10^5 and is calculated as follows:

$$Re = \frac{4W}{\pi \mu D}$$

$$Re = \frac{(4)(2000)}{(\pi)(0.0302)(0.189)} = 4.47 \times 10^5$$

Where

μ = viscosity of water vapor

= 0.0302 lb/(ft)(hr) at 212^oF

Reynold's numbers above 2000 are considered to be in the turbulent region.

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APPENDIX

BASIS FOR $(P_sP_o - 1/2\ P_o^2)(1/W^3) = Kx^3$ (1a)

Pipe evaporator mathematics are evolved from simultaneous solution of the mass flow and the heat flow equations. The relationship between pressure drop and mass flow for two-phase flow is given by Equations 2a and 2b. (Reference 1)

$$\left(\frac{dP}{dL}\right) = \frac{4f\rho V^2\phi^2}{2Dg_c} \tag{2a}$$

or by simplifying

$$\left(\frac{dP}{dL}\right) = \frac{k_1W^2x^2\phi^2}{D^5} \tag{2b}$$

where

(dP/dL) = pressure drop per unit length of pipe for mixed phase flow, psi/ft

ρ = density of vapor phase, lb/ft^3

f = fanning friction factor for vapor phase

V = linear velocity of vapor phase, ft/sec

D = pipe diameter, ft

W = total flow rate, lb/hr

x = fraction evaporated

g_c = gravitation constant, ft/sec^2

ϕ^2 = correction factor for mixed phase flow which can be approximated by the function $\phi^2 = 1+e^{-4.22x + 1.76}$ (Taken from Reference 1)

k_1 = numerical constant containing g_c, ρ, π, f, etc.

The heat flow equation for conduction through the walls of the pipe evaporator is given by Equation 3.

$$W\lambda dx = \pi DU_o\Delta T_m\ (dL) \tag{3}$$

where

λ = Enthalpy difference between vapor and liquid

U_o = over-all heat transfer coefficient

ΔT_m = mean temperature driving force for heat transfer

Equations 2 and 3 are combined by the elimination of dL, and the analytical solution is possible when simplifying assumptions are made for U_o and ΔT_m.

The heat transfer coefficient, U_o, could be represented as a function of the Reynold's number, but the range of experimental data did not justify the additional constants and U_o was assumed to be constant.

The temperature driving force, ΔT_m, was approximated by Equation 4 which represents a linear relation between temperature and pressure over a narrow range of both. (An exact approach would be to leave $\Delta T = \Delta T_m$ and graphically integrate the expression $\int_0^{P_o} \Delta T dP$ where the inlet pressure is P_o and the outlet pressure is zero.

$$\Delta T_m = 1.53\ (P-P_s) \qquad (4)$$

where
P_s = steam pressure, psig
P = evaporator pressure at the point where x is the fraction evaporated, psig

The number -1.53 is carried through the derivation as k_2

Equation 5 is the final differential equation with numerical coefficients represented by k_n.

$$\int_0^{P_o} (P-P_s)\ dP = k_3 \frac{W^3}{D^6} \int_x^0 x^2(1 + k_4 e^{k_5 x})dx \qquad (5)$$

where
$k_3 = \frac{k_1 \lambda}{\pi k_2 U_o}$
$k_4 = e^{1.76}$
$k_5 = -4.22$

The numerical values assigned to k_4, k_5, and k_6 are theoretical and not necessarily the values that will be determined by experiment. Actually the consistency of the experimental data does not justify the use of three arbitrary constants and Equation 6 was found just as acceptable.

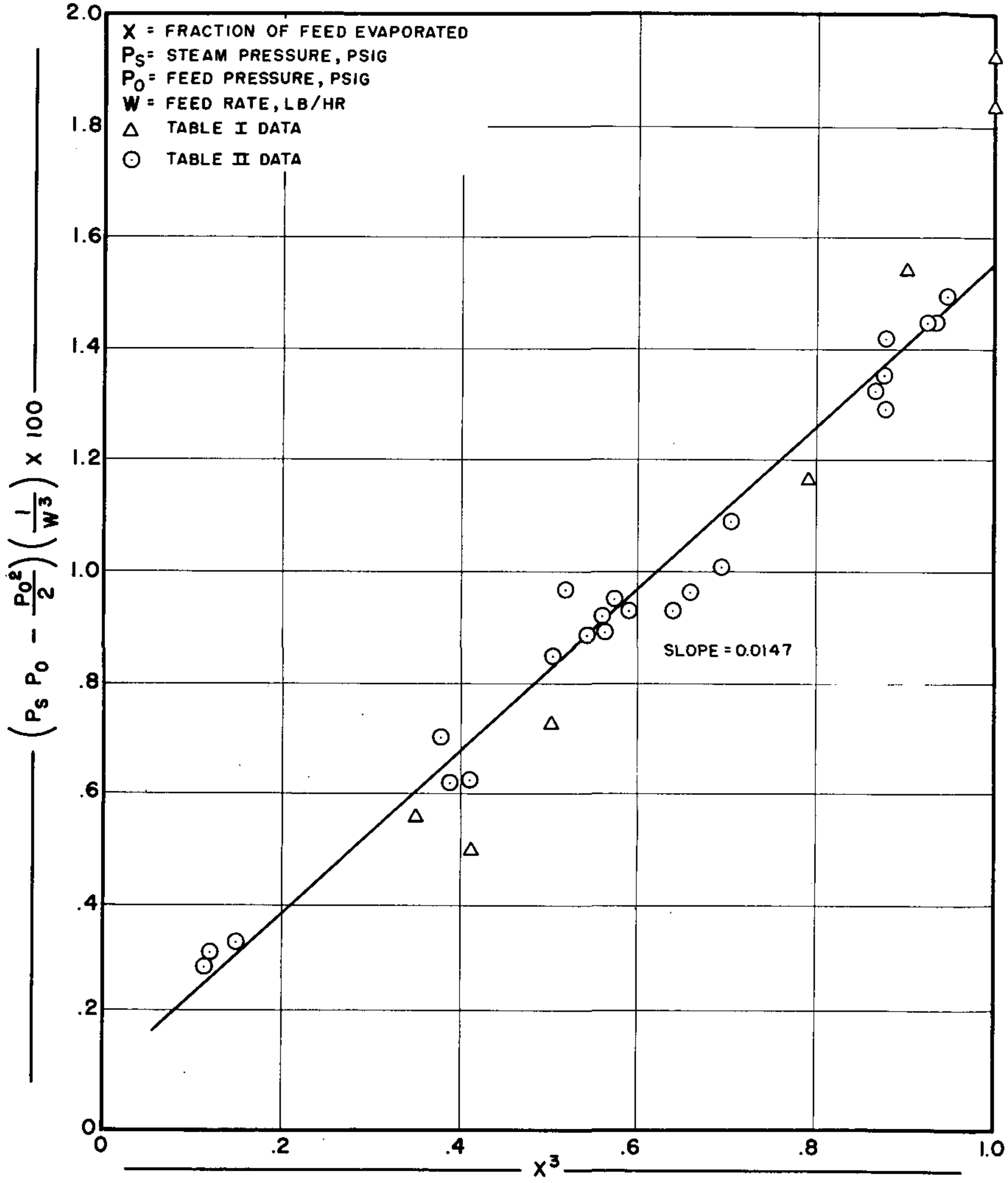
$$\int_{0}^{P_o} (P-P_s)dP = \frac{kW^3}{D^6} \int_{x}^{0} x^2 dx$$

Upon integration Equation 6 becomes Equation 1

$$(P_oP_s - 1/2\ P_o^2)(D^6/W^3) = kx^3 + C \qquad (1)$$

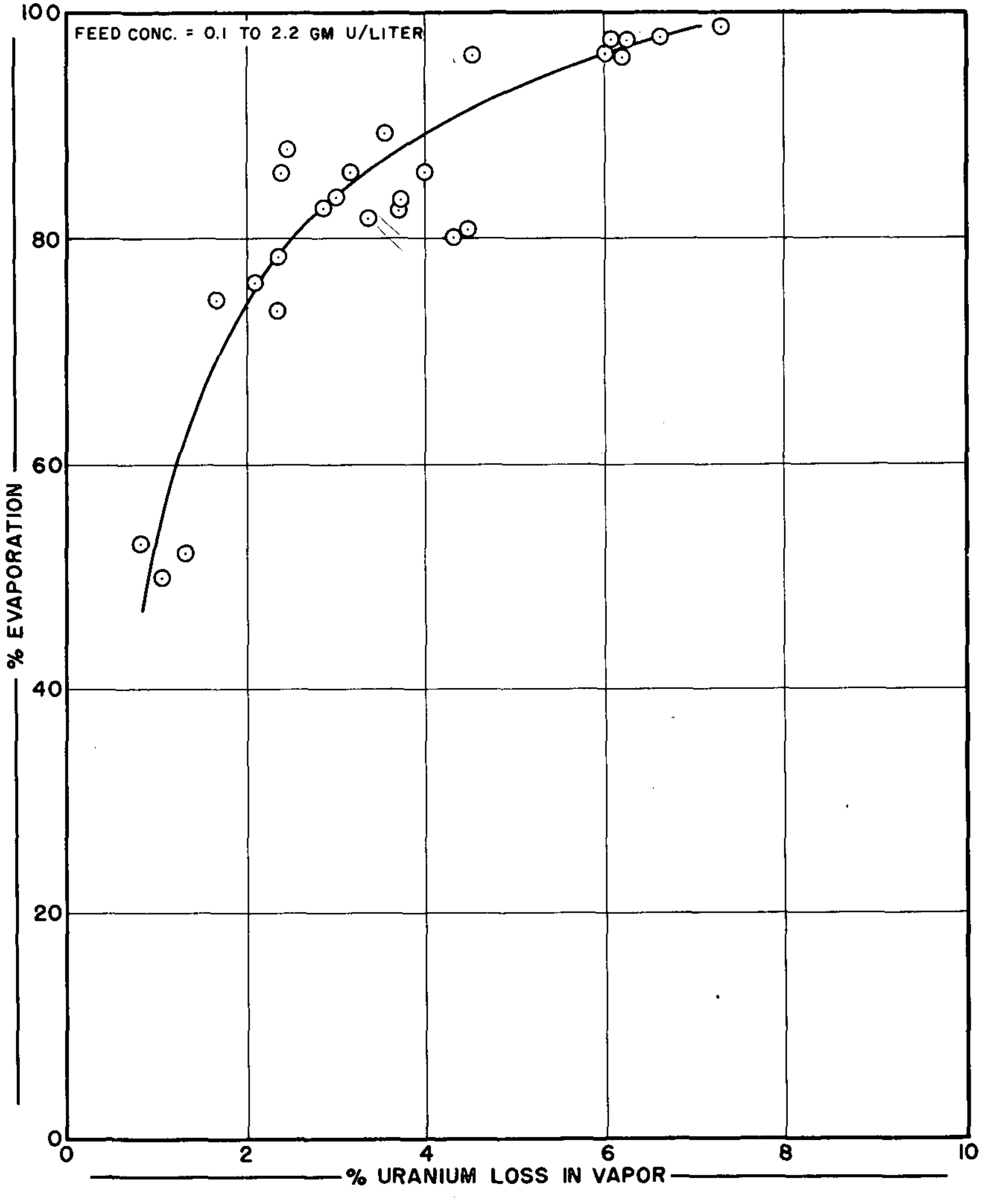
which represents a usable correlating equation for the experimental data.

FIGURE 1



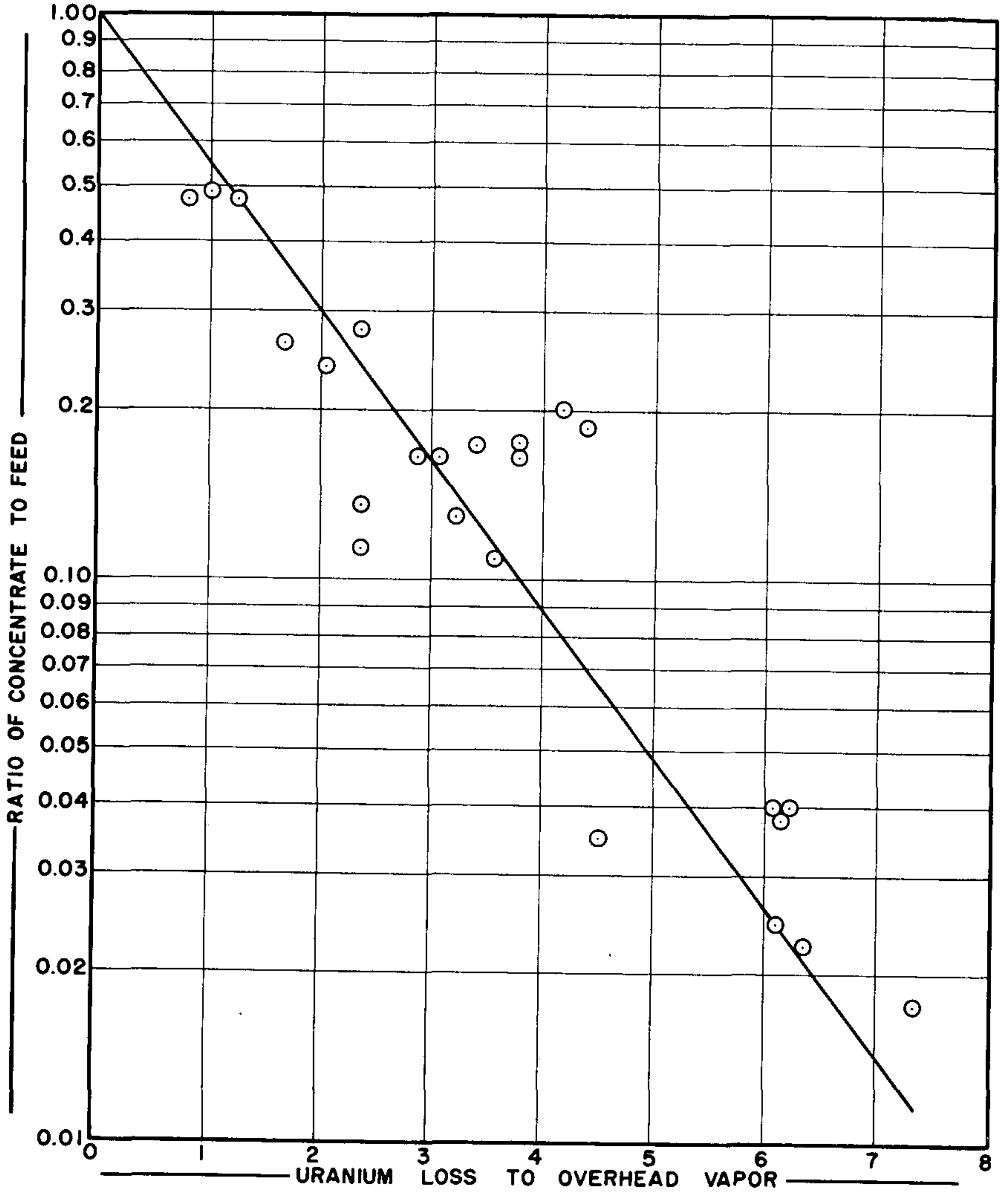
CORRELATION OF DATA ON LABORATORY–SCALE PIPE EVAPORATOR

FIGURE 2



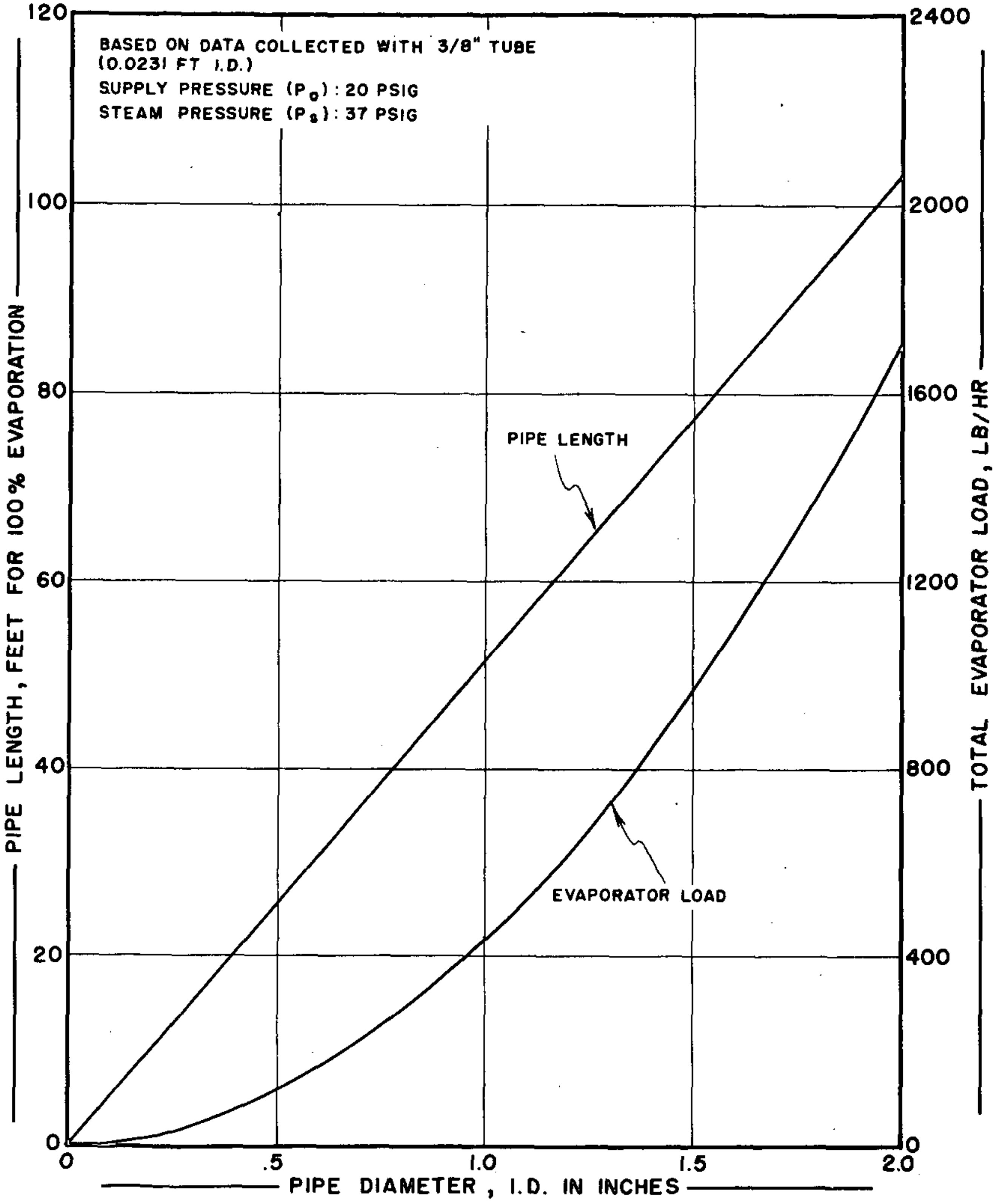
ENTRAINMENT LOSSES - ARITHMETIC PLOT

FIGURE 3



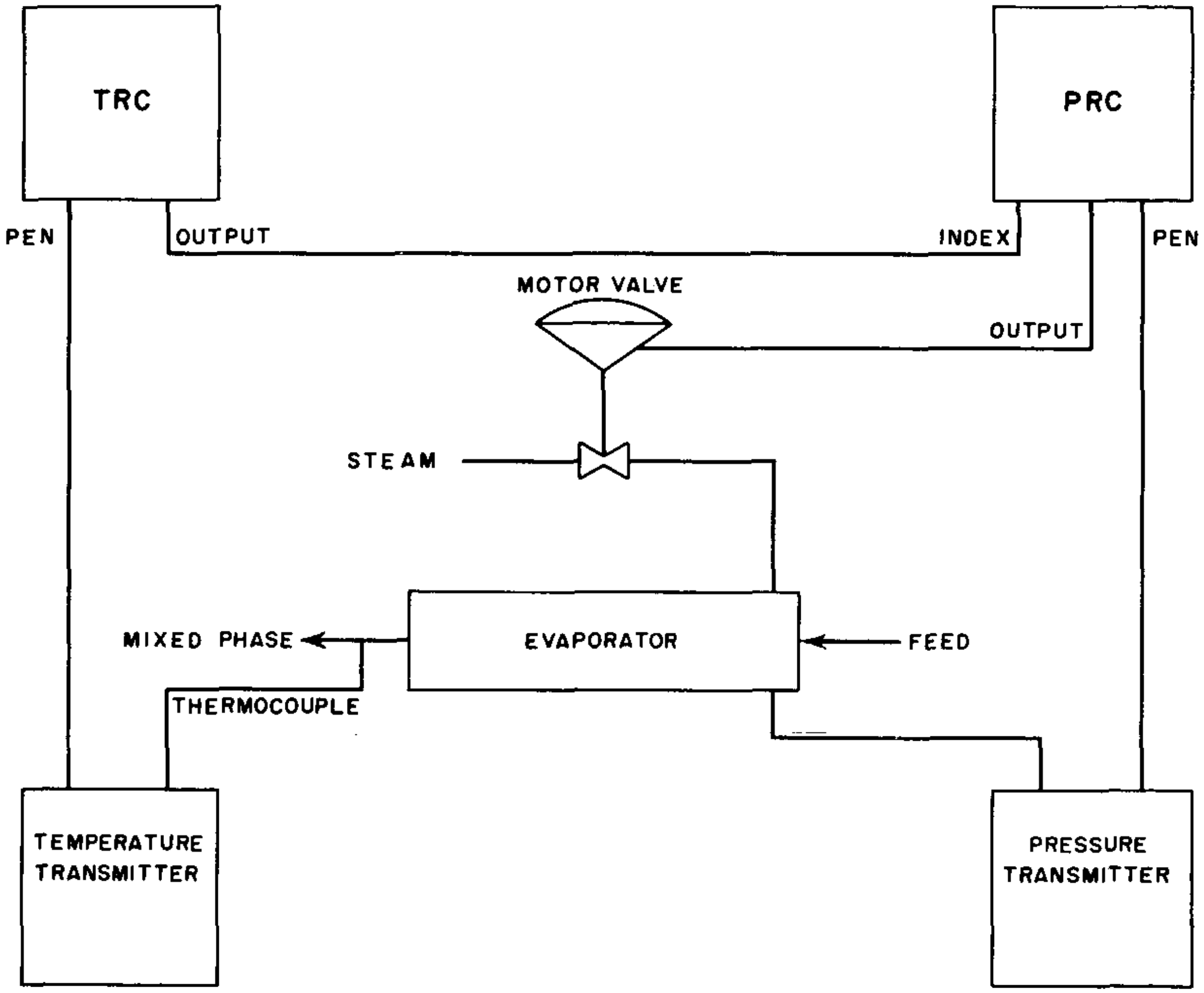
ENTRAINMENT LOSSES - SEMILOGARITHMIC PLOT

FIGURE 4



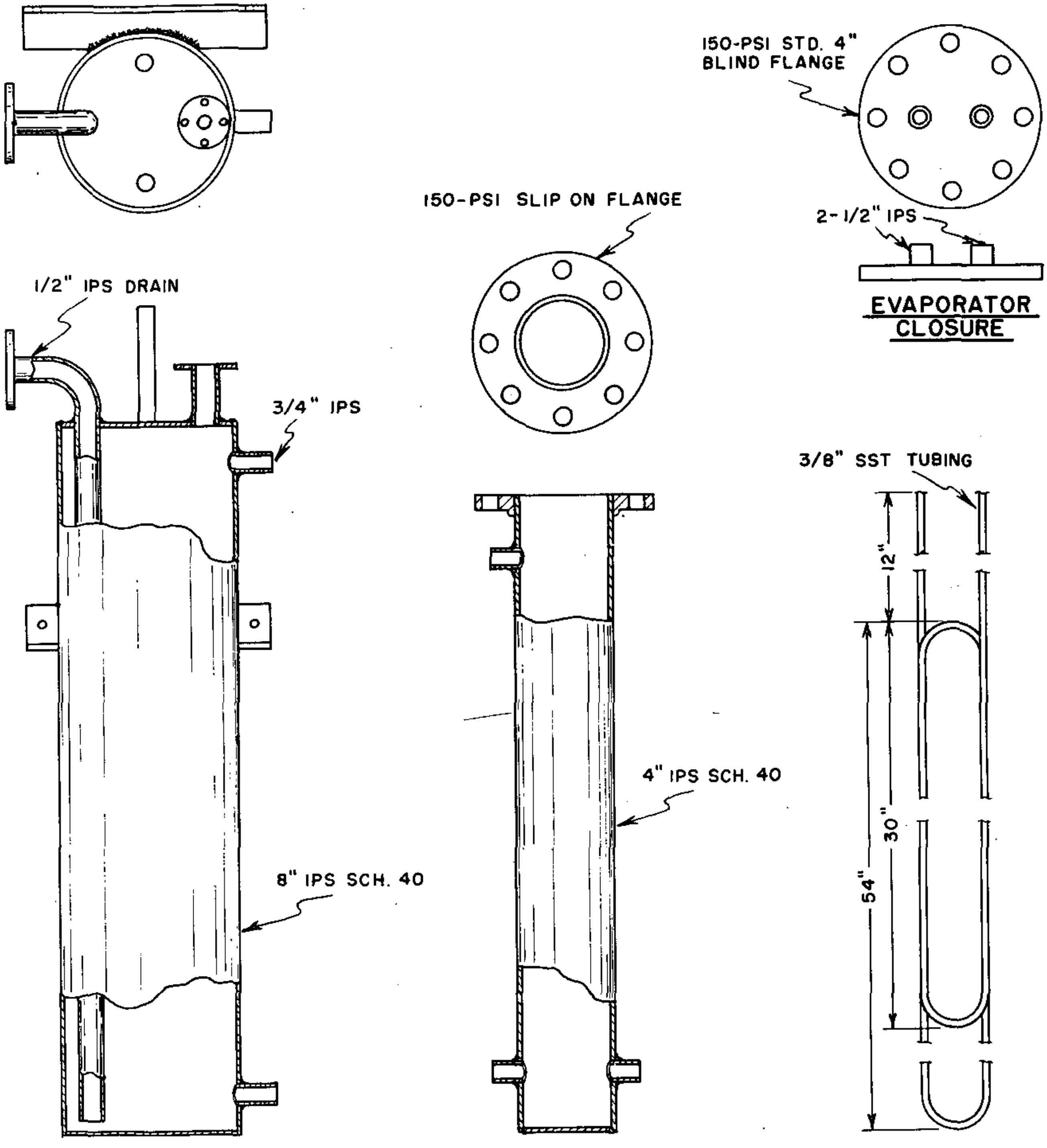
PIPE EVAPORATOR DESIGN CURVES

FIGURE 5



INSTRUMENTATION

FIGURE 6



EVAPORATOR CLOSURE

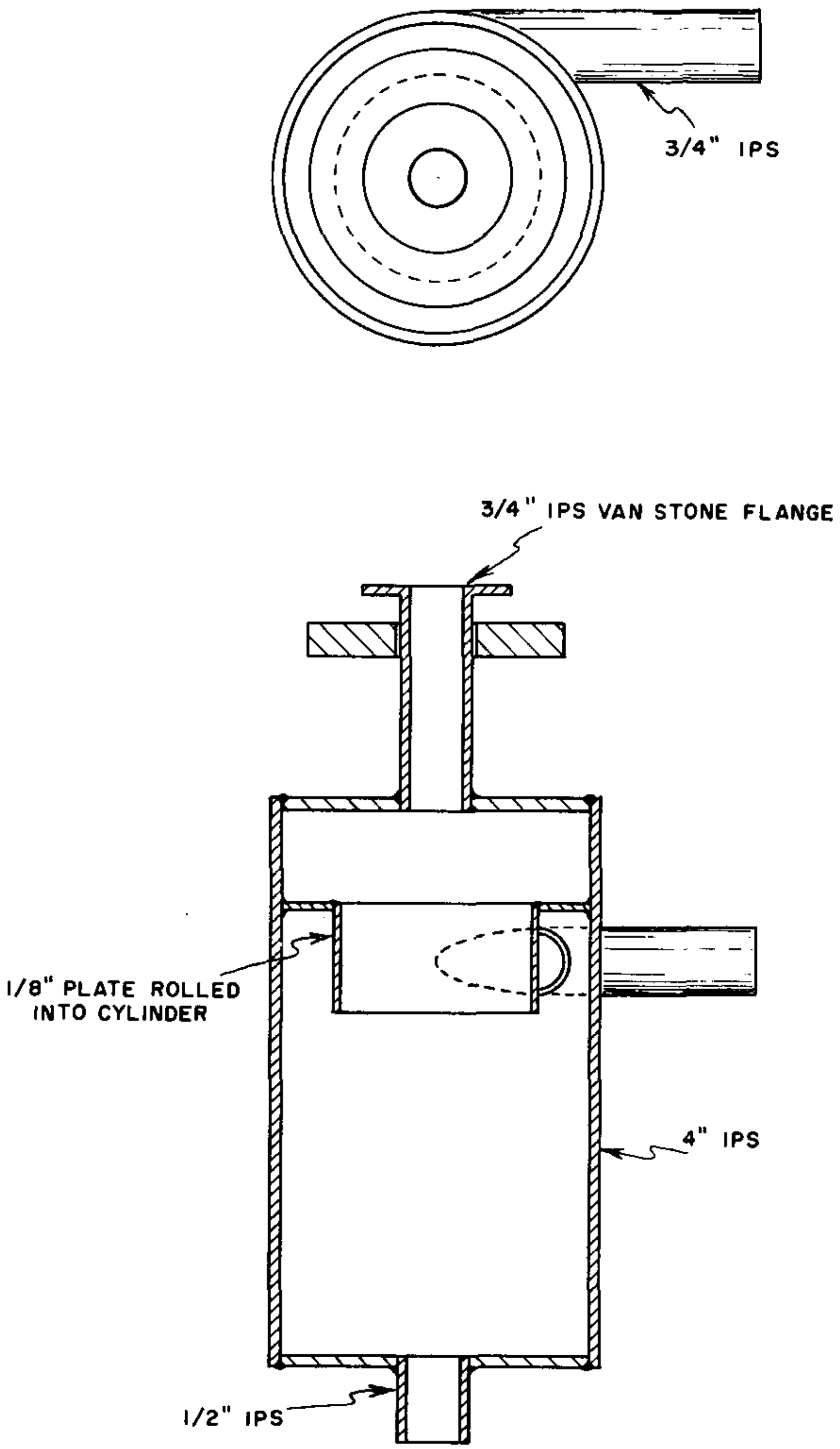
FEED TANK

EVAPORATOR JACKET

EVAPORATOR

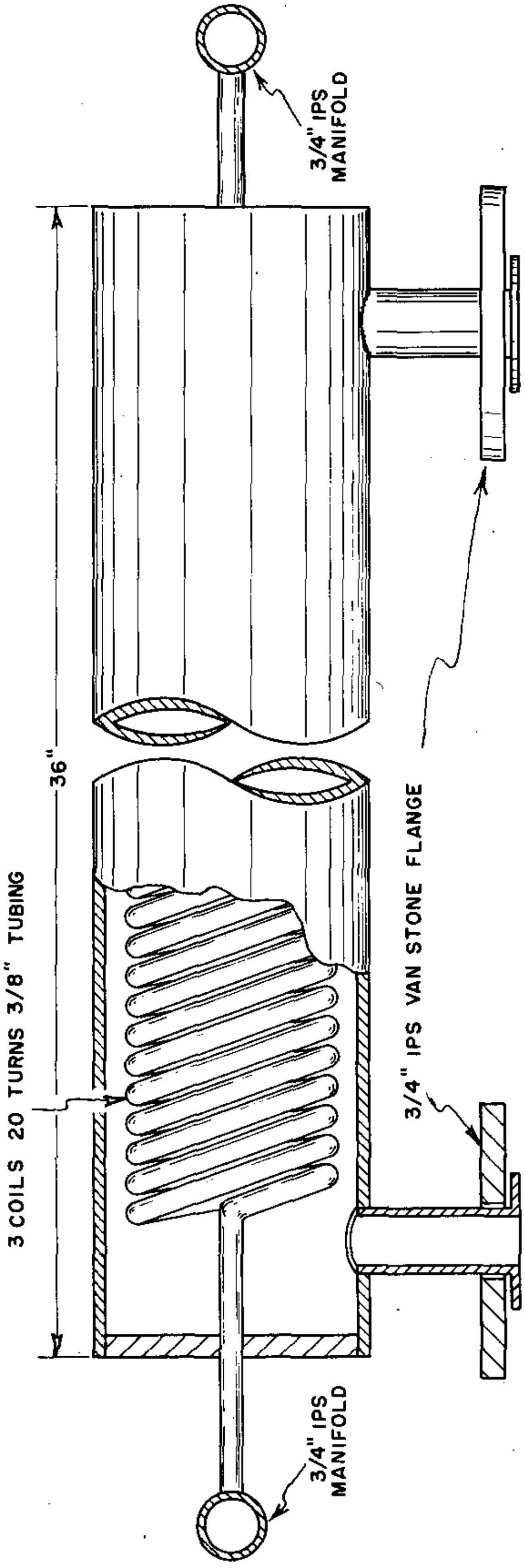
FEED TANK AND PIPE EVAPORATOR DETAILS

FIGURE 7



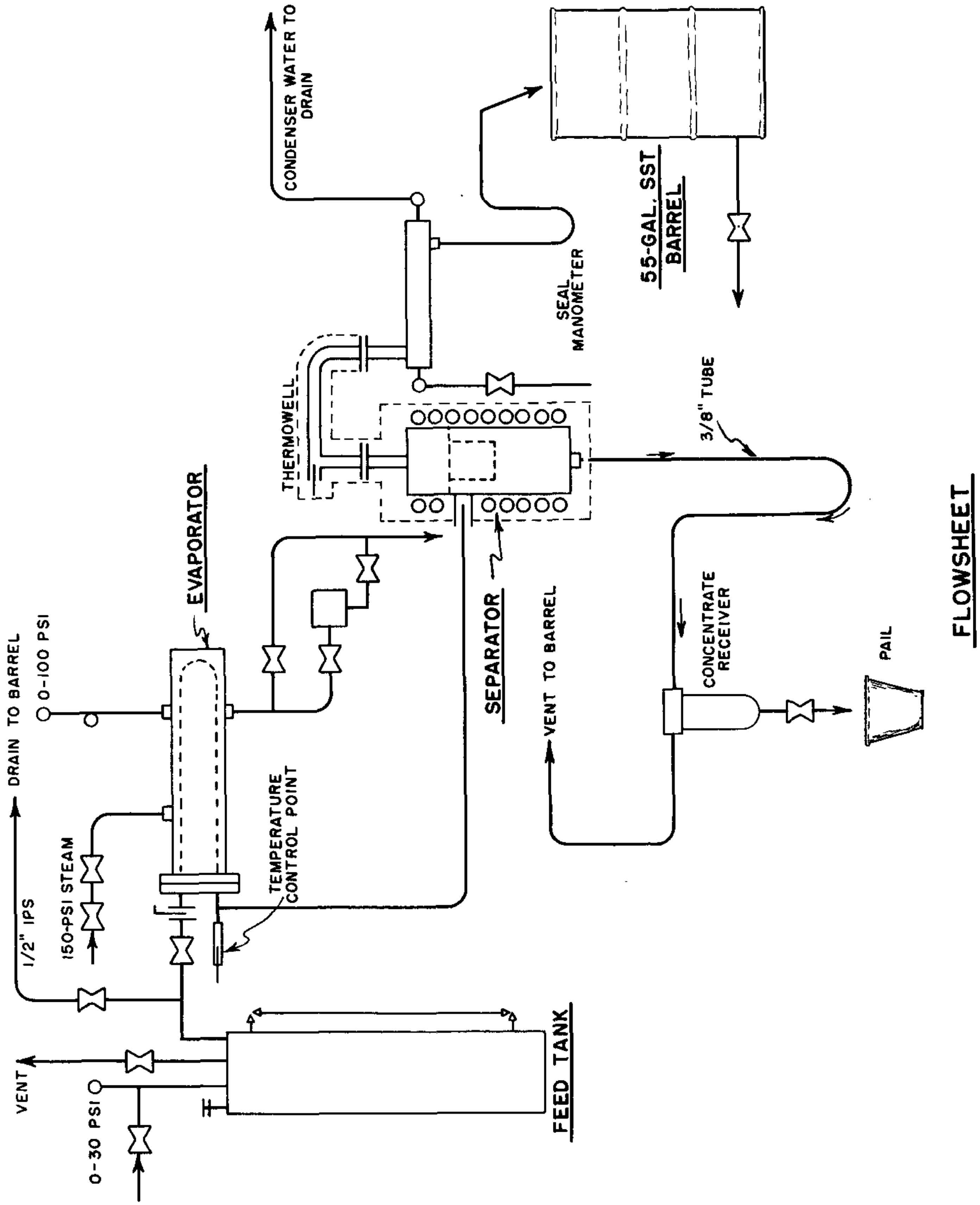
SEPARATOR DETAILS

FIGURE 8



CONDENSER DETAILS

FIGURE 9



FLOWSHEET

TABLE I

MANUAL OPERATION

RUN	STEAM PRESSURE P_s, psig	INLET FEED PRESSURE P_o, psig	THROUGHPUT W, lb/hr	FRACTION EVAPORATED x
1 (a)	41.0	17.0	31.4	0.9913
(b)	38.0	17.0	31.3	0.9968
(c)	40.5	17.0	30.5	0.9989
(d)	40.0	17.0	30.6	0.9995
2 (a)	31.0	15.5	34.5	0.84
(b)	30.0	15.5	35.3	0.75
(c)	30.5	15.5	38.3	0.78
(d)	29.5	15.0	36.2	0.81
(e)	30.3	15.5	33.3	0.85
(f)	30.4	15.5	36.7	0.82
3 (a)	34.0	17.5	30.9	0.977
(b)	35.0	17.5	31.0	0.959
(c)	35.0	17.5	30.6	0.962
(d)	36.0	17.0	30.5	0.971
(e)	33.0	16.8	31.0	0.952
(f)	33.0	16.7	30.9	0.943
4 (a)	26.2	15.0	38.3	0.65
(b)	26.0	14.6	38.0	0.67
(c)	26.0	14.6	37.6	0.68
(d)	26.0	14.6	35.7	0.71
(e)	26.0	14.7	37.2	0.69
5 (a)	45.8	23.0	39.3	0.935
(b)	44.0	23.0	41.1	0.911
(c)	45.5	23.0	39.9	0.928
(d)	43.0	22.6	39.2	0.88
6 (a)	37.0	20.8	47.2	0.61
(b)	35.0	20.9	47.2	0.62
(c)	35.5	20.8	47.2	0.64
(d)	34.8	20.5	47.7	0.59
(e)	35.3	21.0	47.7	0.66
7 (a)	55	23.8	40.5	1.00
(b)	53	24.0	39.1	1.00
(c)	52.2	24.0	36.7	1.00
(d)	51.2	24.2	37.7	1.00

TABLE II

OPERATION WITH INSTRUMENTATION

RUN	STEAM PRESSURE P_s (Avg) psig	INLET FEED PRESSURE P_o (Avg), psig	THROUGHPUT W, lb/hr	FRACTION EVAPORATED, x	$\frac{(P_s P_o - 1/2\, P_o^2)}{W^3} \times 100$	x^3	U_o
31	29.0	16.0	36.3	0.72	0.71	0.38	778
32	31.0	16.0	34.3	0.84	0.91	0.58	771
33	31.0	16.3	33.8	0.81	0.96	0.53	744
34	31.0	16.3	35.3	0.80	0.85	0.51	766
35	36.0	17.0	32.7	0.964	1.34	0.90	726
36	36.0	16.5	32.3	0.961	1.36	0.89	707
37	35.5	17.3	32.3	0.962	1.38	0.89	734
38	36.8	17.5	32.3	0.975	1.46	0.93	719
39	37.2	17.3	32.3	0.978	1.46	0.93	709
40	37.6	17.5	32.3	0.983	1.49	0.95	710
41	31.8	16.5	34.9	0.83	0.91	0.57	770
42	28.8	16.0	37.5	0.75	0.63	0.41	838
43	28.9	15.8	37.5	0.73	0.63	0.39	811
45	32.0	16.5	34.7	0.83	0.94	0.58	760
46	32.0	16.5	34.5	0.84	0.95	0.59	765
47	33.0	16.5	33.3	0.89	1.11	0.71	748
48	31.5	16.5	34.2	0.84	0.96	0.59	770
50	32.3	16.5	34.2	0.87	0.99	0.65	775
51	32.5	17.0	34.2	0.88	1.02	0.63	789
52	32.3	17.5	35.1	0.86	0.95	0.64	798
54	24.3	14.0	42.1	0.53	0.32	0.146	808
55	24.0	13.1	42.2	0.51	0.30	0.131	761
56	24.3	13.5	41.5	0.53	0.33	0.152	776
57	36.6	18.0	32.7	0.962	1.42	0.89	736

TABLE III

FLUID FLOWS AND URANIUM CONCENTRATIONS

RUN NO	FEED RATE, cc/min	PRODUCT RATE, cc/min	FRACTION EVAPORATED, x	URANIUM MATERIAL BALANCE, per cent	FEED ANALYSIS, gm U/liter	PRODUCT ANALYSIS, gm U/liter	OVERHEAD VAPOR ANALYSIS, gm U/liter	PER CENT URANIUM LOSS IN VAPOR
31	274	76.3	0.72	105	2.11	7.75	0.068	2.31
32	260	42.8	0.84	95	2.05	11.8	0.073	2.95
33	256	48.8	0.81	118	1.60	9.57	0.087	4.38
34	267	54.2	0.80	95	2.05	9.35	0.082	4.23
35	248	8.95	0.964	90	1.65	39.1	0.077	4.48
36	245	9.62	0.961	84	2.20	43.8	0.139	6.08
37	245	9.32	0.962	92	2.01	45.5	0.129	6.18
38	245	6.06	0.975	83	2.09	64.0	0.131	6.10
39	245	5.40	0.978	76	1.91	60.3	0.125	6.42
40	245	4.21	0.983	60	1.46	45.6	0.109	7.38
41	264	45.7	0.83	92	0.39	2.05	0.016	3.40
42	284	72.3	0.75	92	0.12	0.43	0.0032	2.03
43	284	75.9	0.73	76	0.31	0.86	0.0072	1.65
45	262	43.7	0.83	86	0.39	1.90	0.017	3.75
46	261	41.7	0.84	129	0.41	3.22	0.015	3.05
47	252	27.6	0.89	95	1.93	16.0	0.092	3.53
48	260	42.3	0.84	92	1.65	9.15	0.075	3.77
50	260	34.6	0.87	113	1.58	13.2	0.071	3.19
51	260	30.8	0.88	103	1.77	14.9	0.049	2.45
52	266	36.5	0.86	101	1.67	11.8	0.046	2.38
54	319	151	0.53	48	1.98	4.75	0.047	1.25
55	320	158	0.51	103	1.78	3.66	0.035	1.00
56	315	147	0.53	101	1.92	4.08	0.030	0.84
57	248	9.42	0.962	44	1.98	17.0	0.125	6.07