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final report
phase #1

ANALYSIS AND TESTING THE PERFORMANCE
OF A CENTRIFUGAL TWO PHASE FLOW SEPARATOR

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by
Alexander Vicson Mirza-Moghadam
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Approved;

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Alexander Vicson Mirza-Moghadam
Department of Mechanical Engineering
University of California
Berkeley, CA 94720
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ABSTRACT

Analysis and testing the performance of an 8", 1.72' high centrifugal cyclone separator for flows up to $4.0 \frac{\text{lbs}}{\text{s}}$ and pressures ranging from 10 to 60 psig. Conclusions drawn are based on inlet steam qualities of 23 to 27 percent ($x\% = \frac{\dot{m}_s}{\dot{m}_t}$) .99% and better steam quality is achieved up to $3 \frac{\text{lbs}}{\text{s}}$ under 50 and 60 psig. Breakdown flow rate is found to be a linear function of separator pressure.

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EXPERIMENT: PERFORMANCE TEST OF SEPARATOR

NOMENCLATURE:

P_{si} = inlet steam pressure, psig

P_{wi} = inlet water pressure, psig

$\dot{m}_{wi}$ = inlet water flow, see calibration graph

P_{SEP} = separator pressure, psig

T_{Th} = calorimeter temperature, °F

P_{Th} = calorimeter pressure, inches of mercury

$\dot{m}_{wo}$ = water flow out, $\frac{\text{lbs}}{\text{s}}$.

$\dot{m}_{so}$ = steam flow out, $\frac{\text{lbs}}{\text{s}}$.

K = stem correction = $.00009 (T_{Th} + 48) (T_{Th} - T_{\infty})$

TOTAL FLOW: Defined as $\dot{m}_{wi} + \dot{m}_{so} = \dot{m}_t$

SUMMARY

A hybrid system consists of two or more energy conversion processes. Due to the variety of thermodynamic states and chemistries in geothermal resources, no one energy conversion machine or process can satisfy all conditions. For this reason, several stages are integrated into one system. The staging of a single-flash system with the Lysholm engine (a 2-phase expander) as the first stage, and a separator leading to the second stage, which consists of a conventional low pressure vapor-phase turbine and a radial outflow reaction turbine, appears to be capable of generating geothermal power within the state of present technology. This study examines the performance of an 8-inch two-phase flow cyclone separator with a 2" inlet diameter, 1.72 feet high, for flow rates ranging from $.5 \frac{\text{lbs}}{\text{s}}$ to $4.0 \frac{\text{lbs}}{\text{s}}$ having steam qualities of 23%-27%, under 10 psig to 60 psig pressure. From observations of the data taken, the separator will yield 99.3% to 99.6% steam at these pressures for flows up to $2.9 \frac{\text{lbs}}{\text{s}}$. Knowledge of this result will enable the hybrid staging to be most efficient in selecting optimum operating conditions.

INTRODUCTION

Geothermal energy is a heat source, and not simply a steam source, for which we are seeking a variety of applications. Exploration on a global basis appears to indicate that geothermal steam fields are comparatively rare, relative to geothermal hot water fields. Thus, the Geysers field, which is the biggest dry steam field discovered in the U.S. so far should not deviate the attention from the discovery of hot water fields. The Wairakei field in New Zealand is a typical hot water field and part of the proper technology for wet steam development was evolved in New Zealand. It

was there that highly efficient separators were developed which are an essential ingredient in the handling of hot water fields. When we drill into a wet steam field, we are drilling into a reservoir where there is hot water under high pressure. When this hot water reaches the well-head, the pressure decreases, hydrostatically. As a result, up to 40% (25% average) of the water flashes into steam. The separator, after separating the 25% steam from the fluid, sends this steam to the power station, and the remaining hot water can be used for 1) geothermal desalination; 2) generating power by use of a radial outflow reaction turbine; 3) extending the life of the reservoir by reinjecting back into the formation.⁴

Depending upon the well-head conditions (T , P , H , $\dot{m}$), there exists an optimum separator pressure and flow rate. As the pressure is reduced, mass flow of steam to the turbine increases and the available energy decreases. The optimum separation pressure is a balance between these two. However, every separator has its breakdown point at which either it cannot physically handle all of the flow, or if it could, as the pressure within the separator is reduced, the quality of the separated steam decreases. Unfortunately, due to the complexity of the mechanism of phase separation inside the vortex flow in the cyclone separator, and the contingency of entrainment of water droplets into the blowing steam inside the vortex, it is rather difficult to determine the optimal design for the separation unit by means of analytical estimates. Extensive experiment has to precede the design, and in so doing, develop empirical relations.

TEST APPARATUS AND PROCEDURE

The separator tested was a centrifugal type with the two phase flow entering tangentially at an inclined angle of 10° . The main cylinder was

closed by two flanges (see Figure I for dimensions). There was an inner tube concentric with the main cylinder, whose length was arbitrarily chosen as $3/4$ of the length of the cylinder. The steam exhausted down this tube. To simulate two-phase flow, cold water at 60°F , was drawn in a 1" diameter pipe by a centrifugal pump driven by a 10HP, 3515 RPM General Electric, A.C. motor. A traditional flow-meter whose calibration curve is included in the appendix measured the water flow. A pressurized, 110 psig, steam line flowed in a 2" pipe, and the water line introduced itself into the steam line. The intersection took place at about 2 feet from the entrance to the separator. The hot water left the separator in a 2" pipe, finding itself in a weigh tank for measuring its flow rate. Periodically, the weigh tank had to be drained or cold water added for volume and temperature control, respectively. The hot water level in the separator was monitored through a sight-glass. A small fraction of the separated steam, after leaving the separator, passed through a $5/32$ nozzle throttling calorimeter to measure steam quality. The separated steam would then exit from a 2" pipe, condense in a conventional condenser and finally, enter another weigh tank. The manner at which flow rates were measured by the "weighing tank" technique was to time the reaching of a certain weight of fluid in the tank. However, for the hot water tank, a numbered scale continuously showed the weight of the water, but for the steam tank, weights of 25 or 50 lbs. were used in conducting the flow timing. A complete schematic diagram of the set-up and instrumentation appears in the appendix.

The procedure of testing was as follows: The desired flow rate and pressure of the two-phase was chosen (i.e., $1.5 \frac{\text{lbs}}{\text{s}}$, 60 psig). Being interested in the inlet quality of 20 ~ 30% steam (i.e., 25%), the amount of water measured by the flow-meter would be fixed (i.e., $1.5 \times .75 = 1.125 \frac{\text{lbs}}{\text{s}}$).

Several flow measurements of the steam weigh-tank and corrections by opening/closing valves, would result to the separator pressure reading 60 psig, the water flow meter showing $1.125 \frac{\text{lbs}}{\text{s}}$ and the steam-out, measuring in the range of $.35 \sim .4 \frac{\text{lbs}}{\text{s}}$. The condenser temperature was then recorded to ensure constant flow rate for subsequent pressures. For example, to find the performance of the separator under 20 psig pressure and the above flow rate, any valve adjustment to give this pressure would be made such that the condenser temperature remained what it was at 60 psig, (without touching the valve to adjusting cold water flow into the condenser), and the water flow-meter would also remain fixed. If another flow rate was chosen on the same testing period, a new reference for condenser temperature was established. At each flow rate, steam quality in the two-phase mixture, and separator pressure, the following data were recorded: 1) run of experiment; 2) campus steam-line pressure, P_{sl} ; 3) city cold water-line pressure, P_{wl} ; 4) throttling temperature, T_{th} ; 5) throttling pressure above atmospheric, P_{th} ; 6) ambient temperature, T_{∞} ; 7) barometric pressure, P_{bar} ; 8) flow rate of outcoming water and steam, $\dot{m}_{wo}$, $\dot{m}_{so}$; 9) condenser temperature, T_c .

Initially, the experiment was conducted without knowledge of the significance of correct or optimum water level height in the separator, and the effect it may have as it fluctuated. This, is why the throttling steam temperature would drastically change as the water level in the separator fluctuated. Thus, separate tests were conducted under set water level heights (Runs 47 → 53). The result was unexpected! High throttling temperatures were achieved at low water levels (4 inches up the sight-glass). Beyond Run 53, all data were recorded with this water level. Thus, performance data were evaluated beyond run 53, only, and uncontrolled water level data were analysed for comparison.

SEPARATOR LAY-OUT

COSMOS ENGINEERS, INC.
BETHESDA, MARYLAND 20034

DRAWN A.V.M.

ISSUED

REV	DWG NO.	SCALE	WT	SHEET	OF
	A 55353	1:10			

TEST RESULTS AND CALCULATIONS

The results of the separator performance are summarized in Figures 1-9. Curves 1 through 6 represent points whose two-phase qualities range from 23 to 27 percent. Thus, error must not be regarded as how the points differ from the curve drawn. The precision of measurements are stated in Table 1 (see Appendix B for details).

<u>Variable</u>	<u>Tolerance</u>
P	± 2 %
T	± .2% (Temp. scale accuracy : $\frac{\pm .5^{\circ}\text{F}}{250^{\circ}\text{F}} = \pm .2\%$)
$\dot{m}_{\text{SO}}$	± 5 %
$\dot{m}_{\text{WO}}$	± .5 %
(water level) h	± 15 %
X%	± .3 %

The method of determining steam quality of the two-phase mixture was to divide the separated steam flow rate by the total flow. To determine the separated steam quality under a particular pressure, the value of the superheated vapor enthalpy, H , for the adjusted atmospheric pressure and corrected throttling temperature was read from steam tables. Then for the prescribed absolute pressure, the fluid and latent heats, H_F and H_{Fg} , were read from (pressure) saturation tables.

$$X_{\text{SO}} = \frac{H - H_F}{H_{Fg}}$$

In Figure 1, separated steam quality ($\sim 99.7\%$) was fairly uniform up to about

$2.9 \frac{\text{lbs}}{\text{s}}$ where the height of the curve began to increase (breakdown point). However, even at $3.5 \frac{\text{lbs}}{\text{s}}$, the quality was 99.4%. At 50 psig, Figure 2 indicated uniform quality (99.6%) up to about the same $2.9 \frac{\text{lbs}}{\text{s}}$, but this time an increase in wetness was observed beyond $3.0 \frac{\text{lbs}}{\text{s}}$, such that at $3.4 \frac{\text{lbs}}{\text{s}}$ the steam quality dropped to 98.1%. An additional increase was observed until $3.7 \frac{\text{lbs}}{\text{s}}$, where wetness was 3%, and it remained at $4.0 \frac{\text{lbs}}{\text{s}}$. In Figure 3, 99.5% steam quality was achieved up to $2.75 \frac{\text{lbs}}{\text{s}}$, where breakdown began. At $3.4 \frac{\text{lbs}}{\text{s}}$, the quality was 97.7%, and remained at $4.2 \frac{\text{lbs}}{\text{s}}$. 30 psig exhibited 99.5% steam up to $2.5 \frac{\text{lbs}}{\text{s}}$ ^{where} more wetness was observed. Between $3 \frac{\text{lbs}}{\text{s}}$ and $4 \frac{\text{lbs}}{\text{s}}$, quality measured 98.1%. Figures 5 and 6 each contain four points only and no information on separator performance was available for higher flow rates. This was because the control valve on the steam tube of the separator was completely opened, and yet the desired pressure (20 or 10 psig) was not reached. Figure 7 showed flow rate versus minimum separator pressure. In this graph, minimum pressure for a flow of $2.8 \frac{\text{lbs}}{\text{s}}$ was 20 psig. For $1.6 \frac{\text{lbs}}{\text{s}}$, minimum pressure was 10 psig. However, in Figure 6, the last two points ($1.6 \frac{\text{lbs}}{\text{s}}$) plotted were with minimum pressure 13 and 15 psig rather than 10 psig. 99.3% steam quality was obtained for pressures of 10 and 20 psig and up to $1.3 \frac{\text{lbs}}{\text{s}}$ where breakdown began. But, even at 20 psig and $2.2 \frac{\text{lbs}}{\text{s}}$, quality was 99.2% and a drastic breakdown could not be detected due to insufficient data points. Therefore, it is questionable to consider the change of steam quality from 99.3% to 99.2% (at 20 psig) as a breakdown point, rather than just a random error in data.

Obviously, Figure 9 considered it to be a breakdown point, just to be conservative. In Figure 9, separator pressure was plotted against breakdown or critical flow rate. Also the quality of steam at each pressure and critical flow was stated. One may consider this graph as the final

result since it specifies separator performance until the beginning of break-down. There appears to be a kind of symmetry in that the curve levels at the two highest and lowest pressures in both steam quality and flow rate. This type curve is best described by the method of least squares to be

$$\dot{m}^* = .955 + P_{SEP}(.0371) \quad 10 < P < 60 \text{ PSIG}$$

The linear relation is chosen to specify that the higher the pressure in the separator, the more flow can be separated effectively. The equation to relate quality and pressure is already known from thermodynamic principals. Figure 8 showed the significance of water level control. The lower the level, i.e., less flow rate, the higher the quality. Two curves based on two separate pressure sets were plotted.

DISCUSSION

The experiment had its disadvantages. Cold water was introduced into the steam line instead of hot water. In this case, since the intersection took place very close to the separator, one cannot be sure whether the flow reached equilibrium or not. In all cases, the flow of water coming out of the separator had increased in the order of 15%. This increase either came from the passage of steam in the exit water pipe, or condensation of steam due to ^{Partial} equilibrium. A preliminary calculation verified the latter case. In run 55, e.g.,: For $P_{SEP} = 50 \text{ psig}$, $h_F = 267.5 \frac{\text{Btu}}{\text{lb}_m}$, $h_g = 1179.1 \frac{\text{Btu}}{\text{lb}_m}$, $h_F = 911.6 \frac{\text{Btu}}{\text{lb}_m}$. How much steam is lost when $1.05 \frac{\text{lb}}{\text{s}}$ of water @

62°F emerges into saturated vapor @ ⁹212°F, $.73 \frac{\text{lb}}{\text{s}}$?

SEPARATOR PERFORMANCE
VS.
TOTAL FLOW & INLET STEAM
QUALITY AT 60 PSIG.

COMPUTER GRAPHICS

1. SHOWN BELOW ARE ALGEBRAIC STATEMENTS THAT LOCATE THE VERTICES OF A HEXAGON. THE CALL PLOT STATEMENT IS USED TO COMMAND THE PLOTTER TO CONNECT THE VERTICES. NOTICE THAT THE CALCULATIONS FROM POINT TO POINT ARE IDENTICAL EXCEPT FOR THE ANGLES. USING A DO STATEMENT, WRITE A PROGRAM THAT WILL INCREMENT THE ANGLE BY 60° SIX TIMES.

```

R = 1.0
GO TO POINT 1
CALL PLOT (R, 0, 3)
GO TO POINT 2
X = R COS 60°
Y = R SIN 60°
CALL PLOT (X, Y, 2)
GO TO POINT 3
X = R COS 120°
Y = R SIN 120°
CALL PLOT (X, Y, 2)
GO TO POINT 4
X = R COS 180°
Y = R SIN 180°
CALL PLOT (X, Y, 2)
GO TO POINT 5
X = R COS 240°
Y = R SIN 240°
CALL PLOT (X, Y, 2)
GO TO POINT 6
X = R COS 300°
Y = R SIN 300°
CALL PLOT (X, Y, 2)
GO TO POINT 1
X = R COS 360°
Y = R SIN 360°
CALL PLOT (X, Y, 2)
RAISE PEN
CALL PLOT (X, Y, 3)
    
```

2. A POLYGON WITH ANY NUMBER OF SIDES INSCRIBED INSIDE A CIRCLE CAN BE PLOTTED IN A MANNER SIMILAR TO PROBLEM 1, BY SPECIFYING THE NUMBER OF SIDES AND THE ANGLE TO BE REPEATED. WRITE A PROGRAM THAT WILL PLOT A POLYGON WITH ANY NUMBER OF SIDES DESIRED.

SEPARATOR PERFORMANCE
VS.
TOTAL FLOW & INLET STEAM
QUALITY AT 50 PSIG.

COMPUTER GRAPHICS

1. ASSUME THAT THE COMPUTER HAS GIVEN YOU THE PRINT OUT OF THE X, Y, Z, AND IPEN VALUES SHOWN. THESE ARE THE COORDINATES OF EACH POINT OF THE PICTORIAL SHOWN. PLOT THE PICTORIAL BY HAND BY MOVING TO THE POINTS INDICATED BY THE COORDINATES TO VERIFY THAT THE DATA IS CORRECT. USE THE COORDINATE AXES GIVEN.

X	Y	Z	IPEN
0	0	0	3
5	0	0	2
5	1	0	2
2	1	0	2
2	4	0	2
0	4	0	2
0	0	0	2
0	0	3	2
5	0	3	2
5	1	3	2
2	1	3	2
2	4	3	2
0	4	3	2
0	0	3	2
0	4	3	3
0	4	0	2
2	4	0	3
2	4	3	2
2	1	3	3
2	1	0	2
5	1	0	3
5	1	3	2
5	0	3	3
5	0	0	2
0	0	0	3

2. WRITE A PROGRAM THAT WILL READ THE DATA INTO STORAGE LOCATIONS X, Y, Z AND IPEN. IN FORTRAN, THE PROGRAM WILL START WITH THE STATEMENT: DIMENSION X(25), Y(25), Z(25), IPEN(25).

3. A PROGRAM FOR DRAWING THE FRONT AND TOP VIEWS IS GIVEN. THE PLOTTER ONLY UNDERSTANDS A TWO-DIMENSIONAL COORDINATE SYSTEM (XPLOT & YPLOT). THE XPLOT IS THE SAME AS X-COORDINATE IN BOTH THE FRONT AND TOP VIEWS. IN THE TOP VIEW, YPLOT IS EQUAL TO THE Z-COORDINATE; THEREFORE, YTRANS IS ADDED TO THE Z-COORDINATES TO PREVENT THE TOP VIEW FROM BEING PLOTTED ON TOP OF THE FRONT VIEW.

COMPLETE THE FORTRAN PROGRAM FOR COMMANDING THE PLOTTER TO DRAW THE RIGHT SIDE VIEW.

```

C  NDATA IS THE NUMBER OF DATA POINTS
    NDATA = 25
    XTRANS = 7.
    YTRANS = 6.
C  THIS DRAWS THE FRONT VIEW
    DO 10 J=1,NDATA
      XPLOT = X(J)
      YPLOT = Y(J)
      IP = IPEN(J)
      CALL PLOT (XPLOT, YPLOT, IP)
10  CONTINUE
C  THIS DRAWS THE TOP VIEW
    DO 20 J=1,NDATA
      XPLOT = X(J)
      YPLOT = Z(J) + YTRANS
      IP = IPEN(J)
      CALL PLOT (XPLOT, YPLOT, IP)
20  CONTINUE
  
```

SEPARATOR PERFORMANCE
VS.
TOTAL FLOW & INLET STEAM
QUALITY AT 40 PSIG.

SEPARATOR PERFORMANCE
VS.
TOTAL FLOW & INLET STEAM
QUALITY AT 30 PSIG.

SEPARATOR PERFORMANCE VS.

TOTAL FLOW & INLET STEAM
QUALITY AT 20 PSIG.

ISOMETRIC ELLIPSES: When sketching isometric pictorials on tracing paper, elliptical views of circles may be sketched by overlaying the tracing paper over the appropriate ellipse given on this sheet.

SEPARATOR PERFORMANCE
VS.
TOTAL FLOW & INLET STEAM
QUALITY AT 10 PSIG.

Quickie Problems

General: These problems will be assigned at intervals throughout the course as an exercise in communication of ideas in a rapid fashion. Rapid communication is a more important objective than the exercise of creativity. Problems are to be sketched on worksheets which are included at the back of this book using notes, views and pictorials in any combination. These will be used early in the course as well as at later stages to emphasize the importance of communication to the beginning student.

Time limit for each problem should not exceed more than 15 minutes. Emphasis is on rapid communication. Develop as many ideas as time will permit.

1. **TV YOKE:** Design a yoke that can be used for attaching a TV set to a classroom ceiling or a vertical wall for closed circuit television.
2. **TOWEL BAR:** Design a towel bar for a bathroom. Indicate the method of attaching the bar to the wall.
3. **LAWN MOWER HANDLE:** Design a handle for lawn mowers. Consider comfort of the user.
4. **FLAG POLE SOCKET:** Design a socket that can be attached to a vertical wall to hold a flag at a 45° angle with the wall.
5. **FAUCET HANDLES:** Sketch several designs for faucet handles that would be convenient for a housewife with soapy hands to use at a kitchen sink.
6. **POOL CUE RACK:** Design a rack for holding four pool cues on a vertical wall.
7. **CABINET HANDLE:** Design a handle for a standard file cabinet. Consider human factors and comfort of the user.
8. **CRUTCHES:** Design a crutch for a person with an injured leg. Consider new features that would be an improvement over the presently used crutches.
9. **CUP HOLDERS:** Design a holder that could be attached to an automobile door or interior to hold a soft drink cup or can.
10. **GATE HINGE:** Design a hinge that could be attached to a 3" tubular steel post to support a 3 foot wide gate.
11. **SOAP DISH:** Design a soap dish that can be attached to a shower head to provide extra space for soap or wash cloths.
12. **TOOTHBRUSH HOLDER:** Sketch ideas for a workable design for a toothbrush holder that can be attached to a bathroom wall. The holder should contain a drinking cup (2-1/2" diameter) and two toothbrushes.
13. **DRAWER HANDLE:** Design a handle to be attached to a 9" x 12" drawer of a typical filing cabinet. This design should take into account the size and comfort required by the range of hand sizes that will be using the handle.
14. **CLOTHES HOOK:** Design a simple clothes hook that can be attached to a closet door for hanging clothes. This hook should be easy to manufacture, economical and simple to attach to a closet door.
15. **PAPER CUP DISPENSER:** Design a paper cup dispenser that could be attached to a kitchen wall. This dispenser should hold a series of cups 2" in diameter that measure 6" when stacked together. This dispenser should be as simple as possible to permit it to be economically produced.
16. **LEVELING BLOCK:** Design a device for leveling a 1000 pound piece of machinery. This is to be placed under a foot of the machines' base and will be adjusted with a bolt turned by a wrench.

Fig. 7

8" DIA, 1.72' LONG CYCLONE (PERFORMANCE CHARACTERISTICS)
 2" DIA INLET/OUTLET AT BREAKDOWN

$$\Delta T = 212 - 62 = 150^\circ \text{F}, \quad C_p = 1 \frac{\text{BTu}}{\text{lb}_m^\circ \text{F}}$$

x 62: partial thermal
equil

$$h = C_p \Delta T = 150 \frac{\text{BTu}}{\text{lb}_m}$$

$$E = \dot{m} h = (1.05 \frac{\text{lb}_m}{\text{s}}) (150 \frac{\text{BTu}}{\text{lb}_m}) = 157.5 \frac{\text{BTu}}{\text{s}}$$

ENERGY NEEDED FOR
1.05 $\frac{\text{lb}_m}{\text{s}}$ OF WATER
TO REACH 212°F.
262°

$$\Delta E_{\text{STEAM}} = (.73)(1179.1) - 157.5 = 703.24 \frac{\text{BTu}}{\text{s}}$$

ENERGY OF
STEAM IN 2-PHASE

$$h = \frac{703.24}{.73} = 963.33 \frac{\text{BTu}}{\text{lb}_m}$$

$$963.33 = 267.5 + X(911.6)$$

$$X = .76$$

$$\dot{m}'_s = (.73 \frac{\text{lb}_m}{\text{s}}) (.76) = \boxed{\frac{.56}{0.5} \frac{\text{lb}_m}{\text{s}}}$$

FLOW RATE OF
STEAM IN 2-PHASE

.73 - .56 = .17 $\frac{\text{lbs}}{\text{s}}$ added to water (~23%) during mixture. Examination of data taken in run 55 indicated $\dot{m}_{\text{so}} = .48 \frac{\text{lbs}}{\text{s}}$ (steam out of separator). Thus, there was ~.08 $\frac{\text{lbs}}{\text{s}}$ steam less than what it should have measured (~14%). This loss most probably generated from the entrainment of steam down the exit water, or calorimeter pipe, and/or loss of heat due to conduction into separator walls. As for the 23% original condensation, this problem could have easily been resolved by drawing hot saturated water (~212°F) into the steam line. But the amount of steam condensed in the separator depends on separator design which in turn is

specified by a maximum allowable vapor velocity

$$V_{\text{MAX}} \approx \left[\left(\frac{\dot{m}_w}{\dot{m}_s} \right) \left(\frac{\rho_s}{\rho_w} \right)^{\frac{1}{2}} \left(\frac{\rho_w - \rho_s}{\rho_s} \right)^{\frac{1}{2}} \right]$$

where subscripts s and w denote steam and water, respectively. The term in

brackets is proportional to a constant (K_v) which in general is about

$$.35 \Rightarrow v_{\max} \approx .35 \left(\frac{\rho_w - \rho_s}{\rho_s} \right)^{\frac{1}{2}}$$

Since this is a design velocity, we can find the proper separator dimension

via: $A_{\min} = \frac{Q_s}{(v_s)_{\max}}$ to give $D_{\min}$. The key to good performance

is to provide a sufficiently low velocity so that vapor and liquid will separate. As mentioned above, inlet velocity is based only on the nominal

gas velocity ($v_i = \frac{\dot{m}_{si}}{\rho_s A_i}$) because it ignores the displacement of the

water phase in the two-phase flow. The displacement of the water phase is virtually impossible to assess because its velocity, even in the absence of slugging flow, would be difficult to calculate. It is certain that its velocity

would differ from that of the steam phase;³

$$60\sqrt{\rho_{\text{mix}}} < v_i < \frac{100}{\sqrt{\rho_{\text{mix}}}}$$

(See Appendix D for details.) Another important subject to be considered in separator performance is the pressure loss and the manner in which a particular size of droplet behaves. The former suggests that the higher the inlet steam velocity, the higher the pressure drop across the separator which in itself, is a kind of loss, both in energy and steam in the separator. The latter suggests the calculation of the spinning speed at any radius and hence the forces acting on the droplets to determine whether they will drift towards the walls of the cyclone and so be separated, or towards the axis and escape with the steam. In addition, the generation of droplets due to the blowing

of steam over the water level in the separator is another problem to be

considered. First, the pressure drop, $\Delta P \sim V^2$, $\frac{\partial P}{\partial r} = \rho \frac{V^2}{r}$
 $P + \rho \frac{V^2}{2g} = \text{const.} \Rightarrow \frac{\partial P}{\partial r} + \rho V \frac{\partial V}{\partial r} = 0 \rightarrow \frac{\partial V}{\partial r} + \frac{V}{r} = 0 \Rightarrow \boxed{Vr = \text{const.}} \quad \text{FREE VORTEX}$

where r is radius along

the separator from the central core to the outer walls. ⁸ $P_2 > P_1$ For water droplet This is the pressure distribution outside the core (steam pipe).

The general characteristics of rotating flows within a stationary container exhibits instabilities and fluctuations. In the vortex flow, the circumferential velocity must be zero at the center and at the outer wall and greater than zero in between. Thus, the flow tends to be stable near the center of rotation and unstable near the outer wall. At very low velocities the viscous forces are predominant and no instability is present. At the normal operating velocities of the cyclone, the region which tends to be unstable is confined to a relatively thin layer near the outer wall. In this layer the circumferential velocity is reduced by viscous shear forces and the thickness of the layer grows to such an extent that it is then able to draw slow-moving fluid from the surrounding. This growth is the result of the instability of the slow-moving fluid with respect to displacement radially inward. The inward diffusion of the chunks of slower moving fluid produce a strong turbulence throughout almost all of the separator. Thus, the flow in the separator is very turbulent even if there is no turbulence convected into the separator by the incoming fluid. This turbulence resulting from the unstable layer is inherent in any device employing a rotating flow within a stationary container. The inward flow of the boundary layer carries with it the smaller water droplets in the boundary layer. This is a mechanism by which some of the droplets that have been separated into the boundary layer are mixed back into the steam in the cyclone and are not separated.¹ In

testing the separator, this theory was verified by the quality of the separated steam. Also, the turbulence was clearly visible in the water level fluctuations and low velocities gave better separation. In a cyclone with a single tangential inlet, the water droplet streak begins its course at the intersection of the bottom wall of the inlet with the cyclone separator.

SPINNING SPEEDS OF VARIOUS RADII

In a free vortex the angular velocity varies as the inverse square of the radius and the tangential velocity varies inversely as the radius. However, in a cyclone vortex, involving a viscous gas, the increase of angular velocity is reduced by the internal friction. Experimental measurements have

shown that in such a case: $U^2 R = \text{const.}^2$

or $\boxed{U_n^2 R = U_i^2 R_i} \text{ (FORCED VORTEX)}$

$r_2 \equiv$ effective exit radius
(CENTRAL CORE)

$r_E \equiv$ exit radius

$$\frac{r_E}{2} \equiv r_2$$

(SPINNING SPEED AT VARIOUS R)

CENTRIFUGAL FORCE ACTING ON THE DROPLET:

$$\boxed{F_c = \frac{m_0 U_n^2}{R} \approx \frac{\rho_0 d_0^3 \omega^2}{R}}$$

In addition, the drag in the Stoke's law region of flow ($d_0 < 100 \mu$)

can be represented by: $\boxed{F_D = K d_0 \mu_s U_n}$

where K is the empirical

design factor = .01 for an 8" diameter cyclone; μ_s ^{absolute} kinematic viscosity of the

$$\text{gas. } \eta = \frac{F_c}{F_D} \approx \frac{U_n \rho_0 d_0^2}{R \mu_s}$$

As a first approximation, the collection

$$\mu_s = 3 \times 10^{-7} \frac{\text{lb-sec}}{\text{ft}^2}$$

efficiency is found to increase with an increase in inlet or tangential velocity, and droplet diameter; it is found to decrease with cyclone diameter. Typical inlet velocities for an 8" diameter cyclone separator are in the order of 50 to 60 $\frac{\text{ft}}{\text{s}}$ (15 to 20 m/s).⁷ However, the problem of high velocities, as mentioned previously, is the creation of the unstable boundary layer and turbulence, i.e., the problem of droplet re-entrainment. Thus, there is an optimum velocity above which separator "breakdown" occurs. This optimum velocity will satisfy both the droplet collection efficiency and minimize turbulence.

What is the optimum inlet velocity for $\dot{M}_T^* = 2.2 \frac{\text{lbm}}{\text{s}}$ $P = 30 \text{ psig}$, $X_{s1} = .25$, $d_1 = 2''$?

Saturated steam tables at 30 psig: $\begin{cases} V_F = .01721 \frac{\text{ft}^3}{\text{lb}_m} \\ V_g = 9.401 \frac{\text{ft}^3}{\text{lb}_m} \end{cases}$

$$\frac{1}{\rho_{\text{mix}}} = V_F + X(V_g - V_F) \Rightarrow \frac{1}{\rho_{\text{mix}}} = .01721 + .25(9.401 - .01721)$$

$$\rho_{\text{mix}} = .423 \frac{\text{lb}_m}{\text{ft}^3} \Rightarrow (2.2 \frac{\text{lb}_m}{\text{s}} \times .25) = (.423 \frac{\text{lb}_m}{\text{ft}^3}) V \frac{\text{ft}}{\text{s}} \left(\frac{\pi (\frac{2}{12})^2}{4} \right) \text{ft}^2 \Rightarrow \boxed{V_i = 59.6 \frac{\text{ft}}{\text{s}}}$$

similar calculations for 40 and 50 psig yield 61.78 and 55.72 $\frac{\text{ft}}{\text{s}}$, respectively.

In the separator testing, one problem which was not dealt with is the problem of brine content and hydrogen sulfide (H_2S) in geothermal flow. In the case of "real" geothermal flow, the separated steam would have to pass some chemical process in order for the H_2S to be neutralized. No wire mesh must be placed in the separator for mist elimination. The reason being that the interference with the path of the vortex and/or increasing residence time of steam has an adverse effect on separator performance.

WIRE MESH MIST
ELIMINATORCONCLUSION

(NOT BENEFICIAL)

The results of the separator testing indicate that the hybrid staging will perform well under $10\frac{1}{20}$ psig. The pressure of interest, $132/1.7 \frac{\text{lbs}}{\text{s}}$.

There are two factors that effect separator performance, mainly flow rate and pressure. The larger the separator, the more flow it can handle. However, a preliminary calculation shows that for a fixed flow velocity, U ,

performance is independent of cyclone size (R):

$$\frac{dv_D}{dt} = a_D = \underbrace{\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial r}}_{\substack{\text{DROPLET} \\ \text{ACCELER.}}} + \underbrace{C_D(\dot{u} - \dot{v}_D)}_{\text{FRICTION TERM}}$$

$$F = \frac{m u^2}{R} = m a_D \Rightarrow a_D \sim \frac{1}{R} \quad (\text{FOR FIXED } U)$$

BUT AT THE SAME TIME: $\tau (\text{TIME}) \sim R$

$$\Delta R \sim a \tau^2 = \left(\frac{1}{R}\right)(R^2) = R \Rightarrow \frac{\Delta R}{R} = \text{CONST. (INDEP. OF } R \text{ FOR FIXED } U)$$

BUT FOR VARYING U , $\frac{U}{R}$ DETERMINES CYCLONE EFFICIENCY.

Consequently, for fixed d , (and constant flow rate), efficiency is invariant to size. For increasing v (i.e., smaller inlet diameter), efficiency increases for a given cyclone diameter.

FUTURE INVESTIGATIONS

This investigation did not concern itself with the manner in which water droplets moved in the cyclone and their re-entrainment into the steam phase.

That is why the author had no experimental support for the droplet collection efficiency theory, and the growth of the unstable boundary layer. Future study should concern itself with the coloring of hot water and a transparent separator in order to examine the behavior of water droplets. Once this is observed, design modification of separator and/or change of operating conditions will show its effect on droplet behavior.

Separator characteristics, largely, depend upon the designed control system and the exact location of pressure gauges. One can assume the total flow and separator pressure analogous to current and voltage, respectively. In the separator, the current divides into steam and water currents, each passing through their respective resistances. The resistances act like the impedances across the steam and water discharge valves. V_s and V_w are pressures downstream of the discharge valves. The effect of these parameters was observed in testing the separator. As the simple relation indicates, increasing or decreasing V cannot by itself determine the variation of I_s/I_w (i.e. steam quality). In the experiment, the reason for better separation observed at higher pressures was merely a manner of coincidence of the designed control valve system. Another design would result to a different separator performance.

Thus, it is necessary to investigate the relation between discharge coefficients and downstream pressures of steam and water. By doing this, the circuit analogy expression will diverge into a more informative relation in predicting steam quality for a given discharge system.

APPENDIX A

DEFINITIONS

TWO-PHASE QUALITY: This quantity has been defined as the ratio of the steam

flow rate, out, to total mass flow rate, $X = \frac{\dot{m}_{s0}}{\dot{m}_t}$.

If the two-phase flow is a homogeneous fluid (no slip in velocity of the two-phases) the above definition is accurate; if the two velocities are not equal, the quality will be the ratio of the vapor and total mass in the control volume of interest. $X' = \frac{m_s}{m_t}$. At comparatively low values of steam quality

($X < .1$), for the range of pressures normally in use, a two-phase mixture will no longer flow as a homogeneous fluid. Equilibrium in temperature and pressure is assumed.

BREAKDOWN: Separator breakdown is defined as the maximum flow rate under a given pressure for which separated steam shall be 99% or better dry. The breakdown point is the beginning of increase in separated steam wetness.

EFFICIENCY: Separator efficiency is defined as the overall performance of the separator in producing high quality, high flow rate steam at low pressure. The former is given by the collection efficiency defined previously as the ratio of the centrifugal force to Stokes viscous force, $\eta_{coll} \approx \frac{u P_0 d_0^2}{12 \mu_s}$.

The latter, high flow rate, is a function of inlet velocity and diameter, but also water exit diameter, for one of the limiting factors of higher flow rates is the ability of the separator to physically handle the water and not overflow. The advantage of a larger cyclone diameter is discredited

when the water cannot be discharged and the level maintained at its optimum point. As for separator pressure, the vessel must initially meet A.S.M.E. boiler and pressure vessel specifications for withstanding high internal pressures and temperatures. As mentioned before, and observed experimentally, the higher the pressure, efficiency becomes higher. Of course, the range of operating pressure must be known in order to find the lowest possible pressure to work with.

APPENDIX B

PRECISION DETERMINATIONS:

The equation used to determine the steam quality, X , has been

$$X = \frac{H - H_F}{H_{FG}} \times 100\%$$

H was found by the throttling temperature and pressure which in turn are functions of: $H = f(T_{th}, P_{th}, \frac{D}{d}, \frac{L}{h}, \dot{m})$. H_F and H_{FG} were found by

looking up the pressure saturation steam tables (P_{SEP}).

Using the estimates of the precision of measurements for each variable, an overall tolerance on H , H_F and H_{FG} is $\pm .2\%$.

It is shown in the discussion of the theory of errors, that the following relation is true regardless of the probability distributions of the component errors where the component errors are errors in the measurements of the independent variables, $x_1, x_2, x_3, \dots, x_n$:

$$u = u(x_1, x_2, \dots, x_n) \Rightarrow$$

$$\sigma_u = \left[\sum_{i=1}^n \left(\frac{\partial u}{\partial x_i} \right)^2 \sigma_{x_i}^2 \right]^{\frac{1}{2}}$$

where $\sigma_{x_i} \equiv$ standard deviation of x_i . Applying this equation to the definition

of X :

$$\frac{\partial X}{\partial H} = \frac{1}{H_{FG}}, \quad \frac{\partial X}{\partial H_F} = \frac{-1}{H_{FG}}, \quad \frac{\partial X}{\partial H_{FG}} = \frac{H_F - H}{H_{FG}^2}$$

$$\sigma_X = \left[\left(\frac{\sigma_H}{H_{FG}} \right)^2 + \left(\frac{-\sigma_{H_F}}{H_{FG}} \right)^2 + \left(\frac{H_F - H}{H_{FG}^2} \sigma_{H_{FG}} \right)^2 \right]^{\frac{1}{2}}$$

THEN:

$$X = X \pm \sigma_X$$

e.g.

$$\text{RUN 95: } H_F = 218.8 \frac{\text{Btu}}{\text{lb}_m}, \quad H_{FG} = 945.3, \quad H = 1153.9 \left\{ X = 98.9\% \right.$$

$$\sigma_{H_F} = 1.4, \quad \sigma_{H_{FG}} = 1.89, \quad \sigma_H = 2.31$$

$$\Rightarrow \sigma_X = 0.003 \text{ or } 0.3\%$$

$$X = 98.9 \pm 0.3\%$$

APPENDIX C

Comparison of throttling temperatures obtained by replacing the 3/32 nozzle with a bigger size (5/32) in the calorimeter.

3/32	
P (PSIG)	T °F
52	238
40	238
30	233
20	220
15	217

5/32	
T °F	P (PSIG)
255	50
260	40
254	30
240	20

APPENDIX D

Checking the size of the separator. The design formula previously mentioned was:

$$V_{MAX} = .35 \left(\frac{P_w - P_s}{P_s} \right)^{\frac{1}{2}}$$

e.g. : $\dot{m}_T = 2.2 \frac{\text{lbm}}{\text{s}}$, $X_{Si} = .25$, $P_{SEP} = 30 \text{ psig}$.

$$\dot{m}_S = 2.2 \times .25 = .55 \frac{\text{lbm}}{\text{s}} \quad , \quad \frac{1}{\rho_{mix}} = V_{mix} = V_F + X(V_G - V_F)$$

$$2.364 = .01721 + .25(9.401 - .01721)$$

$$\frac{P_w}{P_s} = \frac{V_s}{V_w} = \frac{9.401}{.01721} = 546.25$$

$$\Rightarrow V_{MAX} = .35 \left(546.25 - 1 \right)^{\frac{1}{2}} = \underline{8.16 \frac{\text{ft}}{\text{s}}}$$

$$\frac{\pi D_{min}^2}{4} = \frac{.55 / .423}{8.16} \Rightarrow D_{min} = 5.4'' \text{ OR } \boxed{D_{min} = 6''} \text{ MINIMUM CYCLONE DIAMETER TO HANDLE } .55 \frac{\text{lbm}}{\text{s}} \text{ OF STEAM}$$

RANGE OF INLET VELOCITY: $60\sqrt{.423} < V_i < \frac{100}{\sqrt{.423}}$

$$\boxed{39.0 < V_i < 153.75 \frac{\text{ft}}{\text{s}}}$$

TO CALCULATE CYCLONE HEIGHT:

Assume time for vessel to fill up with water is ~ 4 minutes (including time for instrumentation and labor to realize)⁵

THEN: $Q_w = \frac{2.2 \times (1 - .25)}{.423} = 3.9 \frac{\text{ft}^3}{\text{s}} \Rightarrow \text{VOL.} = (3.9 \times \frac{4}{60}) \text{ ft}^3$

$$\frac{\pi D^2}{4} L = (3.9 \frac{\text{ft}^3}{\text{s}}) (\frac{1}{15} \text{ s}) \Rightarrow \boxed{L = 1.32 \text{ ft}} \Rightarrow \frac{L}{D} = \frac{1.32 \times 12}{6}$$

$$\boxed{\frac{L}{D} = 2.64}$$

$$\Rightarrow L \text{ (HEIGHT OF VESSEL EXCLUDING FLANGES)} = 1.32 \text{ FT.}$$

$$\frac{L}{D} = 2.64, \quad D_{\min} = 6''$$

$$39.0 < V_i < 153.75 \Rightarrow 2.47'' < d_i < 1.24''$$

These results are in good proximity with the real dimensions of the tested separator which are:

$$\begin{cases} L = 1.72 \text{ FT} \\ D = 8'' \end{cases} \Rightarrow \boxed{\frac{L}{D} = 2.58}$$

$$L' \text{ (HEIGHT OF STEAM TUBE)} = 1.29 \text{ FT.}$$

$$\boxed{V_i = 59.6 \frac{\text{FT}}{\text{S}}}$$

$$\boxed{d_i = 2''}$$

recommended dimensions in terms of inlet diameter, d ,⁵

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WATER FLOW/CALIBRATION CURVE OF THE FLOWMETER

$$Y = 19.975X - 3.122$$

GOBBERS ENGINEERS, INC.
BETHESDA, MARYLAND 20834

DRAWN V.A.M.

ISSUED

SIZE: **A**
55353

SCALE:

WT.

DWG NO.

SHEET OF

REV

SEPARATOR CLOSE-UP

- STEAM-OUTLET
- THROTTLING CALORAMETER
- CONDENSER TEMP. & PRES.

FLOW-METER

PUMP-MOTOR

OVER-ALL VIEW
OF TESTING FACILITY

DATA SHEET

VALIDITY OF

DATE 2/1/50

TEST OF CHECKING THE DATA
OF UNCONTROLLED WATER
LEVEL

PROJECT

STAGE 36

[illegible]

READING BY A.V.M.
CALCULATIONS A.V.M.

DATA SHEET

DATE _____

TEST OF RESULTS OF CALCULATED
STEAM QUALITY

PROJECT _____

STAGE 37

POINT RUN	OF P ₀		PSIA $\frac{Btu}{lb}$		→ %		%							
	T+K T _H	P ₀	mit	P _{SEP}	H	H _F	H _{FG}	X _{sin}	X _{so}					
81	219.2	15.0	4.05	44.7	1153.9	242.9	928.9	23	98.1					
82	271.3		2.91	74.7	1179.	277.1	904.7	22	99.7					
83	235.9		3.44	64.7	1162.1	267.2	911.8	26	98.1					
84	221.35		3.4	54.7	1154.96	255.9	919.9	26	97.7					
85	218.2		3.5	44.7	1153.42	242.9	928.9	28	98.0					
86	270.2		2.71	74.7	1178.54	277.1	904.7	26	99.6					
87	262.9		2.84	64.7	1175.1	267.2	911.8	26	99.6					
88	254.6		2.74	54.7	1171.1	255.9	919.9	24	99.5					
89	219.3	✓	2.93	44.7	1153.9	242.9	928.9	24	98.1					
90	271.3	15.47	2.27	74.7	1178.9	277.4	904.5	24	99.7					
91	265	15.42	2.19	64.7	1175.9	267.5	911.6	26	99.6					
92	254.6	15.33	2.22	54.7	1171.1	256.3	919.6	25	99.5					
93	248.3	15.22	2.19	44.7	1168	243.4	928.6	25	99.6					
94	231.7	15.1	2.16	34.7	1160.	227.9	939.2	25	99.2					
95	219.3	15.1	2.15	29.7	1153.9	218.8	945.3	25	98.9					

READING BY A.V.M.
CALCULATIONS A.V.M.

DATA SHEET

DATE 2/1/80

TEST OF RESULTS OF
CALCULATED QUALITY

PROJECT

STAGE 38

POINT RUN	OF P _{SLA}		P _{SLA} <u>BLK</u> <u>1150</u>		→ %		%							
	T+K T _H	P ₀	M ₇	P _{SLP}	H	H _F	H _{FG}	X _{sin}	X _{SO}					
64	231.7	14.95	1.12	34.7	1161.0	227.4	939.5	26	99.8					
65	219.2		1.15	24.7	1153.9	207.7	952.6	26	99.3					
66	215.1		1.15	21.7	1151.85	200.6	957.3	25	99.4					
67	217.3		1.58	74.7	1172.5	277.14	904.7	24	99.7					
68	265.0		1.58	64.7	1176.0	267.2	911.8	18	99.7					
69	255.6		1.60	54.7	1171.6	255.9	919.9	24	99.5					
70	245.2	V	1.57	44.7	1166.6	242.9	928.9	23	99.4					
71	231.7	15.0	1.57	34.7	1160.0	227.4	939.5	23	99.3					
72	219.2		1.54	24.7	1153.9	207.7	952.6	24	99.3					
73	265.0		3.49	74.7	1176.1	277.14	904.7	24	99.4					
74	215.13		3.7	64.7	1151.9	267.2	911.8	27	97.0					
75	240.17		4.2	54.7	1164.2	255.9	928.9	24	97.8					
76	258.8		4.0	64.7	1173.1	267.2	911.8	20	99.4					
77	215.13		4.02	64.7	1151.8	267.2	911.8	23	97.0					
78	217.2		2.96	64.7	1152.9	267.2	911.8	32	97.1					
79	215.13		2.89	54.7	1151.9	255.9	919.9	31	97.4					
80	227.5	V	4.38	54.7	1158	255.9	919.9	20	98.1					

READING BY A.V.M.
CALCULATIONS A.V.A.

DATA SHEET

DATE 1/30/80

TEST OF RESULTS OF
CALCULATED STEAM QUALITY

PROJECT

STAGE 39

POINT RUN	°F	psia	psia	$\frac{H_{2O}}{lbm}$				%	%					
	T _{th} K	P _o	m _T	P _{SEP}	H	H _F	H _{Fg}	X _{sin}	X ₅₀					
47	223.4	15.00	1.12	41.7	1155.96	238.6	931.9	50%	99.3					
48	216.17		1.09	42.7	1152.38	240.1	930.9	53	98.4					
49	267.13		1.05	67.7	1177.07	270.3	909.6	50	99.7					
50	267.13		1.07	67.7	1177.07	270.3	909.6	49	99.7					
51	262.95		1.09	66.7	1175.0	269.2	910.3	48	99.5					
52	251.5		1.07	67.7	1169.62	270.3	909.6	49	99.9					
53	240	V	1.17	42.7	1164.1	240.07	930.9	44	98.0					
54	271.3	14.95	1.79	74.7	1179.0	277.14	904.7	27	99.7					
55	262.9		1.78	64.7	1175.1	267.18	911.8	27	99.6					
56	254.6		1.78	54.7	1171.1	255.9	919.9	25	99.5					
57	244.2		1.82	44.7	1166.1	242.9	928.9	24	99.4					
58	229.6		1.82	34.7	1159.0	227.1	939.5	26	99.2					
59	219.2		1.82	27.7	1153.9	214.2	948.3	27	99.1					
60	271.3		1.11	74.7	1179.0	272.14	904.7	25	99.7					
61	262.9		1.1	64.7	1175.1	262.2	911.8	27	99.6					
62	254.6		1.1	54.7	1171.1	255.9	919.9	26	99.5					
63	244.2	V	1.08	44.7	1166.1	242.9	928.9	27	99.4					

READING BY A.V.M.
CALCULATIONS A.V.M.

DATA SHEET

DATE _____

TEST OF: 8" DIA. TWO PHASE FLOW
SEPARATOR.

PROJECT. _____

STAGE. _____

40

POINT RUN	psig		psig		psig		psig		lb/sec		K	$T_{th}+K$	T_{oo}	X_{sin}	X_{so}
	$P_{s in}$	$P_{w in}$	m_{win}	P_{sep}	T_{th}^{OF}	P_{th}	m_{wo}	T_c	m_{so}						
$m_T = .52$ 12/18/79	1	106	71	21.2	25	211	14.74	0.04		0.48				92%	
$m_T = .36$	2	108	70	17.5	37	225		0.16		0.20				55%	
$m_T = 3.225$	3	106	180	23	25	218		3.1		0.125				3.8%	
$m_T = 5.58$	4	102	180	23	35	228		5.08		0.50				8.9%	
$m_T = 4.42$	5	95	175	22	40	210		3.5		0.92				20.8%	
$m_T = 4.08$ 12/21/79	6	92	165	25	20	210	↓	3.33		0.75				18%	
1.99	7	108	78	7.5	40	240	14.77	0.52	116	0.47				47%	
1.01	8	105	75	7.5	30	240	14.77	0.54	↓	0.47				46%	
1.04	9	106	75	7.5	20	226	14.77	0.55	↓	0.49				47%	
1.09	10	107	75	7.5	10	214.5 AVE.	14.74	0.55	↓	0.54				49%	
1.44	11	103	75	12	40	246	14.83	0.77	130	0.67				46%	
1.43	12	103	75	12	30	236	14.79	0.81	↓	0.62				43%	
1.45	13	105	75	12	20	210	14.74	0.80	↓	0.65				45%	
1.48	14	103	75	12	17	210	14.74	0.82	↓	0.66				45%	
2.19	15	96	67.5	17	40	212	14.81	1.19	120	1				45%	
2.12	16	96	182	17	50	212	14.87	1.16	↓	0.96				45%	

READING BY A.V.M

CALCULATION " A.V.M.

DATA SHEET

$P_{\text{barom.}} = 29.7$

$T_{\infty} = 78^{\circ}\text{F}$
 79°F

DATE _____

PROJECT _____

STAGE 41

TEST OF _____

POINT RUN	P_{sin}	P_{win}	$\dot{m}_{\text{win}}$	P_{sep}	T_{Th}	P_{Th}	$\dot{m}_{\text{wo}}$	$\dot{m}_{\text{so}}$	T_{c}	K	$T_{\text{Th}+K}$	T_{co}	X_{sin}	X_{so}
$\dot{m}_T = 1.18$ 17	109	205	.8	43	250	$\frac{14.7}{14.7}$.7	.92	.26	98	4.6	254.6	78	22%	↑
$\dot{m}_T = 1.2$ 18	109	205	.8	30	225	+5	.72	.28		3.6	228.6		23%	↑
$\dot{m}_T = 1.22$ 19	109	205	.8	20	222	+5	.92	.30		3.5	225.5		24%	↑
$\dot{m}_T = 1.33$ 20	109	205	.8	10	210	+4	1.0	.33	↓	3.0	213.		24%	BAI
$\dot{m}_T = 1.78$ 21	104	205	1.1	40	242	+6	1.33	.45	120	4.28	246.3		25%	↑
$\dot{m}_T = 1.56$ 22	105	202	1.1	30	218	+5	1.1	.46		3.35	221.35		29%	↑
$\dot{m}_T = 1.63$ 23	105	205	1.1	20	210	+4	1.1	.53	↓	3.0	213.		32%	↑
$\dot{m}_T = 2.26$ 24	102	202	1.45	40	210	6	1.8	.46	118	3.0	213.		20%	↑
$\dot{m}_T = 2.27$ $1/16/80$ 25	102	205	1.45	50	240	+7	1.8	.47	↓	4.2	244.2	↓	20%	↑
$\dot{m}_T = 1.67$ 26	60	180	.8	50	212	$\frac{14.7}{14.7}$ +.5	.79	.88	109	3.13	215.13	78	52%	↑
$\dot{m}_T = 1.47$ 27	62	175	.64	60	213	+4	.58	.89	109	3.17	215.17	79	60%	↑
$\dot{m}_T = 1.43$ 28	84	180	.66	60	213	+4	.74	.69	100	3.17	215.17	79	48%	↑
$\dot{m}_T = 1.49$ 29	80	175	.66	70	212	+3	.77	.72	100	3.13	215.13	79	48%	↑
$\dot{m}_T = 1.48$ 30	82	180	.66	40	213	+4	.8	.68	100	3.14	216.14	79	46%	↑
$\dot{m}_T = 1.78$ 31	67	180	.915	60	251	+4	1.1	.68	102	4.6	255.6	79	38%	↑
$\dot{m}_T = 1.68$ 32	67	177	.85	60	254	+4	.97	.71	102	4.75	258.7	79	42%	↑

READING BY _____

CALCULATIONS _____

$$y = 45y + 68.5$$

DATA SHEET

DATE _____

TEST OF _____

PROJECT _____

STAGE 42

[illegible]

READING BY _____

CALCULATIONS

DATA SHEET

$P_{h_{cor}} = 297'$

DATE 1/23/80

TEST OF WATER LEVEL IN SEPARATOR(h)
AND ITS EFFECT ON SEPARATOR
PERFORMANCE

PROJECT STIEDEL
 STAGE 43

POINT RUN		h	P _{sin}	P _{win}	m _{win}	P _{sep}	T _{Th}	m _{wo}	m _{so}	T _C	K	T _{Th+K}	T _∞	X _{sin}	X _{so}
1.61*	45	4±.3	105	185	6	26	235	1.07	.54	84	4.0	239	78	33%	
1.56*	46	6±.5	105	185	6	26	232	1.02	.54	84	3.88	235.88		34%	
1.12	47	7±.5	107	185	6	27	220	.56	.56	84	3.42	223.4		50%	98.4
1.09	48	8±.5	108	185	5.6	28	213	.51	.58	86	3.17	216.17		53%	98.1
1.05	49	4±.5	108	185	4.6	53	262	.52	.53	84	5.13	267.13		50%	99.7
1.07	50	6±.5	108	185	4.6	53	262	.54	.53	84	5.13	267.13		49%	99.7
1.09	51	7±.5	107	185	4.6	52	258	.56	.53	85	4.95	262.95		48%	99.5
1.07	52	3±.5	108	185	4.6	53	247	.54	.53	85	4.48	251.48		49%	98.9
1.17	53	4±.5	109	185	5.8	28	236	.66	.51	84	4.0	240.	V	44%	99.3

*: BAD DATA; CORRECTION: RUN 53

READING BY _____

WATER LEVEL MUST BE: $4 < h < 6$

CALCULATIONS

$p_b = 29.80$

DATA SHEET

4 < h < 6

DATE 1/25/80

TEST OF PERFORMANCE OF SEPARATOR
FOR $m_2 = [1.5, 1.0, 1.5] \frac{10}{5}$, QUALITY inn =

PROJECT

STAGE 44

psig.															
POINT RUN	P_{sin}	P_{win}	m_{min}	P_{sep}	T_{Th}	K	T_{Th+K}	m_{wo}	m_{so}	T_C	T_{∞}	X_{sin}	$X_{so}^{\%}$		
1.79	54	97	196	1.05	60	2.66	5.3	271.3	1.3	.48	108	78	27%	99.7	
1.78	55	98	198	1.05	50	2.58	4.95	262.95	1.3	.48	108		27%	99.6	
1.78	56	100	199	1.05	40	2.50	4.6	254.6	1.33	.45	108		25%	99.5	
1.82	57	102	200	1.05	30	2.40	4.2	244.2	1.38	.44	106		24%	99.4	
1.82 LOWEST $P_{sep} * 1.82$	58	102	200	1.05	20	2.26	3.6	229.6	1.35	.47	108	V	26%	99.2	
	59	102	202	1.05	(13)*	2.16	3.2	219.2	1.33	.49	108		27%	99.1	
1.11	60	106	200	.66	60	2.66	5.3	271.3	.83	.28	88		25%	99.7	
1.10	61	106	200	.66	50	2.58	4.95	262.95	.80	.30	88		27%	99.6	
1.10	62	106	201	.66	40	2.50	4.6	254.6	.81	.29	88		26%	99.5	
1.08	63	106	200	.66	30	2.40	4.2	244.2	.79	.29	88		27%	99.4	
1.12	64	106	198	.66	20	2.28	3.7	231.7	.83	.29	88		26%	99.8	
1.15	65	107	201	.66	10	2.16	3.2	219.2	.85	.30	88		26%	99.3	
1.15*	66	108	199	.66	(7)*	2.12	3.1	215.1	.86	.29	88		25%	99.4	
.58	67	110	207	.36	60	2.67	5.3	272.3	.44	.14	94		24%	99.7	
.58	68	110	206	.36	50	2.60	5.0	265	.47	.11	94	V	18%	99.7	
.60	69	111	207	.36	40	2.51	4.65	255.6	.46	.14	97		24%	99.5	
.57	70	111	205	.36	30	2.41	4.2	245.2	.44	.13	98		23%	99.4	

* LOWEST POSSIBLE P_{sep} READING BY
CALCULATIONS

$$P_{bar} = 29.68$$

DATA SHEET

$$P_0 = 14.7 + \frac{1}{2} = 15.1$$

DATE 1/28/80

TEST OF SEPARATOR PERFORMANCE

PROJECT

STAGE 45

POINT RUN	P_{sin}	P_{win}	m_{win}	P_{sep}	T_{th}	K	$T_{th}+K$	m_{wo}	m_{so}	T_c	T_{oo}	X_{sin}	X_{so}
.57	71	111	206	.36	20	228	3.7	231.7	.44	.13	98	78	23% 99.3
.54	72	111	205	.36	10	216	3.2	219.2	.41	.13	100	↓	24% 99.3
3.49	73	80	185	-	74.7	260	5.0	265.0	2.63	.86	120	78	24% 99.4
3.7	74	79	180	-	64.7	212	3.13	215.13	2.7	1.0	122		27% 97.9 BAD
4.2	75	78	176	-	54.7	237	3.17	240.17	3.2	1.0	120	78	24% 97.8
4.0	76	79	185	-	64.7	254	4.78	258.78	3.2	.8	117		20% 99.4
1/30/80 402	77	82	190		64.7	212	3.13	215.13	3.08	.94	117		23% 97.0 BAD
2.96	78	82	190		64.7	214	3.2	217.2	2.0	.96	117		32% 97.1
2.89	79	87	187		54.7	212	3.13	215.13	1.97	.92	110		31% 97.4
4.38	80	75	180		54.7	224	3.57	227.5	3.45	.93	110		20% 98.1
4.05	81	80	182	*	44.7	216	3.2	219.2	3.12	.93	108		23% 98.1
2.91	82	87	165		74.7	266	5.3	271.3	2.27	.64			22% 99.7
3.44	83	85	185		64.7	232	3.88	235.9	2.55	.89	105		26% 98.1
3.4	84	84	175		54.7	218	3.35	221.35	2.5	.90	105	✓	26% 97.7
3.5	85	84	181	*	44.7	215	3.24	218.24	2.5	1.0	105		28% 98.0
2.71	86	91	185		74.7	265	5.26	270.2	2.0	.71	97		26% 99.6
2.84	87	92	185		64.7	258	4.95	262.9	2.08	.76	97		26% 99.6

*: LOWER P NOT POSSIBLE.

READING BY A.M.
CALCULATIONS A.M.

DATA SHEET

$$P_{pwi} = 29.95 \quad P_o = 14.7 + \frac{.5}{2}$$

TEST OF _____

DATE _____

PROJECT _____

STAGE 46

Biot														
POINT RUN	P_{sin}	P_{wh}	m_{win}	P_{sep}	T_{Th}	K	$T_{Th}+K$	m_{wh}	m_{so}	T_c	T_{∞}	X_{sin}	X_{so}	h
88	94	185	-	54.7	250	4.6	254.6	2.08	.66	94	78	24%	99.5	3
89	93	185		*44.7	216	3.27	219.27	2.22	.71	96		24%	98.1	4
900	103	196	24.5	74.7	$P_{Th}=1^{\circ}$ 266	5.3	271.3	1.72	.55	108		24%		3.9
901	103	190	24.5	64.7	$P_{Th}=9$ 260	5.04	265.0	1.63	.56	108		26%		4.0
92	105	196	24.5	54.7	$P_{Th}=7$ 250	4.6	254.6	1.66	.56	108		25%		4.0
93	110	196	24.5	44.7	$P_{Th}=5$ 244	4.36	248.3	1.64	.55	108		25%		3.8
94	110	196	24.5	34.7	$P_{Th}=25$ 228	3.7	231.7	1.61	.55	108		25%		3.8
95	109	200	24.5	*29.7	$P_{Th}=25$ 216	3.27	219.3	1.61	.54	108		25%		3.9
96														
97														
98														
99														
100														

* lowest possible P_{sep}

READING BY _____
 CALCULATIONS _____