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**PATHFINDER ATOMIC POWER PLANT
STEAM SEPARATOR DEVELOPMENT**

MASTER

by

G. C. Kutsch, D. H. Swanson and H. W. Yant

June 15, 1962

Submitted to

**U. S. ATOMIC ENERGY COMMISSION
NORTHERN STATES POWER COMPANY**

and

CENTRAL UTILITIES ATOMIC POWER ASSOCIATES

by

**ALLIS-CHALMERS MANUFACTURING COMPANY
ATOMIC ENERGY DIVISION
Milwaukee 1, Wisconsin**

Ref: AEC Contract No. AT(11-1)-589

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ALLIS-CHALMERS MANUFACTURING COMPANY

Under

Agreement dated 2nd Day of May 1957, as Amended
betweenAllis-Chalmers Mfg. Co. & Northern States Power Co.
under

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PATHFINDER ATOMIC POWER PLANT

STEAM SEPARATOR

DEVELOPMENT

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FOREWORD

One of a series of reports on research and development in connection with the design of Pathfinder Atomic Power Plant, this particular report deals with Steam Separation Tests.

The report describes the conception, design and full-scale air-water testing of separators that remove steam from the recirculation water in the Pathfinder reactor.

The Pathfinder plant will be located at a site near Sioux Falls, South Dakota and is scheduled for operation in the latter part of 1962. Owners and operators of the plant will be Northern States Power Company of Minneapolis, Minnesota. Allis-Chalmers is performing the research, development and design, and is responsible for plant construction.

The U.S. Atomic Energy Commission, through Contract No. AT(11-1)-589 with Northern States Power Company, and Central Utilities Atomic Power Associates (CUAPA) are sponsors of the research and development program.

The plant's reactor will be of the Controlled Recirculation Boiling Water type with integral Nuclear Superheater.

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SUMMARY

The objective of this project was to develop a steam separator for the Pathfinder reactor, to conduct a full-scale operational test of the selected steam separator concept and to evaluate the results.

The first step was to review conventional steam separators and compare characteristics with Pathfinder requirements. The conventional centrifugal, crossover and upcomer type, separators were found to have high pressure drops at Pathfinder flow rates. Some of them also caused re-entrainment of steam at the water outlets. Testing of a downcomer type centrifugal separator - in scale model form - showed promise of high efficiency and low pressure drop. The bulk of the test program was, therefore, concentrated on this type of separator.

A full-scale separator model was developed through the combination of development testing on scale models and the application of existing "theory of centrifugal separation" principles. Design limits for the separator were set according to the requirements of the Pathfinder reactor. This model was put through full-scale air-water tests at the Allis-Chalmers Greendale Laboratories. Modifications were made from time to time until the full-scale model met Pathfinder design requirements.

Conclusions

The final separator model selected as a result of these tests has a cylindrical body 10 in. in diameter and 8-1/2 ft long. The steam-water mixture enters the cylinder through a tangential inlet measuring 1.8 in. wide and 4-1/2 ft long. A conical outlet nozzle contains the vortex formed...and a 4 in. diameter pipe exhausts steam to the reactor steam dome. (Illustration "A" in this Summary Section shows the full size

model operating in the air-water test loop). Forty-five of these separators are planned for installation in the downcomer region of the Pathfinder reactor.

The design limits for the Pathfinder separators were 1 per cent by volume carryunder (void-to-mixture flow ratio at separator water outlet), and 3-1/2 ft maximum head loss. The split of upcomer voids between downcomer and free surface was to be determined.

These design limits were met in the following manner:

The measured carryunder was 0.1 per cent by volume, head loss was 3.7 ft at the reactor design flow rates of 1330 gpm/separator and 175 cfm (free air) supplied to the loop, 25 per cent to 40 per cent of upcomer voids carried into the separator at rated conditions.

Illustration "A"...Full size steam separator model operating in the air-water test loop at 1000 gpm water, 45 cfm air. Top view - looking down the standpipe. (A-C B-39030)

1. INTRODUCTION

The Controlled Recirculation Boiling Reactor (CRBR) for Pathfinder Atomic Power Plant is a medium sized reactor with gross thermal power of 199.6 mw. Reactor operating conditions are 600 psig and 489F. Conditions at the superheater outlet are 535 psig and 825F. The gross electrical capability of the power plant itself is 66 mw. A list of reactor and plant design characteristics is shown in Table 1. A cross section of this reactor with its nuclear superheater is illustrated in Fig. 1.

The outstanding features of the Pathfinder CRBR are controlled, forced recirculation of coolant-moderator, and integral nuclear superheat. It is the forced recirculation aspect which gives rise to the steam separation problem studied under this test program.

Under normal operating conditions, the recirculation rate is 60,000 gpm. This flow is provided by three recirculation pumps rated at 20,000 gpm each - with the pumps located external to the reactor vessel. At this recirculation rate, downcomer water velocity was more than 2 fps. The high carryunder of steam that would occur presented these problems:

(a) lowering the Net Positive Suction Head (NPSH) on the recirculation pumps, and (b) the causing of instabilities in the boiling core because of steam entering the fuel element inlets. Although the high downcomer velocities might have been lowered by the use of a larger diameter vessel, the increased vessel cost would have been more than the cost for internal steam separation.

TABLE I

DESIGN DATA

PATHFINDER REACTOR

Plant

Power, boiler region.....157,200 kw
Power, superheater region..... 42,400 kw
Steam flow at rated power.....616,125 lbs/hr
Total core power.....199,600 kw
Gross electrical capability..... 66,000 kw
Net electrical output..... 62,000 kw
Net efficiency.....31.0 per cent
Steam outlet pressure (reactor)..... 535 psig
Reactor operating pressure..... 600 psig
Temperature, boiler region..... 489 F
Outlet temperature, superheater region..... 825 F
Gross heat rate.....10,199 Btu/kw-hr
Reactor building size.....50 ft dia x 120 ft

Reactor

Vessel size (overall).....11 ft 6 in o.d. 36 ft l in
Total core dimensions..... 6 ft x 6 ft
Dimensions of superheater region.....6 ft x 30 in
Fuel, boiler (Zr-2 clad).....approx. 2.2 per cent enriched UO₂
Fuel, superheater (S.S.clad)....approx. 93 per cent enriched UO₂
Fuel, loading, boiler (U-235).....145.6 kg
Fuel, loading, superheater (U-235)..... 42 kg
Power Density (boiler core coolant).....87 kw/liter
Average heat flux, boiler region.....128,000 Btu/hr-ft²
Average heat flux, superheater region..... 77,800 Btu/hr-ft²
Maximum heat flux, boiler region.....462,000 Btu/hr-ft²
Maximum heat flux, superheater region.....245,000 Btu/hr-ft²
Recirculation rate.....60,000 gpm
Recirculation pump power 794 kw
Neutron Flux.....approx. 5×10^{13} n/cm²-sec

Fig. 1...Controlled recirculation boiling reactor (CRBR) with nuclear superheater for Pathfinder Atomic Power Plant. (206530)

A. Scope of the Problem

In the CRBR, the function of steam separators is to remove steam carryunder from the recirculation water with a minimum of dynamic pressure drop.

Net Positive Suction Head is made up of static water, plus the head available above vapor pressure for water at reactor temperature, minus the dynamic flow losses in the suction side of the recirculation loop. The static head is 16 ft at reactor conditions. The greatest portion of NPSH is made up of vapor pressure above the existing pressure, since about 15 ft of head is gained for every degree of subcooling below the saturation temperature. The effect of steam carryunder on NPSH from loss of subcooling only is shown in Fig. 2.

At rated conditions in Pathfinder, the expected subcooling at core inlet is 3 F, corresponding to about 45 ft of NPSH from subcooling. A small variation in reactor temperature, however, can change NPSH drastically. It was estimated that the high downcomer velocities (> 2 fps) would carry under enough steam to reduce NPSH substantially. Tests confirmed that about 5.5 per cent by volume steam carryunder could be expected, which would reduce NPSH from subcooling by about 15 ft (see Fig. 2). To ensure a minimum recirculation pump NPSH of 24 ft for sustained operation, the design requirement for steam separation was set at one per cent by volume, maximum allowable steam carryunder. Fig. 2 calculations indicate that 2.6 ft of NPSH would be lost when a one per cent steam volume fraction existed in the downcomer.

An analysis of expected dynamic pressure drop in the recirculation loop showed that the pressure drop available for steam separators was $3.5 \text{ ft} \pm .5 \text{ ft}$ of H_2O .

Because of the variable recirculation rate, it became necessary in this test program to define separator characteristics over the whole reactor operating range.

It was also found desirable to obtain data on the split of steam flow between riser and downcomer.

Fig. 2...Loss in NPSH from steam carryunder only vs per cent carryunder by volume...Pathfinder CRBR...steam dome pressure - 615 psia saturated...3.6 F initial subcooling assumed.
(43-024-037)

2. INITIAL INVESTIGATIONS

Preliminary work was started as far back as December of 1956 in regard to development of a separating device to limit steam carryunder in a boiling water reactor.

Available conventional separators were checked for applicability in the CRBR. The following is an analysis of each type considered.

A. Centrifugal Separators

Available data on conventional centrifugal separators (Babcock-Wilcox, Foster-Wheeler and Combustion Engineering) showed that this type of separator had insufficient flow capacity for the space available. At high flow rates, excessive pressure drop was the limiting factor. Data presented by Moyer (Ref. 1) and by Ongman and Steamer (Ref. 2) confirm this observation.

B. Crossover Separators

In this type, separation is accomplished by directing the flow of an up-comer steam-water mixture across the flow of downcomer exiting water. The flow may be directed by radial arcs, inclined radial arcs, corrugated plates, or chevrons.

Two types using radial arcs were evaluated on the basis of Moyer's data (Ref. 1). The findings indicated that there was not enough core periphery in the CRBR to admit the number required to handle total water flow.

Separators with chevrons, and corrugated-plate crossover types separate by a scrubbing action between steam and water. Their common failing is that re-entrainment occurs after the mixture has passed through. A chevron type

unit was tested on an air-water loop (Fig. 3) at the Allis-Chalmers Greendale Laboratories. Angle irons, 1 in. by 1 in. size, were placed in a tank at an angle. As the air-water mixture passed through, air separated and was expelled from the upper end of the angle irons. Although this chevron type separator showed inherent low pressure drop, the re-entrainment problem eliminated it from consideration for use in CRBR.

C. Upcomer Types

Separators of the upcomer type are installed above a boiler core, and usually above individual fuel elements. They perform their function by directing the upward flow into a spiral pattern, with the separation taking place in the vortex that is formed. The steam exhausts from the top, while water exits downward through a shroud around the separator.

Several types of upcomer separators were conceived and tested at Allis-Chalmers for their application in CRBR. These were limited to 4 in. in diameter because of conflict with above-the-core control rod drives. Figures 4 through 13 show examples of the models tested. These separators were designed for attachment to boiler fuel element outlets. However, the upcomer types tested had too much pressure drop at required flow rates. The possibility also exists that the added pressure drop at the core exit might cause hydraulic instability in the core. Re-entrainment of separated steam is also a problem with the upcomer type separator. Some of the upcomer types, however, (those shown in Fig. 8 and Fig. 9), indicated good possibilities for reactors without above-the-core control rod drives. These separators had the minimum pressure drop and re-entrainment characteristics.

Fig. 3...Test setup for chevron type crossover steam separator. (43-024-667)

7-30-57

7-31-57

Fig. 4...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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8-1-57

Fig. 5...

4-in. diameter upcomer steam separators.
tested for Pathfinder reactor.

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8-6-57

Fig. 6...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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8-7-57

Fig. 7...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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Fig. 8...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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Fig. 9...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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9-11-57

Fig. 10...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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9-16-57

Fig. 11...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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9-30-57

Fig. 12...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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10-2-57

10-2-57

Fig. 13...

4-in. diameter upcomer steam separators
tested for Pathfinder reactor.

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D. Downcomer Types

Parallel with the consideration of conventional, crossover and upcomer type separators, a test program had started on a downcomer type separator. This type operates by admitting a steam-water mixture tangentially into a long vertical cylinder. A vortex is formed and is pinched off by an outlet nozzle at some convenient length of cylinder. (A U.S. patent on this type of separator (Ref. 13) was subsequently issued to D.H. Swanson and R.B. Jerman of Allis-Chalmers).

1. One-in. diameter vortex tube tests.

The first tests were run on 1 in. I.D. plastic cylinders shown in Fig. 14. Purposes of these tests were to establish length-to-width ratios on the tangential inlet and to observe the vortex shape with flowing water.

Several L/W ratios were tried. Examples are shown in Fig. 15, A through D. Test objectives were to obtain the maximum flow rate at constant pressure drop while maintaining a uniform, open vortex. Comparison of C and D in Fig. 15 shows two different vortex shapes for about the same inlet area. An L/W ratio of about 10 gave the most satisfactory vortex. Lower ratios (as in illustration C) caused a narrower, more erratic vortex. Higher ratios (as in illustration A) caused shorter vortices, and the water piled up below the inlet nozzle.

2. Five-in. diameter model downcomer separator tests (Plexiglas model).

To gain knowledge regarding the effect of increasing vortex tube diameter and inlet size, and also to develop a method of terminating the vortex, a 5 in. dia. model was made. The test loop consisted of a cylindrical test

Fig. 14...1-in. I.D. vortex tube model ~ (43-024-730)

Fig. 15...1-in. I.D. vortex tube test -- shape and length of vortex with varying inlet dimensions. (43-024-731, 43-024-732)

section 23 in. O.D. and 70 in. high. Fig. 16 illustrates this loop with a separator model included in the test setup.

For the purpose of maintaining the L/W ratio at about 10 (as in the 1 in. dia. model test), the first vortex had these inlet dimensions: length 14 in., width 1.5 in., L/W ratio 9.35. This model produced a narrow, erratic vortex that intermittently closed.

The model was altered in several steps, by lengthening and narrowing the inlet, until a stable vortex was obtained. The final inlet configuration was: length 39 in., width 3/4 in., L/W ratio 52. This gave a vortex of approximately 1 in. dia. which extended out the bottom of the tube. The final configuration corresponds to equality of Froude numbers (V^2/D) for the 1 in. and 5 in. dia. models (see Table II). Pressure drop through the vortex tube was 2 to 3 ft of water at the model flow rate of 525 gpm water.

The air added to the loop was forced to the center and traveled up the vortex void. This was visually observed. Air, however, was carried under into the recirculation line because the vortex did not terminate.

Further testing on the 5 in. dia. model centered on developing an outlet configuration that would close the vortex. Several sets of straightening vanes were inserted in the tube section below the inlet nozzle. These vanes are shown in Fig. 17, A through D. In designing these vanes, the objective was to terminate the vortex by converting the rotational velocities to axial flow; the transition to be made without causing turbulence. This proved difficult to accomplish. At the design model flow rate of 525 gpm, the vortex was not terminated by the vanes, but was accomplished at lower flow rates.

Fig. 16...Test setup for 5-in. diameter vortex tube test. (43-024-733)

TABLE II

PROPERTIES OF EXPERIMENTAL VORTEX TUBES

	<u>1 in. Dia.</u>	<u>5 in. Dia.</u>
Inlet Dimensions	0.3 in. x 3.0 in.	3/4 in. x 39.0 in.
Inlet Area	.000685 sq. ft.	0.203 sq. ft.
Water flow - gpm	6.6 gpm.	500 gpm.
Water flow - cfs	0.0147 cfs.	1.114 cfs.
Inlet Velocity - fps	2.35 fps	5.5 fps
$\frac{V}{\sqrt{g \times r}} = F$	2.03	2.13
L/W	10	52
$\frac{\rho \times V \times D}{\mu} = R_e$	18.8×10^5	220×10^5

FIG 17. VANED OUTLET CONFIGURATION FOR 5 IN.
(CONTINUED) DIA LUCITE VORTEX TUBE MODEL TEST

Fig. 17...Vaned outlet configuration for 5-in. diameter lucite vortex tube model test. (43-024-734, 43-024-735)

The next attempt at terminating the vortex was to close off the bottom of the vortex tube and allow the flow to pass radially from the tube wall. By letting the flow pass out in thin jets, the rotational velocities would be converted and mixed with the downward flow without inefficient turbulence. To facilitate this test, the loop was modified so that the vortex tube passed through a support plate located above the recirculation water outlet. The outlet nozzle could be observed during these tests.

The various outlet nozzle configurations tried in this phase of testing are shown in Fig. 18, A through D. The perforated and vertical slotted outlets shown in illustrations A and B (Fig. 18) proved unsatisfactory. As the vortex entered these outlets it increased in radius, and bubbles were carried down into the recirculation water outlet. It was decided that, since the radius of the vortex increased as it lowered into the radial outlet flow section, decreasing the wall radius which supported the vortex should cause it to become smaller, and ultimately terminate. The outlet sections C and D (Fig. 18) were tested to verify this theory. These outlets functioned satisfactorily at the model flow rate of 525 gpm, and at somewhat above this figure. No voids were carried under into the recirculation pump suction, and flow did not oscillate.

3. Test of a 5 in. dia. aluminum separator model with steam at atmospheric conditions.

Purposes of testing the 5 in. aluminum model were to observe the effect of using atmospheric steam-water instead of air-water at 70 degrees, and to check a calculation of the length of vortex required for separation.

A - PERFORATED OUTLET

B - VERTICAL SLOT OUTLET

C - CONICAL OUTLET
WITH TRIANGLE
SLOTS

D - CONICAL
OUTLET WITH
RECTANGLE SLOTS

Fig. 18...Radial flow outlet nozzle configuration for
5-in. diameter lucite vortex tube model test. (43-024-736)

A comparison of liquid properties between 70 degrees and 212 degrees shows that the largest variance is in viscosity. At 70 degrees, viscosity is about 3-1/2 times higher than that at 212 degrees. The surface tension of water, in contact with air or steam, is about 20 per cent higher at 70 degrees than at 212 degrees. a comparison of the density numbers, $\frac{\rho_l - \rho_g}{\rho_l}$ shows that this number is almost the same for the two temperatures and for air-water or steam-water.

In order to calculate the required length of the separator, several assumptions were made. The mechanism which promotes separation of bubbles in a centrifugal separator is the force of acceleration between the gas and liquid phases. This force of acceleration is opposed by the drag force - a function of bubble radius, bubble velocity, viscosity, and liquid density or, in briefer form, a function of bubble Reynolds Number $(2 R_b \rho_g U) / \mu$. The bouyancy force also has some small effect on the motion of the bubble.

Based on literature searches made by Peebles (Ref. 4 and 5), it became evident that it is hard to decide what final bubble size is achieved under varying liquid-gas properties. There is very little correlation among the observations made by various experimenters regarding bubbles rising in various liquids and in varied-size columns. From observations made of air bubbles in water in the loop shown in Fig. 16, it was determined that a bubble size of 0.1 in. diameter was a fair assumption. By using the Stokes law Theory of Drag and some of the observations of Kozlov and Mologin (Ref. 6) for 0.1 in. diameter bubbles, it was further indicated that a final bubble velocity of about 1.0 ft per second could be expected in a $1 g_c$ field, in a direction opposite to the force of acceleration.

Based on Newton's equation for terminal velocity of a 0.1 in. dia. sphere, it was then assumed that the velocity would be proportional to the square root of the acceleration field. By observing the flow in the 5 in. dia. Lucite vortex tube, the diameter of the vortex was measured as 1.5 in. Based on the 500 gpm flow rate and known flow areas, the velocity components were calculated to be $V_t = 8.5$ fps tangentially and $V_a = 9.04$ fps axially. The average acceleration, based on the tangential velocity and an average radius of flow annulus, was calculated by the equation $a = \frac{V_t^2}{r}$. For the 5 in. vortex tube, this value was 23.4 g_c 's. From the assumption that radial bubble velocity is proportional to $\sqrt{g}$ field, and using 1 fps in a 1 g_c field, the radial bubble velocity was estimated at 4.7 fps. This gives a bubble hold-up time of .0303 sec., and required vertical separating length (below the inlet nozzle) of about 5 in.

On the basis of these estimates, the 5 in. dia. aluminum model was fabricated with the dimensions as shown in Fig. 19. The model was run at the design flow rate of 500 gpm and approximately 70 cfm of steam was introduced. It was observed that the discharge of the separator was relatively free from bubble carryunder. Fewer bubbles were noted than in the previous (air-water) test. Constant flow rate from the recirculation pump indicated that no significant amount of voids was entering the pump suction.

It was concluded that, as estimated, the 1 ft long separating length allowed below the separator inlet was sufficient. The test performance also indicated that the property differences between steam-water and air-water testing had little effect on the mechanism of centrifugal separation.

Fig. 19...5-in. I.D. aluminum model of Pathfinder steam separator. (43-024-668)

4. Test of two 3 in. dia. lucite separator models.

In order to study the flow patterns developed, and the general operation of downcomer type separators in a group, an existing test apparatus was modified to permit testing of two 3 in. dia. model separators with air-water flow. The 3 in. model flow rate was proportioned from the 5 in. dia. model on the basis of constant Froude No., $F = \frac{V}{\sqrt{(g_c)(r)}}$ for the inlet nozzle. This water flow rate was 180 gpm per separator. The flow areas in the test apparatus were then sized on this same basis to match reactor conditions for two separators. The model air flow was set at a value which resulted in a mixture of 50 per cent air and 50 per cent water, by volume.

Fig. 20 shows the separator test apparatus with no separators and no flow. The air-water mixture enters through the bottom of the tank and is discharged through the mesh cone. The recirculation water outlet is on the side of the tank, at the bottom-right, below the separator support plate. Fig. 21 shows the loop operating at rated flow conditions with no separators in place. This is evidence of much carryunder. The 3 in. dia. separators, operating at rated model conditions, are shown in Fig. 22. The vortex can be seen in the separators, and no carryunder is evident in the separator discharge plenum.

2.1 CONCLUSIONS FROM INITIAL INVESTIGATIONS

Investigation of existing types of steam separators showed that they could not be applied in the Pathfinder reactor.

Tests conducted for development purposes on upcomer and crossover type separators showed that pressure drop and space considerations would not allow their use.

Fig. 20...3-inch separator model test apparatus without separators.
(196505-A)

Fig. 21...Test apparatus operating at equivalent reactor flows without separators. (196505-C)

Fig. 22...Test loop operating at equivalent water and air flow rates -
with separators. (196505-B)

The relative success of tests on downcomer type separators indicated that this type was the best suited for Pathfinder. By utilizing the downcomer area for separation, a more economical-size vessel could be used and the centrifugal principle (which offered the best mechanism for compact, efficient separation) could be applied.

It was decided that the major effort would be concentrated on developing a downcomer type steam separator.

3. DEVELOPMENT OF REACTOR MODEL FLOW RATES AND DIMENSIONS BASED ON SCALE MODEL PERFORMANCE

The next step in developing a steam separator was to arrive at an optimum diameter, inlet nozzle and length which would handle the required amount of steam-water mixture while occupying a minimum of space in the reactor vessel.

At this stage in reactor design, the boiler core diameter had been fairly well established at six ft. Without separators, a vessel diameter of 14 ft. would be required to limit downcomer velocity to 1 fps, and therefore eliminate carryunder of steam.

The final outcome of an economics-and-flow configuration study was to set the vessel I.D. at 11 ft and use 10-in. dia. downcomer type steam separators. The following analysis was made to arrive at the flow and dimensional characteristics of the reactor model separator.

A. Inlet nozzle design.

The Froude numbers for the 1-in. and 5-in. dia. models (listed in Table II) average out to about 2. On the basis of keeping this number the same, to obtain a configuration with the same gravitation effects, the inlet velocity of 7.7 fps was obtained for the 10 in. separators. The height of the core and position of the separator support shelf limited the length of the inlet nozzle to 4.5 ft. This was later substantiated to be the calculated, required separation length below the inlet nozzle, and the required length of the outlet nozzle. To get a first estimate of the design inlet nozzle width, based on the 5-in. dia. model tests (see Table II), an L/W ratio of 52 was used. This is not necessarily the optimum L/W ratio, since it can vary, depending upon the actual diameter of the vortex tube.

Since L/W was 52 and $L = 54$ in., the first estimated inlet width (W) was 1.05 in. Designing for water flow only, on a consistent basis with the 1-in. dia. and 5-in. dia. tests, the expected water flow rate can be calculated. The 1.05 in. x 54 in. inlet gives an inlet area of .39 sq. ft. The expected flow rate is then: $.39 (7.7) = 3.0$ cfs or 1350 gpm.

The next consideration in sizing the inlet was to make an estimate of the pressure drop that would be incurred.

On the basis of water flow only, using the orifice flow head loss equation $Q = C(A)\sqrt{2g\Delta h}$, a Δh of 1.4 ft was obtained. This was based on an orifice flow coefficient of $C = 0.7$. Actual head loss, of course, would depend upon the entering void fraction through the separator inlet. For estimating purposes, the inlet drop for two-phase flow could be as much as 50 per cent greater than the liquid phase drop. For this reason it was thought that the inlet width might have to be increased to 2 in. or 3 in. (As will be noted later, the test model separator was fabricated with an adjustable width inlet nozzle to compensate for the two-phase Δh factor).

Reynolds number of the 10-in. dia. model was checked at this flow rate to ensure that frictional effects would be the same as in previous models. The Reynolds number, calculated on the same basis as in Table II, was 1680×10^5 , showing that it, also, was in the turbulent region.

B. Separation length design.

The length required below the inlet nozzle for separation was calculated on the same basis as that of the 5 in. dia. model. Since the Froude number was kept the same, it was expected that the gravitational effect would be similar. The bubble holdup time in the 10-in. model was calculated

at .079 seconds. A vortex diameter of 1.5 in. was assumed. Using the same axial and tangential velocities as in the 5-in. model, and based on a calculated radial bubble velocity of 4.5 fps, a required separating length of 0.66 ft was calculated. A more complete analysis of required separating length, based on test data, is presented in Section 4-C.

C. Outlet nozzle design.

As determined from the 5-in. lucite model tests of outlet configurations, the reactor model outlet was to be conical shaped with slits for radial flow. The reactor model outlet was scaled up from the previous tests to give the same decrease in radius per unit length. The outlet area was sized on the basis of expected outlet drop, with a flow outlet which would decrease the rotational velocities at a rate slower than the rate of decrease in radius. The model outlet cone was 18 in. long, with a 4 in. dia. disc in the bottom. The outlet area was formed by eight slits situated vertically. Each slit was 1-in. wide by $17\frac{5}{16}$ in. long. Since both rotational and axial flow is discharged, it was assumed that about 40 per cent of the open area was effective, or $.96 \text{ sq ft} \times 40 \text{ per cent} = .385 \text{ sq ft}$. The expected outlet drop was .8 ft, to give a total expected drop of about 3.5 ft.

D. Full scale model description

Figure 23 is a drawing of the separator model arrived at from the initial estimates. Inlet nozzle width was made adjustable to a 3 in. opening. The outlet cone was fabricated and attached in such a manner that it could be removed and modified when warranted during the tests.

Fig. 23...First 10-in. diameter separator model fabricated from 16 gage galvanized sheet metal -- with adjustable inlet and removable outlet nozzle. (43-024-879)

4. THEORETICAL AND PHYSICAL FACTORS AFFECTING CENTRIFUGAL SEPARATION

The determination of required separating length for the centrifugal separator uses equations developed by Peebles (Ref. 4). These equations require that we know the following variables:

Viscosity	μ	#M/Ft Sec.
Mass Density	ρ	#M/Ft ³
Surface Tension	γ	#F/Ft
Bubble Radius	R_b	Ft
Vortex Diameter		Ft
Tangential Velocity of Fluid in Separator	V_t	Ft/Sec

Some of these variables are readily available from tables. The bubble size, vortex diameter, and tangential velocity will be derived or developed as needed.

A. Estimation of bubble size.

Bubble velocity in a fluid is influenced in part by bubble size. A correlation is needed so that this size can be determined for given conditions of temperature and velocity of the fluid surrounding the bubble. The bubble size can be estimated (considering the existing physical properties) as suggested in "Steam Volume Fraction in a Bubbling Two-Phase Mixture," by Wilson, Grenda, and Patterson (Ref. 8). In this approach, a force balance is made on the bubble film. The ring stress on the bubble film is proportional to the density difference between the vapor in the bubble and the water displaced by the bubble.

Displacement:

$$\text{Bubble Volume} - V = 1/6 \pi D_b^3$$

$$V/2 = 1/12 \pi D_b^3$$

External force on Bubble Film:

$$F = 1/12 \pi D_b^3 (\rho_l - \rho_g) \frac{g}{g_c}$$

Circumferential Reaction Force in the Film:

$$F = \gamma \pi D_b$$

Equating:

$$\gamma \pi D_b = 1/12 \pi D_b^3 (\rho_l - \rho_g) \frac{g}{g_c}$$

$$D_b^2 = \frac{12 \gamma}{(\rho_l - \rho_g) \frac{g}{g_c}}$$

Bubble Diameter:

$$D_b = 3.47 \sqrt{\frac{\gamma}{(\rho_l - \rho_g) \frac{g}{g_c}}}$$

(Units of D_b are Ft.)

Turbulence, drag forces and an increased gravitational field will cause a bubble in moving fluid to be reduced from its stationary fluid size. Data by M. Petrick in the report "Nuclear Superheat Meeting No. 3" (Ref. 9), shows a reduction in bubble diameter of 2.5 times with a water flow increase from 1 to 4 ft/sec for an air-water mixture. Since velocities in the centrifugal separator are considerably higher than 4 ft/sec, and the gravitational field is greater than one, a size reduction factor of 3.3 is used for the first approximation.

Thus for 175° water (air-water test loop conditions):

$$\gamma = .0044 \text{ #F/Ft} \quad (\text{Ref. 12})$$

$$\rho_l = 60.6 \text{ #M/Ft}^3$$

$$\rho_g = .1 \text{ #M/Ft}^3$$

Then:

$$D_b = (.30) (3.47) \sqrt{\frac{\gamma}{\rho_l - \rho_g}}$$

$$D_b = (.30) (3.47) \sqrt{\frac{.0044}{60.5}}$$

$$D_b = (0.00857) (12) = 0.103 \text{ in.}, \text{ Use } 0.100 \text{ in.}$$

$$R_b = \frac{(0.100)}{24} = 0.00417 \text{ Ft.}$$

For 490° Steam-Water:

$$\gamma = 0.00172 \text{ #F/Ft.}$$

$$\rho_l = 49.5 \text{ #M/Ft.}$$

$$\rho_g = 1.35 \text{ #M/Ft}$$

Then:

$$D_b = (.30) (3.47) \sqrt{\frac{\gamma}{\rho_l - \rho_g}}$$

$$D_b = (.30) (3.47) \sqrt{\frac{.00175}{48.15}}$$

$$D_b = (0.00602) (12) = 0.0723 \text{ in.}$$

$$R_b = \frac{0.0723}{24} = 0.00301 \text{ Ft.}$$

The derived bubble diameter of .10 in. for the 175° air-water mixture compares favorably with bubble size as seen in pictures taken with a high speed camera of the separator discharge.

Pictures were also taken of bubbles in a 600 psig test loop (Figs. 24-A, 24-B and 25). Observed steam bubble size was slightly above .10 in. After correcting for the difference in water velocities and resultant decrease in bubble size due to drag forces and turbulence, these bubbles compare in size to the derived size estimate for the 490° steam-water bubble.

B. Analysis of vortex velocities and configuration.

Once the size of bubble has been determined, forces on the bubble in the vortex must be analyzed. The vortex in the centrifugal steam separator is generated by water entering the separator tangentially through the inlet nozzle. If a relatively long inlet is used, the influence of the inlet velocity should be spread over a greater length of the separating zone. This should tend to keep drag at the wall from reducing the peripheral velocity.

A number of observers have used the equation $VR^n = C$ to describe tangential velocities in a vortex. This equation relates the inlet velocity with the average peripheral velocity of the rotating fluid. The exponent "n" has been allowed to vary from 0.5 to 1.0. The conditions where "n" is less than 1.0 are characterized by the separator inlet being relatively small, compared to the separating zone as a whole.

The Pathfinder centrifugal separator is assumed to have a "free" vortex, that is, angular momentum is conserved. It is also assumed that the peripheral velocity of the vortex is equal to the fluid velocity in the inlet nozzle. This allows the prediction of rotational velocities within the separator, using the equation $VR = \text{Constant}$.

The above approach does ignore the fact that the tangential velocity

One-inch scale
for convenient
measuring.

Fig. 24-A...Photo through 6-in. diameter window of air-water test loop at separator discharge. Test conditions: 175 F., 1000 gpm water flow rate. Separator and bubble pictures are about 7/8 actual size because of perspective. Fluid exit velocities are in the order of 15-20 fps, which causes smaller bubble sizes than would be expected from temperature effects alone. Water level has been lowered for this picture - so that vortex can be seen in the discharge nozzle.

One-inch scale
for convenient
measuring.

Fig. 24-B...Same as Fig. 24-A except that test conditions were: 175F, 1500 gpm water flow rate.

One inch on scale equals
one-half inch in photograph.

Fig. 25...Photos of steam bubbles taken at various times through 1/2-in. diameter sight glass in a 4-in. diameter test column. Test conditions: 600 psig ...saturated, superficial steam velocity about 1.5 fps.

seems to fall off for the liquid in the innermost section of the vortex. The depth of this inner layer is slight, compared to the total radial distance traveled by the bubble, so the decrease in bubble velocity in this zone does not greatly affect the bubble transit time.

Prediction of the axial velocity in the separator is less definite. Knowing the displacement of the vortex core, the average axial velocity can be determined for a given flow rate. Velocity measurements by R. H. Wilson (Ref. 10) and Ohashi and Maeda (Ref. 11) indicate that the downward component is highest at the separator wall. At the inner surface of the vortex the downward velocity is usually low and may even be upward. Observation of the Pathfinder separator has shown that the inner surface layer of water in the vortex was often climbing upward. In the absence of more definitive knowledge of these secondary currents, the axial bubble travel must be defined in terms of the average axial water velocity.

Using these assumptions, the velocity profile for the separator is determined at flow rates of 1000 and 1500 gpm in the following calculations:

PATHFINDER ... 1000 gpm:

Inlet $1.80 \text{ in.} \times 54 \text{ in.} = 97 \text{ sq. in.} = .675 \text{ sq ft}$

Inlet Head Loss $\approx .825 \text{ ft water}$ (from later data)

1000 gpm = 134 cfm = 2.23 cfs

Vortex $\approx$ 2.5 in. dia.

Inlet Coefficient:

$$Q = CA\sqrt{2gh}$$

$$A = .675$$

$$h = .825$$

$$Q = 2.23$$

$$C = \frac{Q}{A\sqrt{2gh}}$$

$$A\sqrt{2gh}$$

$$C = \frac{2.23}{.675 \sqrt{64.4 \times .825}}$$

$$.675 \sqrt{64.4 \times .825}$$

$$= \underline{\underline{.455}}$$

Inlet Velocity:

$$V = \frac{Q}{AC}$$

$$= \frac{2.23}{.675 \times .455}$$

$$.675 \times .455$$

$$= \underline{\underline{7.25}} \text{ FT/SEC}$$

Tangential Velocity:

$$V_r = C$$

$$r = 5'' = .417'$$

$$4'' = .333'$$

$$3'' = .250'$$

$$2'' = .167'$$

$$1.25'' = .104'$$

$$V_r = 7.25 \times .417 = 3.02 = C$$

$$\begin{aligned}
 V &= \frac{C}{R} &= \frac{3.02}{.333} &= 9.1 \text{ FT/SEC} \\
 &&= \frac{3.02}{.25} &= 12.1 \text{ FT/SEC} \\
 &&= \frac{3.02}{.167} &= 18.1 \text{ FT/SEC} \\
 &&= \frac{3.02}{.104} &= 29.0 \text{ FT/SEC}
 \end{aligned}$$

The mean tangential velocity is ~ 13 FT/SEC. At a radius of 2.75" (See Fig. 26).

PATHFINDER...1500 gpm:

Inlet Head Loss $\approx$ 2.30 Ft. water (From later data)

1500 gpm = 201 CFM = 3.35 CFS

Vortex $\approx$ 4" Dia.

Inlet Coefficient:

$$A = .675$$

$$h = 2.3$$

$$Q = 3.35$$

$$C = \frac{Q}{A \sqrt{2gh}}$$

$$C = \frac{3.35}{.675 \sqrt{64.4 \times 23}} = .409$$

Inlet Velocity:

$$V = \frac{Q}{AC}$$

$$V = \frac{3.35}{.675 \times .409} = \underline{12.15 \text{ FT/SEC}}$$

Tangential Velocity:

$$V_r = C$$

$$12.15 \times .417 = \frac{5.07}{.333} = C$$

$$\frac{5.07}{.333} = 15.2$$

$$\frac{5.07}{.25} = 20.2$$

$$\frac{5.07}{.167} = 30.4$$

The mean tangential velocity is ~ 18.5 fps at a radius of 3.25 in.

(see Fig. 26).

C. Bubble trajectory in the separator and required separating length.

1. Calculate the bubble velocity in the separator for 1000 gpm, 175° air-water mixture. Assume Peebles (Ref. 4), Region 4, and a .10 in. dia. bubble entering the separator.

Region Lower Limit:

$$U_{\min.} = \frac{1.505}{(g)^{.25}} \frac{(y g_c)^{.75}}{\rho_l} \frac{1}{R_b}$$

$$= \frac{1.503}{2.38} \left[\frac{.0044 \times .32.2}{60.5} \right]^{.75} \frac{1}{.00417}$$

$$= .632 \times .0100 \times 240 = 1.52 \text{ FT/SEC}$$

Region 4 minimum velocity - this bubble

Bubble Velocity (to check whether these bubbles are in Region 4):

$$U = .825 \left[\frac{N_G^4}{g} \right]^{.083} \left[\frac{y g_c}{\rho_l} \right]^{.25}$$

Fig. 26...Pathfinder separator tangential velocity profile. (43-025-093)

$$N_G = \frac{8 v_t^2 (\rho_1 - \rho_2)}{3 R \rho_1}$$

$$v_t = 13 \text{ FT/SEC}$$

$$R = 0.23 \text{ FT}$$

$$N_G = \frac{(8) 13^2 (60.5)}{3 (23) 60.6}$$

$$N_G = 1,950 \text{ ft/sec}^2$$

$$U = .825 \left(\frac{(1,950^4)}{(32.2)} \right)^{.083} \left(\frac{(.0044 \cdot 32.2)}{60.6} \right)^{.25}$$

$$U = (.825) (.9365) (10) (.218) \quad 1.60 \text{ FT/SEC. Radially}$$

The velocity places this bubble within Region 4. The velocity assumption was correct.

Radial Travel: (from $r = 5 \text{ in.}$ to $r = 1.5 \text{ in.}$)

$$.417' - .104' = .313'$$

Transit Time:

$$T = \frac{s}{u}$$

$$T = \frac{.313}{1.68} = .186 \text{ seconds}$$

Within the separating zone the average axial component (assumed proportional to the average downward water velocity) is 4.39 fps. The average tangential velocity is 13 fps.

Estimated bubble Path:

$$\text{Axially : } 4.39 \times .186 = \underline{.82 \text{ Ft.}}$$

$$\text{Tangentially: } 13.0 \times .186 = \underline{2.42 \text{ Ft.}}$$

The tangential travel is approximately 1.5 revolutions, which agrees with photographs taken of the Pathfinder vortex when operating at 1000 gpm.

2. Calculate the bubble velocity in the separator for 1500 gpm and 175° air-water. Assume Peebles Region 4 (Ref. 4) for a .10" diameter bubble.

$$U = .825 \left[\frac{N_G^4}{g} \right]^{.083} \left[\frac{(\gamma_{gc})}{\rho_1} \right]^{.25}$$

$$N_G = \frac{8 v_t^2 (\rho_1 - \rho_2)}{3 R \rho_1}$$

$$v_t = 18.5$$

$$N_G = \frac{8 \times 18.5 (60.5)}{3 \times .271 \times 60.6}$$

$$R = .271$$

$$N_G = 3370 \text{ ft/sec}^2$$

$$U = .825 \left(\frac{3370^4}{32.2} \right)^{.083} \left(\frac{.0044 \times 32.2}{60.6} \right)^{.25}$$

$$U = (.825) (1.124) (10) (.218) = 2.02 \text{ FT/SEC Radially}$$

Radial Travel: (from 5" radius to 2-1/4" radius)

$$.417' - .167' = \underline{.250 \text{ Ft}}$$

Transit Time:

$$T = \frac{s}{u}$$

$$T = \frac{.250}{2.02} = \underline{.124 \text{ seconds}}$$

Within the separating zone the average axial component is 7.26 fps.

Average tangential velocity is 18.5 fps.

Estimated Bubble Path:

$$\text{Axially} \quad 7.26 \times .124 = .90 \text{ ft.}$$

$$\text{Tangentially} \quad 18.5 \times .124 = 2.30 \text{ ft.}$$

The tangential travel is approximately 1.3 revolutions.

The higher accelerating forces of the 1500 gpm condition, combined with the reduced radial travel required because of the increased size of the vortex core, give a shorter residence time for the bubble at 1500 gpm.

At the same time the increased axial velocity draws the bubble farther down in the separator than the 1000 gpm condition. The 1500 gpm rate is thus more severe in terms of required length for separation.

Since the 1500 gpm condition is the more severe, it is used in the calculations to define separator requirements for steam-water service. In considering the steam-water mixtures at increasing temperatures and pressures, it is assumed that the inlet nozzle and outlet nozzle head losses (which are expressed in ft of water) remain constant for a given rate of volume flow. Therefore, the vortex dimensions will remain the same. The variables will be density, which influences the acceleration force on the bubble, and surface tension, which influences the bubble size.

Applying the equations used in the previous calculations at varying increments of temperature, the following data was developed for 1500 gpm water flow. This data is plotted in Fig. 27.

Temp.	D_b	Region 4 Min. $U_{min.}$	fps	U	Transit Time	Axial Travel
175°	.10"	1.52		2.02 ft	.124 sec	.90 ft
300°	.092	1.53		1.94	.129	.94
400°	.082	1.40		1.83	.137	1.00
489°	.074	1.35		1.69	.148	1.07
500°	.070	1.33		1.65	.152	1.10
600°	.049	1.09		1.35	.185	1.34

Fig. 27...Pathfinder separator calculated bubble path...1500 gpm, varying temperature...postulate .10-in. air-water bubble 175F. (43-024-C72)

Equations

Bubble size for Pathfinder operating conditions

$$D_B = (.30) (3.47) \sqrt{\frac{\gamma}{\rho_1 - \rho_2}}$$

Region 4 minimum velocity limitations

$$U_{\min} = \frac{1.505}{g^{.25}} \left(\frac{\gamma g_c}{\rho_1} \right)^{.75} \frac{1}{R_b}$$

Peebles (Ref. 4)

Region 4 bubble velocity equation

$$U = .825 \left[\frac{N_G^4}{g} \right]^{.083} \left[\frac{\gamma_{gc}}{\rho_1} \right]^{.25}$$

Peebles (Ref. 4)

D. The effect of viscosity on carryunder.

A steam bubble (or an air bubble as in air-water tests) encounters two different environs in the downcomer-separator system. The bubble entering the downcomer is in the gravitational field of the earth, or, for convenience, a $1 g_c$ field. During this phase, "gravity" separation takes place. Acceleration forces on the bubble are due to the water displaced by the bubble in the $1 g_c$ field. When a bubble enters the centrifugal steam separator, it is exposed to increased acceleration forces. A gravitational field of $30 g_c$'s is not exceptional for the separator. The bubble is thus exposed to considerably greater accelerating forces.

Carryunder measurements were made with the downcomer section of the test loop, with the separator removed (Fig. 28). The test loop is described in Section 6 of this report. Air and water flow rates were maintained constant while the temperature was increased from 90F to 180F. Carryunder decreased considerably with the increase in water temperature. Tests of the loop with the separator in place (and under the same conditions) did

Fig. 28...Separator standpipe -- per cent carryunder vs water temperature -- constant water flow and air flow...4-ft static water level above riser... without separator. (43-024-325)

not show any major change in carryunder rate from that for the separator at the initial low temperature. (See Fig. 29).

Physical properties of the water which vary for this temperature range are:

Surface tension ...varying from 4.85×10^{-3} to 4.30×10^{-3} lb/ft ($\sim 12\%$)

Density...varying from 62.1 to 60.35 lb/ft³ ($\sim 3\%$)

Viscosity...varying from 51.4×10^{-5} to 22.0×10^{-5} lb/ft.sec ($\sim 130\%$)

It would appear that viscosity should have the greatest effect on bubble velocity if it is relevant to our equations. The equations of Peebles and Garbor (Ref. 5), and of Peebles (Ref. 4) describe bubble behavior for the conditions experienced in this system. In Region 2, as defined by Peebles and Garbor, bubble velocity increased proportionally with $\sqrt{1/\mu}$ as temperature increases. Region 3 is beyond the Stokes Law approach and is not affected by viscosity.

Analysis of the .10 in. dia. bubble in the downcomer shows it to be governed primarily by Region 3 considerations, and thus unaffected by viscosity. On the other hand, the bubble size range of .050 in. to .075 in. falls within the boundaries of Region 2 for the lower temperatures, shifting into Region 3 in the 140F to 170F temperature range. These bubbles are among the smallest sizes expected in the test loop. They certainly are below the estimated mean bubble size.

It is then concluded that initial carryunder in the downcomer is caused by small bubbles whose velocity is initially affected by the viscosity considerations of Region 2. As the temperature increases, the viscosity decreases and fewer bubbles are carried down, the curve of decreasing carryunder levels off as Region 3 conditions prevail, governing the bubble

Fig. 29...Per cent carryunder vs water temperature °F -- constant water flow and air flow...with modified Pathfinder separator (3-in. wide inlet nozzle. (43-024-328)

velocity proportionally with surface tension and density.

Carryunder measurements taken with the centrifugal separator do not show the same effect of viscosity as in the case of the downcomer alone. The increased gravitational field in the separator, and consequently increased accelerating forces, increase bubble velocities to beyond the limits of the viscosity sensitive Region 2. The bubble in the separator is governed by the equations of Region 3 and Region 4 where the pertinent variables are surface tension and density. For this temperature range, the changes in surface tension and density are relatively slight. Carryunder is then essentially constant for these restricted conditions.

Actual testing of the Pathfinder steam separator was done at temperatures above 170F, so that viscosity differences which would affect the downcomer could be minimized and bubble velocities in the downcomer maintained in Regions 3 and 4. At actual operating conditions (490F, 600 psig), both the downcomer and the separator operate in Regions 3 and 4.

5. SCOPE OF FULL SCALE AIR-WATER TEST PROGRAM

A. Test Conditions and Flow Capacities required.

Water Flow Rate. As previously indicated in Section 3-A, design of the prototype separator on the basis of equal Froude numbers resulted in a water flow rate of about 1350 gpm. It was decided that the loop water flow rate capacity for the test would have to be at least 1500 gpm.

Taking into consideration the space available in the downcomer and the predicted design flow rate, a reference design of 45 10-in. dia. separators was set. This gives a flow rate of 1330 gpm per separator for the Pathfinder installation.

Test Conditions. As related in the preceding Section, a study was made of the effect of loop conditions on the various physical properties of air and water. It was noted that by testing at about 175F, about two-thirds of the difference in water viscosity between room temperature and reactor temperature (489F) could be eliminated. This is shown on Fig. 30.

Calculations were made to determine what loop pressure would have to be maintained to make air density correspond more closely to that of reactor steam. This was calculated to be 320 psia, for $p_{stm} = p_{air}$. A test loop rated for this pressure was not economically feasible, especially in view of the fact that the density number, $\frac{(\rho_l - \rho_g)}{\rho_l}$, which is a measure of acceleration effect) is almost identical for air-water and steam-water. It was required, however, that the separator exhaust steam to the same pressure as that existing above the water interface. Since it was planned that the separator would exhaust outside the loop, and that orifices were to be used to measure this exhaust, it was decided to maintain loop air

Fig. 30...Viscosity of water vs temperature. (43-025-097)

pressure at 5 psig so the drop across the outlet orifice would not be significant. Another reason for testing at lower pressure was that maximum visual observation of flow pattern was desired. Observation ports are expensive items to install in pressure vessels and their use during tests can be dangerous.

Air Flow Rate. At rated conditions, the reactor boiler core steam output is 614,000 lb/hr. This is 7680 cfm. Since it was decided to use 1/45th of the volume flow rate for the loop tests, a required air flow at 175F and 5 psig was 170 cfm. This flow rate corrected to the orifice measuring conditions was 175 cfm free air. (Note: 175 cfm was also about the maximum output of the laboratory air compressor).

B. Test Loop Measurements Required.

Flow rates:

Total water recirculation flow rate.

Total air supplied to test section.

Air flow exhausted from the separator.

Carryunder measurement...air flow from the separator
water outlet nozzle.

Air flow from the "steam dome" area, i.e., air which
separated from the water interface and was
not carried into the separator.

Pressures and Pressure Drops:

Separator inlet nozzle pressure drop.

Separator outlet nozzle pressure drop.

Separator total drop.

Pressure in the "steam dome" area above the water interface.

Pressure of the collected carryunder.

Temperatures:

Temperature of loop water.

Temperature of collected carryunder.

C. Visual observation requirements.

Visual observation of flow patterns in the test loop was required at the following locations:

Approximate position of the expected air-water interface.

Just above the outlet from the simulated core discharge.

In the "steam dome" area for observation of moisture carryover above the interface.

At the position of the separator water outlet nozzle for observation of the outlet flow pattern, observation of bubble size, and possibly for observation of the vortex tail in the outlet nozzle.

D. Height of standpipe required.

The required height of water (hot, with no voids present) had been set at 4 ft above the fuel. This was based upon requirements for shielding and reflector.

The top of the separator inlet nozzle was to be located at the top of the core outlet. Total estimated height required in the test loop standpipe was the sum of downcomer height, plus static water level height, plus steam dome area high enough to allow for increase in interface height from addition of voids. The separator was to be supported about 5 ft off the floor to facilitate observation. For water height and downcomer height, 8 ft. was added, and 5 ft was added for interface expansion, or a total of 18 ft. To allow for a steam dome height and to provide room for observation of the surface, the final standpipe height was therefore set at about 25 ft.

6. DESCRIPTION OF TEST EQUIPMENT AND PROCEDURE

A. Description of test loop.

Components....as shown in Fig. 31, water flow was maintained by a 12x12 type SE-Allis-Chalmers pump (item 7). This pump is rated at 4000 gpm and 80 ft head. Water flow was regulated by means of an 8-in. gate valve (item 25). The diagonal pipe section which directs flow into the standpipe also houses a 30 kw bayonet type immersion heater. Air was introduced into the water flow at a point about half way up the diagonal pipe. This air flow is regulated by a 2-in. gate valve.

The standpipe section of the test loop is 19-3/4 in. inside diameter. The diagonal pipe discharges the air and water mixture through a baffle sleeve (item 18) in the standpipe. This sleeve prevents the mixture's full force from hitting the separator and the baffles simulate the height of the boiling core. The separator model fits into the support plate in the bottom of the standpipe. Thus all of the water flow was made to pass through the inlet nozzle of the separator. The separator was supported near its top by two bolt clamps inserted in brackets mounted inside the standpipe. The standpipe cover had a 3-in. pipe coupling welded into it, to allow air to discharge from the steam dome area. The cover also had provision for passing a 2-in. pipe from the steam separator exhaust. As previously described in this report, several 6-in. diameter plexiglas portholes were spaced along the standpipe to allow observation of the flow.

A filter, consisting of six cotton wound cartridges, was connected around the recirculation pump, and a small portion of the loop flow was continuously passed through the filter. Thus the water was maintained clear for good visual observation.

Fig. 31...Steam separator test assembly. (43-500-649)

As the air-water mixture flowed through the separator, the vortex was formed and separation of air and water took place. The water discharged from the separator then passed through a diffusion tank (item 3). This diffusion tank was slightly inclined with the horizontal, so that any air which was carried under would rise and pass to the downstream end. An air collection drum (item 12) was mounted at the downstream end. A sight glass on this drum showed how much air had been collected, and a valve on the top allowed the collected air to be dispelled.

Instrumentation...The instrumentation on the steam separator test loop was designed to measure flow rates and pressure drops throughout the system. Water flow rate was measured by means of a sharp-edged orifice, located just before the 8-in. throttle valve. Air flow rates in the test loop were measured by orifices in the supply air line, the separator exhaust line, and the steam dome discharge pipe. The air carried under was measured by timing the rate of water displacement over a known length on the collection drum sight glass.

Pressure taps were placed on the steam separator model and in the standpipe, so that total pressure drop through the separator could be measured and, also, the separator inlet drop. The separator outlet drop could then be obtained by subtraction from the total drop, or it could be measured directly. Static pressure was measured in the supply air line and in the air space of the collection drum. A static pressure tap in the standpipe was used to indicate water level. Another tap in the steam dome area measured air pressure above the interface.

Temperature in the collection drum air space was measured by a dial thermometer. A thermocouple was installed in the diagonal pipe to measure

temperature of air-water mixture entering the loop. Temperature of the supply air was also measured.

Simulation of reactor conditions... In order to closely simulate reactor conditions, flow areas in the standpipe were proportioned so that velocities in the test loop would duplicate those expected in the reactor. The flow areas of the downcomer around the separator, and, specifically, the outlet of the baffle sleeve, were sized on the basis of the continuity equation to give these velocities with 1/45th of the reactor flows (see view B-B on Fig. 31). This also assures equality of Froude numbers, and virtual equality of Reynolds numbers, between test loop and reactor. The baffle sleeve in the standpipe also duplicates the position of the boiling core outlet in relation to the top of the separator inlet nozzle.

The configuration of piping used for maintaining 5 psig in the loop is shown on Fig. 32. The air flow was throttled downstream of both separator and steam dome discharge orifices to maintain the desired pressure.

B. Description of test procedure.

The procedure for bringing the loop up to test conditions and taking data was as follows:

1. Loop was filled with water to desired level and was heated to 175F.
2. The desired air supply flow rate was set and the exhaust valve was throttled to give 5 psig in the steam dome.
3. Data were then taken at various water flows.
4. A new air supply flow rate and throttle valve setting was applied to the loop and data were again taken at several selected water flow rates.

Fig. 32...Steam separator test loop air exhaust piping.
(43-024-851)

The static pressure and pressure drop data were taken in inches of mercury. These were converted to feet of water at test conditions to put head loss on a common basis.

Carryunder air was measured by timing the displacement of the water level in the air collection tank. A 6-in. length was marked off on the collection tank sight glass. As water level was depressed by the air being collected, the total time for the level to traverse six inches was timed. Since the volume of the tank was known for the timing length, the volume flow rate of carryunder could be calculated. Volume per cent of the water flow rate was then calculated and plotted versus flow rate.

$$\text{Volume per cent carryunder} = \frac{\text{Volume air carryunder}}{\text{Vol. water flow} + \text{Vol. air carryunder}}$$

Since the volume flow of air carried under was very small in comparison with the water flow rate, this usually reduced to simply dividing the volume rate of carryunder by the volume flow rate of water. Pressure and temperature of the collected air were measured for the purpose of converting back to free air and accounting for all of the air supplied to the loop.

As an accountability check, air flow from both the separator exhaust and steam dome exhaust was measured. Once it had been established that the flows added up, the exhaust flows were plotted as a per cent of supply air, e.g., per cent air into separator and per cent air from steam dome exhaust, all on a free air basis. This gave an indication of the split of air between steam dome and separator.

7. RESULTS AND DISCUSSION OF RESULTS

A. Description of tests and results, leading to the final model configuration

A.1. Modification of outlet and inlet nozzles...Initial tests on the separator model were run to set nozzle dimensions and obtain the desired pressure drops. The first inlet nozzle was 1-5/16 in. wide and 54 in. long. The outlet nozzle was as shown in Fig. 23. With water flow only, the head loss at 1300 gpm was 4.5 ft., total drop, splitting with 2.0 ft and 2.5 ft between inlet and outlet nozzles respectively. For the inlet nozzle, this meant that the flow coefficient was actually about 0.5 instead of the 0.7 assumed. On this basis, a new required inlet area was calculated from the head loss equation. The adjustable inlet was opened to 2-1/16 in. and the expected head loss with water only was 1.5 ft.

Although the outlet nozzle head drop was not too far out of proportion with the expected drop, a trial run with air flow showed that excessive air was being carried under. Closer analysis of the rectangular slits of the outlet nozzle showed that the topmost section of these slits was passing a greater proportion of water flow than the lower portion. This was logical, because the rotational velocities (and, therefore, the forces pushing the water against the wall) were higher. The water that flowed from the upper part of the slits seemed to carry some of the air out with the water, causing high carryunder. If the outflow of water could be restricted to an area where the rotational velocities had been decreased, lower carryunder should result. To facilitate this principle, a new outlet cone was designed which had triangle-shaped slits as shown in Fig. 33. This outlet cone had eight triangular slits that were 1-1/2 in. wide at the bottom. The cone was 24 in. long, and the bottom disc was

Fig. 33...Modified steam separator outlet nozzle. (43-024-855)

6 in. in diameter. This gave an outlet area of 0.97 sq ft. Outlet discs of 4-in. and 5-in. diameter were fabricated at this time also, for trial at a later date.

The modified separator model was tested at 1300 gpm with about 40 cfm of air entering the separator. A total separator head loss of 3.1 ft was measured, with 1.6 ft occurring across the inlet nozzle. This configuration was considered adequate until more extensive tests proved otherwise. It was expected that the final dimensions would be dictated by a balance of maximum separator efficiency with total head loss. As the inlet nozzle is modified to give higher separating forces by virtue of higher inlet velocities, the inlet head loss increases. Higher centrifugal velocities also promote a longer vortex, so that a limit is reached when the outlet nozzle can no longer "pinch off" the vortex. The result is high carryunder. As the bottom of the vortex comes closer to the outlet nozzle, the rotational velocities are higher. Thus, because of the high velocity vector angle at the outlet plane, there is a decrease in effective outlet area. This, naturally, causes greater outlet nozzle head loss.

A.2. Initial tests to set operating characteristics over reactor flow range...During the preliminary shakedown tests on the test loop and 1st separator model, the model was accidentally damaged. Two weeks of testing had been accomplished, with data being taken on pressure drop and accuracy of the flow measuring orifices being checked. The model was damaged when screws holding the adjustable inlet came loose. The inlet nozzle closed and the full discharge head from the pump was applied across the model. This sudden force caused it to collapse.

A new separator model was fabricated from 16 gauge galvanized sheet metal. The new model was dimensionally the same as the first model except for these changes:

1. When the smashed model was removed, measurement of the inlet length showed that it was 59 in. long instead of the 54 in. specified. The inlet width on the new model was increased to $2\frac{5}{16}$ in. and had the correct 54 in. length. This gave the same inlet area as the first model.
2. Observation of the outlet nozzle of the first model showed that it vibrated at high flows because the narrow outlet strips were not stiff enough. On the new model, six triangle-shaped openings were specified and a 5-in. disc was used to close off the bottom. The new model outlet nozzle had the same outlet area as the first model.

Testing on the new model showed that total separator head loss at rated flows was 3.2 ft., splitting 50-50 between inlet and outlet nozzles.

The split of supply air flow between separator inlet and steam dome area was the subject of much scrutiny during early tests. As indicated by the outlet air orifices, this split oscillated between 20 and 50 per cent from the separator at rated conditions. It was expected that the amount of air carried into the separator would be a function of water flow rate only, if the other physical parameters (pressure, temperature, etc.) were held constant. But the oscillation still occurred even with all the loop variables held constant. It was thought that uneven, permanent pressure loss in the exhaust-air measuring orifices was causing pressure to build up, possibly in the separator, when 50 per cent of supply air passed into it.

Then, when pressure in the separator air space exceeded steam dome pressure, more air would flow from the steam dome exhaust, causing a pressure buildup in this area, and a resulting swing in the other direction. Static pressure taps were installed in the separator air space and in the steam dome areas to check this theory.

At rated flows of 1330 gpm and about 190 cfm air, the pressure in the separator was found to be about 1-in. Hg higher than steam dome pressure when 50 per cent of the supply air was passed. The differential would switch in the other direction when air flow from the separator dropped to 20 per cent. Two individual throttle valves were installed in the exhaust lines to see if the pressure could be equalized by throttling one or the other, thus obtaining a constant split of supply air. The pressures in the two areas could be controlled by means of these valves but, despite the many combinations of throttling attempted, the oscillation of flow from the separator persisted. In fact, it got worse.

In an attempt to eliminate any differences in exhaust pressure losses between separator and steam dome, the piping was removed from these areas. The test loop was then run and the fluctuation of inlet pressure drop was used as an indication of any oscillation of air flow into the separator. This fluctuation occurred because of higher drop when two-phase flow was present in greater or lesser amounts. The tests indicated that the same oscillation of air flow occurred even though pressures in steam dome and separator were equal.

Attempts to stabilize air flow to the separator continued. The next step was to narrow the inlet slit to 1.8 in. With higher inlet velocities, it was reasoned that there would be no opportunity for air to be held in the

downcomer, thus avoiding air holdup that might be causing oscillation. The test results indicated some improvement in stability, and pressure drop at rated air-and-water flow rates increased to 3.7 ft. H₂O.

Data obtained after this modification indicated that, at design flow rates, the percentage of air from the separator varied only ± 5 per cent from an average value (roughly 50 to 60 per cent from the separator). The stability improvement was partially attributed to the fact that the loop was pressurized to 5 psig for the first time. With a higher total pressure in the loop, the per cent variation between separator and steam dome area was smaller. Several values of exhaust air measurement were taken at each data point to cover the full range of fluctuation and to get an average reading covering about 5 minutes of constant loop conditions.

Previous to the study of fluctuation in the separator air exhaust, a test was run to determine the effect of changing water level on separator characteristics. The new model with a 2-5/16 in. inlet width was run at 1330 gpm and 190 cfm supply air, with the water level being varied. This was done to verify the earlier estimate of the required separating length below the separator inlet nozzle. Fig. 34 is a plot of carryunder volume fraction vs static water level above the separator inlet. Data from the test showed that separator efficiency stayed practically constant, passing less than 0.2 per cent carryunder down to 1 ft of static cover. This was well within the design limit of 1 per cent by volume allowable carryunder. Reactor water level could never be allowed to fall as low as 1 ft above the fuel, for reasons of shielding and reflecting. The per cent of supply air passing into the separator stayed constant over the entire level range at about 55 per cent (~ 110 cfm).

VOLUME FRACTION % CARRY UNDER $\cdot \frac{\text{VOL. AIR CARRY UNDER (100)}}{\text{VOL. H}_2\text{O FLOW} + \text{VOL. AIR CARRY UNDER}}$

Fig. 34...Volume fraction per cent carryunder vs water level above separator inlet...water flow rate - 1330 gpm constant...air to test loop - 200 cfm(s) with average 55 per cent air into separator. Data runs 52 through 57. Water temperature $\approx$ 100 F. (43-025-092)

Later, the separator was removed from the loop and the 5-in. disc on the bottom of the outlet nozzle was replaced by a 4-in. diameter disc. The immediate tests that followed this change indicated that there was a small improvement in carryunder characteristics. Total pressure drop ran about 0.7 ft H_2O higher than before. Fig. 35 shows the separator model with all modifications made to this point.

The next phase of the program involved extensive testing under a wide variety of conditions, to define the operating characteristics of the separator for future comparisons. Water flow rates 600, 900, 1000, 1200, 1330 (design point), and 1500 gpm were set as data points. Air flow to the loop was maintained at the following rates: 30, 60, 90, 120, 145, and 175 cfm free air. This gave upcomer void volume fractions in a range from 13.0 per cent to 68.7 per cent. Most of the data was taken with a static water level 4 ft above the top of the separator inlet. Since velocity and gravitational effects in the test loop duplicated reactor effects as near as possible, it is assumed that the same relative water-void interface will be reached in the Pathfinder reactor under similar flow conditions. The curves for this test phase are included in the Appendix, curves A-1 to A-5, inclusive.

Total separator head loss at rated flow was 4.4 ft H_2O at the rated point. The increase over earlier values was apparently caused by the smaller outlet nozzle disc.

With the static water level at 4 ft above the separator inlet, carryunder at the rated point was 0.213 per cent by volume. The per cent of supply air (175 cfm, free) which passed into the separator was 62 per cent.

Fig. 35...Full size 10-in. diameter Pathfinder steam separator model. Inlet nozzle: 54-in. long and 1.8-in. wide. Outlet nozzle: Six triangle-shaped slits, 24-in. long. 4-in. diameter disc on the bottom.

With the static level at 2 ft above the separator inlet, carryunder was 0.3 to 0.4 per cent by volume. At the same rated point, the per cent of supply air passing into the separator was about 55 per cent.

The plots of carryunder and per cent of air into the separator for the 4 ft water level showed quite a bit of scatter, even though several readings were taken at each data point and averaged. The pressure drop curves showed the expected squared relationship with flow, except in the area above 1100 gpm water flow rate. There the air passing into the separator became appreciable, causing an added drop which is expected in two-phase flow. Because of the previously mentioned scatter and the unexpectedly high per cent of supply air carried into the separator, succeeding tests were aimed at detecting the causes of these phenomena. It was also desired to change flow paths with baffles, in order to observe the effect. Tests were therefore planned, utilizing various size screens, baffle plates and vanes. These were designed to study the effect of bubble size (screens), turbulence (vanes) and to artificially change the stream lines in the area where air-water flow crossed over to the downcomer.

A-3. Tests using screens over the baffle sleeve outlet...These tests were conducted in a manner similar to that described for the initial tests. The same air and water flow rate range was covered.

The purpose of these tests was to determine the effect of changing the mixture outlet configuration. This would give an indication of the change in per cent of supply air into the separator which is caused by: (1) Change in bubble size in the mixture and (2) Elimination of channeling or "jetting" of the mixture from the baffle sleeve outlet.

Tests were run at the standard loop conditions of 5 psig and 175F. Measurements were taken of per cent of air entering the separator, per cent by volume of carryunder, and separator pressure drop. To simulate a progressive change in bubble size, testing was done with a No. 6 mesh screen and also a No. 10 mesh screen over the bubble sleeve outlet (see Fig. 36). The No. 6 mesh screen had a nominal aperture of .1320 in. and the No. 10 mesh screen an aperture of .0787 in. For the sake of comparison, data for one air flow was plotted along with data from tests without a screen. Data was taken with the static water level 4 ft and also 2 ft above the top of the separator inlet nozzle. All data from these tests with screens are included in Appendix "B" of this report.

Fig. 37 shows a plot of percentage of air carried into the separator for the three conditions (No. 6 screen, No. 10 screen, and no screen) as a function of recirculation water flow rate, with 145 cfm free air supplied to the loop. It was expected that, as the screen size varied, the curves would show a progressive change in percentage of air into the separator. It can be seen from Fig. 37, however, that no distinguishable progressive change took place. The "no screen" curve falls between the curves for the No. 6 and No. 10 screens. It can be safely said that there was no noticeable difference in the percentage of air carried into the separator under the three different test conditions.

Fig. 38 is a plot of per cent carryunder vs water flow for the same three test conditions. These curves also indicate no progressive change in function as the screen size was varied. The curves have the same relative position as previously shown in Fig. 37 (per cent air into separator vs water flow rate).

Fig. 36...Cutaway view of steam separator test section.
(43-002-199)

Fig. 37...Per cent of supply air from separator vs water flow rate for 145 cfm air supply. 4-ft static water level, no vane, for various screen sizes.
 (43-024-234)

Fig. 38...Per cent carryunder vs water flow rate for 145 cfm air for various screen sizes. 4-ft static water level, no vane. (43-024-235)

Figures 39 and 40 show carryunder plotted vs air supplied to the loop and air entering the separator, respectively. These curves, too, show no conclusive change in function with variation of screen size. It is interesting to note that Fig. 40 gives the same curve shape as the plotting for "separator entering void fraction vs carryunder." The slope of this curve shows that the separator is less sensitive to changes in "entering air flow" than it is to changes in "water flow rate," as indicated by the slope of the curves.

The curves of total separator head loss vs water flow rate are shown in Section "B" of the Appendix (curves B-1 to B-16, incl.), for both 4 ft and 2 ft static water levels. As would be expected, these curves show only the relative effect of more air entering the separator, since the separator nozzle configurations were the same for all these tests.

A comparison of the curves with 2 ft static water level and curves with the 4 ft level shows that the curves of per cent air into the separator have very similar configurations. However, when the curves of carryunder vs total air supplied or vs air from the separator are compared for 2 ft and 4 ft levels with the No. 6 mesh screen (B-3, B-4 and B-7 in the Appendix), it is seen that the 2 ft level curves turn up sharply at the highest entering void fractions for any water flow rate. This can be explained by the fact that, at higher inlet velocities, the depth of the vortex increases until it begins to push out of the separator discharge nozzle. The same phenomenon does not occur with a 4 ft static water level. In that case, the top of the vortex is oriented at a higher level, and the bottom of the vortex is relatively two ft higher at the same velocities than it would be for the 2 ft static water level.

Fig. 39...Per cent carryunder vs total air supplied to loop for 1330 gpm water flow and various screen sizes. 4-ft static water level, no vanes.
 (43-025-090)

Fig. 40...Per cent carryunder vs air from separator for 1330 gpm water flow rate and various screen sizes. 4-ft static water level, no vanes.
(43-025-089)

A-4. Conclusions reached from tests with screens... Since the data taken with various screen configurations produced no conclusive proof of a bubble size effect, it was concluded that if such an effect existed it was overshadowed by the centrifugal action of the separator. Observations of the flow patterns in the standpipe, however, showed that the turbulence above the downcomer sometimes took the form of a vortex. The vortex would alternately form and collapse, and would carry under more than the usual amount of air.

It was suspected that, because of the circular configuration of the standpipe, these vortices might be causing the erratic data on air carried into the separator. Since the downcomer area in the reactor is an annulus, it was assumed that this form of turbulence would not exist there. The next phase of testing was, therefore, concentrated on breaking up the observed vortex formation above the downcomer through the use of flow-straightening vanes.

A-5. Test of an inclined perforated flow-deflection plate for effect on separator carryunder performance... While these flow-straightening vanes were being fabricated, a test was run to determine the effect of a flow-deflection baffle designed for installation above the boiling core in the reactor. Purpose of this baffle was to aid in leveling out the two-phase interface above the boiling core.

A summary of an air-water test run on various flow deflector types in a transparent test loop can be found in Report ACNP-6105 by Wilson and Grenda (Ref. 7). The flow baffle finally decided upon was a perforated plate to be inclined at 45° with the horizontal, and to be installed with its centerline 2 ft above the reactor core. This is shown in Fig. 1, the

reactor cross section drawing.

The baffle plate fabricated for the steam separator test was made from 1/8-in. thick steel plate. About 30 per cent of the plate area was removed by perforating the plate with 1-in. diameter holes (Fig. 41). Fig. 42 shows the plate installed in the standpipe.

The steam separator model was tested with the inclined perforated baffle plate in two different positions and at 45° and 60° from the horizontal in each position. Fig. 43 illustrates the two positions used. Position "A", centered 2 ft above the baffle sleeve outlet, duplicates the proposed position of the baffle plate in the Pathfinder reactor. Position "B", with the top of the baffle plate even with the static water level (4 ft above the baffle sleeve outlet), was used for a comparison of the relative effect of the baffle on separator operation. The No. 6 mesh screen remained in place over the baffle sleeve outlet during this test.

Since the only purpose of this test was to get a general idea of effect on separator performance, the test was run at two air flow rates (75 cfm and 145 cfm) and at three water flow rates at each of these two air flow rates. The run was repeated for each baffle plate position and angle.

A-6. Results of tests with perforated, inclined baffle plates...The most significant result of the inclined baffle plate test is shown in Fig. 44, a plot of per cent of total supply air from the separator vs water flow rate for 145 cfm total air supplied to the loop. The curves for both angles in position "B", and for the 60° angle in position "A", compare closely with data for the No. 6 mesh screen with no inclined baffle plate. The curve for the 45° angle in position "A" remained high (at about 85 per cent of total air going into the separator) over the entire water flow rate range.

Fig. 41...Inclined perforated baffle plate showing fabrication.

Fig. 42...Inclined perforated baffle plate as installed in Steam Separator Test Loop. View looking down into standpipe.

Fig. 43...Positions of inclined
 perforated baffle in steam
 separator test loop.
 (43-024-880)

Fig. 44...Per cent air from separator vs water flow rate with baffle plates at positions A and B (see Fig. 43). 145 cfm air supply, 4-ft static water level, No. 6 screen and no vanes. Test runs No. 145 through No. 169. (43-025-096)

At this particular plate angle and mixture flow rate, the stream lines were evidently deflected. The result was that circulation of the mixture into the downcomer took place under the baffle plate, with an essentially static pool existing above the baffle plate. Fig. 45 shows that this condition was not reached with a lower mixture flow rate of 75 cfm total air supplied.

In order to graphically determine if the added per cent of air into the separator had any effect on carryunder, Fig. 46 was plotted. This plot (per cent by volume carryunder vs water flow rate, 145 cfm total air supplied) came very near to, or lower than, similar data for No. 6 mesh screen with no baffle plate. For comparison, Fig. 46 shows a curve of the "no baffle plate" data for the 145 cfm air supply flow rate. The data for both positions and both baffle plate angles plotted in a very narrow range. One curve could be drawn through the data. The high per cent of supply air which entered the separator during the "position A , 45° angle" test had no effect on the carryunder characteristics of the separator.

A-7. Conclusions reached from tests with perforated, inclined baffle plates... It was concluded that the presence of the inclined baffle in the reactor would have no adverse effect on operation of the steam separators. If a large percentage of the steam was deflected into the separators, however, this would not be an efficient use of the available steam release area. Later tests by Wilson and Grenda (Ref. 7) on horizontal perforated baffle plates in steam water showed that these produced a more quiescent surface than was achieved with the inclined baffle. Since there also were some inherent fabrication problems with the inclined baffle, it was subsequently decided to use horizontal perforated baffles in the Pathfinder reactor. They are to be placed above the boiling core at a distance which is expected to have no effect on separator operation.

Fig. 45...Per cent of supply air from separator vs water flow rate with baffle plates at positions A and B (see Fig. 43). 75 cfm air supply, 4-ft static water level, No. 6 screen and no vanes. Test runs 145 through 169. (43-025-095)

Fig. 46...Per cent carryunder vs water flow rate for 145 cfm air supply flow rates. 4-ft static water level, No. 6 mesh screen, no vanes. Inclined perforated baffle test conditions as indicated in drawing. Test runs 145 through 169. (43-025-088)

A-8. Tests using flow-straightening vanes in the downcomer region...

As previously mentioned, it was found desirable to run tests on the separator model with straightening vanes in the downcomer, in order to break up the vortex formation that was being observed. These fluctuating vortices carried under a large volume of the air which was present above the downcomer.

Two different vane configurations (single vane and multi-vane) were tried. All screens were removed from the baffle sleeve outlet. Fig. 47, A and B, shows sketches of the vanes installed in the downcomer. A final test was then made with a No. 16 mesh screen attached over the baffle sleeve outlet.

A-9. Results of tests with flow-straightening vanes... Fig. 48 shows the comparative curves of per cent of supply air (145 cfm) which entered the separator under these conditions: no vanes, single vane, multi-vane, and multi-vane with No. 16 screen. Data for single vane and multi-vane tests were almost identical. The curve for no vanes or screens shows that a much higher per cent of supply air entered the separator for that condition. The curve also shows that the percentage of total air entering the separator at the reactor recirculation rate of 1330 gpm decreased from 45 per cent to about 20 per cent when the vanes were added. In addition, observation showed that turbulence (in the form of alternately forming vortices) was not now present in the region above the downcomer.

The curve for data with the addition of a No. 16 screen falls between the "no vane" data and the "no screen with vane" data. It was originally thought that the smaller mesh screen might show more of an effect on bubble

FIGURE 47-B MULTI-VANE CONFIGURATION

Fig. 47...Single vane and
multivane configurations in
the downcomer region.
(43-024-878)

FIGURE 47-A SINGLE VANE CONFIGURATION

Fig. 48...Per cent of supply air from separator vs water flow rate for 145 cfm air. No vane, single vane and multivane. 4-ft static water level. (43-024-233)

size than the larger mesh screens previously tested. However, observation disclosed that a more noticeable effect of this screen was the spreading of entering air over a bigger area above the downcomer. Without the screen, the air left the baffle sleeve outlet in the form of a smaller, high velocity stream. But with the air stream spread out, the air bubbles followed the stream lines of water flow more closely - and more was carried into the downcomer. Without the screen, the flow of air would rise to separate at the free surface. Because of the vanes, there was no vortical turbulence to carry it back into the downcomer. With the No. 16 mesh screen installed and the air stream spread out, it logically followed that more air entered the separator.

A.10. Conclusions reached on test loop configurations...At this point in the program, some speculation was made as to which of the test loop configurations would be closest to duplicating the gravitational and frictional conditions existing in the Pathfinder reactor. The outlet from Pathfinder's boiler core fuel elements is made up of 81 fuel rods, .342-in. in diameter, evenly spaced in a 5-in. x 5-in. box. Comparing this outlet pattern to those tried in the loop, it was speculated that the mixture flow distribution would be somewhere between that obtained with the No. 16 mesh screen and multi-vane, and the multivane with no screen. Thus, at 1330 gpm per separator, a percentage somewhere between 45 per cent and 20 per cent of the steam produced (at full power) would be carried into the separators.

The effect of these different configurations on the carryunder characteristics of the separator is interesting. Fig. 49 (per cent by volume carryunder vs water flow rate) shows data for the same configurations

Fig. 49...Per cent carryunder vs water flow rate for 145 cfm air supply flow rate. Various vane configurations, 4-ft static water level. (43-024-231)

as Fig. 48. The data obtained for the configurations were almost identical. Even though the volume of air entering the separator more than doubled, the carryunder increased only 29 per cent (from .092 per cent by volume to .119 per cent). This is an extremely low amount of carryunder.

Although it is difficult to predict the exact configuration which would duplicate the Pathfinder reactor flow conditions, it is believed that the "no-screen, multi-vane" configuration is the most representative. On this basis, most of the analysis of separator characteristics will be confined to the data on this configuration.

B. Discussion of Results from Test of Final Separator Model and Test Loop Configuration (no screen, multi-vane flow straightening device in the downcomer...Runs No. 438 through No. 481).

B.1. Per cent of supply air into separator vs water flow rate.

This is plotted in Fig. 50 for the final configuration. Below 1100 gpm recirculation flow, the effect of increasing downcomer velocity is almost a single line for all supply air flow rates, indicating that the percentage into the separator was not affected by upcomer void fraction up to this point. Beyond 1100 gpm, however, the curves spread apart on the basis of increasing upcomer void-to-mixture volume ratio, indicating its relative effect. As more and more bubbles are carried into the downcomer, the velocity increases. With the water flow constant, more bubbles are carried down. This effect of increased velocity in the downcomer is shown in the curves.

The curve for 30 cfm air supplied to the loop is the only data which deviates from this pattern. A possible reason for the larger percentage of the 30 cfm air flow being carried down is the fact that there was less

Fig. 50...Per cent of supply air from separator vs water flow rate for various air supply flow rates. 4-ft static water level, no screen, multivane. Test runs 438 through 481. (43-025-087)

vertical jetting of the mixture from the baffle sleeve outlet. The resultant mixture velocity vector would be at a more nearly horizontal angle than for higher mixture flow rates--(higher supply air flow rates at the same water flow rate). Therefore, at 30 cfm the air did not rise out of the realm of influence of downcomer flow, and a larger percentage of it was carried into the separator.

B.2. Per cent by volume carryunder vs water flow rate for various air supply flow rates...(Fig. 51).

These curves of carryunder voids vs recirculation flow rate are displaced vertically in the order of increased upcomer void-to-mixture volume ratio (supply air flow rate). This characteristic of the separator is of value because it simulates quantities which can be measured in the Pathfinder reactor system for evaluation. From this curve, an idea can be obtained as to expected carryunder at any reactor recirculation flow rate and reactor power level (steam flow rate). The effect of gas flow rates on carryunder is shown in later sections of this data analysis.

B.3. Per cent by volume carryunder vs air supply flow rate for various water flow rates...(Fig. 52).

This function of carryunder shows the effect of increasing riser voids on expected carryunder. In fact, the plot of carryunder vs upcomer void-to-mixture volumetric flow ratio would have the same shape, since each curve is for constant recirculation water flow rate.

The curves show how carryunder would vary in the reactor if recirculation flow rate was held constant and the power level (steam flow rate) was varied.

Fig. 51...Per cent carryunder vs water flow rate for various air supply flow rates.
 4-ft static water level, no screen, multivane. Test runs 438 through 481.
 (43-025-086)

Fig. 52...Per cent carryunder vs total air supplied to loop for various water flow rates. 4-ft static water level, no screen, multivane. Test runs 438 through 481. (43-025-082)

It is quite evident that carryunder is much less a function of riser voids than it is of recirculation water flow rate.

As was noted on Fig. 50, more air is carried into the separator as the supply-air-flow rate is increased. Downcomer velocity becomes greater as air is carried into the downcomer. This effect is also reflected in Fig. 52. It can be seen that the curves are almost flat up to 1100 gpm. At water flow rates higher than 1100 gpm, the carryunder function increases with the increasing of the air supply flow rate above 90 cfm. Although the carryunder grows with the increase of air supply flow, the curves level out again when maximum possible per cent of supply air (for that water flow rate) has been carried into the downcomer.

B-4. Per cent by volume carryunder vs cfm air into the separator for various water flow rates...(Fig. 53).

Carryunder as a function of air entering the separator was plotted in an attempt to define separator efficiency.

NOTE: At this point, some mention should be made as to the method of calculating cfm of air flow into the separator. Air flow rates from the separator exhaust and from the pool area around the separator were measured separately in terms of saturated air at 175F.

Ratio of these flow rates $\times 100$ was the basis for obtaining the per cent of supply air entering the separator. This was done to get a parameter which could be applied to any upcomer flow rate (reactor or test loop). However, in order to find out what the volumetric flow rate is at the entrance to the separator, some estimate must be made as to the degree of saturation that the air has reached.

Analytical methods of determining the degree of saturation are difficult, if not almost impossible. To do that, one must assume an air flow configuration to get an area of bubble exposure to the hot water, and then make some estimates as to the heat transfer and diffusion rates through the bubble wall.

Fig. 53...Per cent carryunder vs air flow rate from separator for various water flow rates. 4-ft static water level, no screen, multivane. Test runs 438 through 481. (43-025-080)

The supply air flow rate was known in terms of free air. Calculations on the basis of perfect gas laws showed that, if the air entering the separator was 100 per cent saturated at the pressure and temperature of the separator inlet, the volume flow rate would be 22 per cent higher than that of free air.

Since it was not known what per cent saturation was attained, a conservative estimate of separator capacity was required. Therefore, the values of air entering the separator (cfm) were assumed to be free air, calculated by taking the "per cent of supply air from the separator multiplied by the supply air flow rate". This is the value plotted in Fig. 53.

Actually, the air flow rate measured from the separator is not the amount which entered the separator since some of this air was carried under. The measured volume flow rate of air carryunder, however, was never more than 0.3 cfm of air, so this omission will have little effect on the calculation. Air carried under was measured at the pressure and temperature of the separator outlet. Therefore, the per cent of moisture saturation of this air does not affect the calculation of per cent carryunder (void-to-mixture volume flow ratio). The measured amount of air into the separator may not be the actual amount, but since pressure and temperature were held constant, the actual air should vary by a constant amount from that measured. The trend of the plot, therefore, should be the same as if the actual volume of air into the separator was measured.

Fig. 53 shows a rapid increase in carryunder with the increase of air into the separator - up to 20 cfm. After this point, and also at higher water flow rates, the curves level off. Thus, carryunder is not a significant function of volume of air entering the separator. Above 1200 gpm water flow rate, carryunder is fairly constant at constant water flow rate, even though air into the separator increases.

Although this plot shows the function of carryunder with void-handling capacity of the separator, a plot was desired which would tell more about the relative efficiency of the separator as a unit over the operating range.

B-5. Separator efficiency vs water flow rate for various air supply flow rates...(Fig. 54).

A separator efficiency, per cent, was defined as

$$\left(1 - \frac{Q_{\text{air carried under}}}{Q_{\text{air into separator}}}\right) 100,$$

where the numerator and denominator of the fraction are cfm air carried under and cfm air into the separator, respectively. The cfm air into separator was calculated by adding the cfm air carryunder to the air flow from the separator exhaust (as per cent of cfm supply free air).

Fig. 54 shows the efficiency of the separator over the whole range of operating air and water flow rates. In order to show the trend of the data, the efficiency scale had to be expanded so that only a 2 per cent spread is shown. It should be noted that, even at the lowest point, efficiency does not drop below 98.3 per cent.

The efficiency curves show a dip which occurs at about 1200 gpm for 175, 145, and 120 cfm air supply flow rate and at progressively higher water flow rates for 90 and 60 cfm. Figures 50 and 51 show the reason for this dip. Figure 50 shows that the per cent of supply air into the separator increases slightly with increasing water flow rate, up to about 1200 gpm. After 1200 gpm, there is a sharp increase in the per cent of supply air into the separator. Figure 51 shows that the rate of increase in carryunder with increasing water flow rate is fairly constant over the whole water flow rate range. However, up to 1200 gpm the rate of increase

Fig. 54...Separator efficiency vs water flow rate for various air supply flow rates. 4-ft static water level, no screen, multivane in downcomer. Test runs 438 through 481. (43-025-079)

in carryunder with water flow is faster than the increase in the per cent of supply air into the separator. This explains the decline of efficiency up to 1200 gpm. Since $Q_{\text{air carryunder}}$ is increasing faster than $Q_{\text{air into separator}}$, the fraction is getting larger and the efficiency function decreases. Above 1200 gpm (see Fig. 50), air into the separator is increasing at a faster rate, the fraction $\frac{Q_{\text{air c.u.}}}{Q_{\text{air into separator}}}$ is becoming smaller, and the efficiency function increases.

As noted earlier, the estimated actual amount of air into the separator is a conservative figure. Therefore the actual efficiency is, in general, somewhat higher than Fig. 54 indicates.

B-6. Total separator head loss vs water flow rate for various air supply flow rates...(Fig. 55).

This curve is a typical plot of head loss vs water flow rate for the separator, regardless of loop configuration. After initial modifications to the separator model, the separator profile was not changed for the balance of the testing. This head loss curve can therefore be considered typical.

The plot shows, again, the effect of more air entering the separator, in this case, above 1100 gpm. The curves fan out after this flow rate and distribute themselves in the order of increasing air supply flow rate. This is because of the added two-phase pressure drop where air entered the separator.

Fig. 55...Total separator head loss vs water flow rate for various air supply flow rates. 4-ft static water level, no screen, multivane in downcomer. Test runs 438 through 481. (43-025-076)

8. TEST PROGRAM CONCLUSIONS

To start the program, some testing was done and a literature review was made to ascertain the type of primary separator most applicable to the Pathfinder reactor. The crossover type and certain conventional upcomer type separators were eliminated because of too high a head loss at required flows, or because of re-entrainment at the separator water outlets. These types of separators also conflicted with above-the-core control rod drives in the Pathfinder reactor.

Testing was done on several upcomer types, some of which showed promise, but the problems of head loss and re-entrainment also eliminated the upcomer types.

It was concluded that the downcomer type of separator was most suitable. Preliminary tests had shown them to have high efficiency with moderate head loss over a substantial change in outside water level. Their position and space requirements were quite suitable to Pathfinder design.

Fig. 56 is a sketch of the final Pathfinder separator model and Fig. 57 is a photo of this model. The separator is 10 in. in diameter and about 14.5 ft. long. It is fabricated from No. 12 gage (.109 in.) stainless steel (304) sheet. The separators will be assembled in groups of three and welded together (Fig. 58). These assemblies will then be installed in the Pathfinder reactor with the inlet nozzles facing the core. Plans call for 30 separators in the outside row and 15 separators in the row nearest to the core. The separator support shelf has openings with an offset lip into which the separator support rings fit. This ensures a tight fit so that all the water passes through the separators.

Fig. 56...Sketch of final Pathfinder steam separator model, 10-in. O.D. (43-301-534)

Fig. 57...Final Pathfinder separator model.

NO.	DESCRIPTION	MATERIAL	QTY.	UNIT	WGT.
1	STEAM SEPARATOR GROUP ASSEMBLY		1	ASSEMBLY	65.20
2	STEAM SEPARATOR		3	SEPARATORS	15.00
3	SUPPORT SHELL		1	SHELL	1.00
4	SPACER PLATE		1	PLATE	0.50
5	TEST TUBE		1	TUBE	0.50
6	PIN		2	PINS	0.10
7	WELD ROD (ASTM A518-55T CLC.300 $\frac{3}{16}$)	ACI 119	7	RODS	0.10
8	FABRICATION SPEC.		1	SPEC.	0.01
9	MARKING SPEC.		1	SPEC.	0.01
10	CLEANING SPEC.		1	SPEC.	0.01
11	PENETRANT INSPEC. SPEC.		1	SPEC.	0.01
12	WELD PROC. SPEC.		1	SPEC.	0.01
13	PICKLING PROC. SPEC.		1	SPEC.	0.01
14	LUG		1	LUG	0.50
15	BRACE		1	BRACE	0.50
16	WELD PROC. SPEC.		1	SPEC.	0.01
17	PICKLING PROC. SPEC.		1	SPEC.	0.01

Fig. 58...Steam separator group assembly. (43-500-933)

Design Characteristics (attainment of design objectives)...Separator carry-under (amount of voids passing through the separator water outlet) is the most important characteristic from the standpoint of effect on reactor operation. The correlations of Peebles (Ref. 4 and 5) offered a method of illustrating the effects of the physical properties of steam-water at 489F (saturated) on separation. A hydro-dynamic analysis of the separating forces in the separator was made, taking into consideration the bubble size, separator tangential and axial velocity profiles. On this basis, a required separating length for bubbles entering at the lowest possible point was calculated for the reactor conditions. This was found to be 1 ft below the inlet nozzle at rated flow.

Air-water tests showed the amount of carryunder to be negligible at rated conditions. In fact, the amount would not have been measurable at all except for the measuring method used in the tests. At the rated point (1330 gpm), 0.1 per cent by volume carryunder was measured (0.18 cfm per separator). This is a factor of ten lower than the design limit of 1 per cent.

The split of upcomer voids between downcomer and the free surface is of interest, not only for its effect on separation, but also for the most efficient use of available free surface. In early tests, vorticial turbulence above the downcomer caused 50 to 60 per cent of upcomer voids to enter the separator. Flow-straightening devices were added to the loop to make the flow path more similar to Pathfinder reactor conditions. Under these reactor conditions, 25 to 40 per cent of the upcomer voids will enter the separator at rated flow, depending upon the shape of the core outlet. The separator outlet pipes occupy only 2.5 per cent of the available free surface

area in the handling of this 25 to 40 per cent of total voids.

With the design characteristics as outlined, total head loss through the separator was 3.7 ft at the rated water flow.

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A P P E N D I X

Pathfinder Steam Separator Test Curves

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NOTE: The following tests were run with a multi vane in the downcomer, but with a No. 16 mesh screen over the upcomer (baffle-sleeve outlet). They are included in the section on vanes because the comparative effect is more evident.

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Appendix - Section A

Curves: A-1 through A-5

Initial tests...no screens and no vanes.

Test Conditions:

Air-Water

Water Temperature - 175 F

Standpipe Pressure - 5 psig

Test Runs:

No. 270 through No. 309

Test File No. 2-4300-00364

43-024-938

A-1...Per cent carry under vs water flow rate for various air flows. 4-ft static water level, no screens, no vanes.
(Test runs 270 through 309.)

A-2...Per cent air from separator vs water flow rate for various air flow rates. 4-ft static water level, no screens, no vanes. (Test runs 270 through 309.)

A-3...Per cent carryunder vs total air supplied to loop for various water flow rates. 4-ft static water level, screens, no vanes. (Test runs 270 through 309.)

A-4...Per cent carryunder vs air from separator for various water flow rates. 4-ft static water level, no screens, no vanes. (Test runs 270 through 309.)

43-024-989

A-5...Total head loss vs water flow rate for various air flow rates. 4-ft static water level, no screens, no vanes.
 (Test runs 270 through 309.)

Appendix - Section B

Curves: B-1 through B-16

Tests with No. 6 mesh and No. 10 mesh
screens over baffle sleeve outlet.

Test Conditions:

Air-Water

Water temperature - 175 F

Standpipe Pressure - 5 psig

Test Runs:

B-1 through B-8...Runs No. 113 through
No. 144 for 4 ft static water level.
...Runs No. 170 through
No. 194 for 2 ft static water level.

B-9 through B-16...Runs No. 199 through
No. 229 for 4 ft static water level.
...Runs No. 234 through
No. 269 for 2 ft static water level.

Test File No. 2-4300-00364

B-1...Per cent carryunder vs water flow rate for various air flow rates. No. 6 mesh screen, 4-ft and 2-ft static water levels for comparison, no vanes. (Test runs 113 through 144 for 4-ft level, 170 through 194 for 2-ft level.)

B-2...Per cent of supply air from separator vs water flow rate for various air flow rates. No. 6 mesh screen, 4-ft static water level, no vanes. (Test runs 113 through 144.)

B-3...Per cent carryunder vs total air supplied to loop for various water flow rates. 4-ft and 2-ft static water levels for comparison. No. 6 mesh screen, no vanes. (Test runs 113 through 144 for 4-ft level, 170 through 194 for 2-ft level.)

B-4...Per cent carryunder vs air from separator for various water flow rates. 4-ft water level, No. 6 mesh screen, no vanes. (Test runs 113 through 144.)

43-024-990

B-5...Total head loss vs water flow rate for various air flow rates. 4-ft static water level, No. 6 mesh screen, no vanes. (Test runs 113 through 144.)

43-024-948

B-6...Per cent of supply air from separator vs water flow rate for various air flow rates. 2-ft static water level, No. 6 mesh screen, no vanes. (Test runs 170 through 194.)

B-7...Per cent carryunder vs air flow rate from separators for various water flow rates. 2-ft static water level, No. 6 mesh screen, no vanes. (Test runs 170 through 194.)

43-024-991

B-8...Total separator head loss vs water flow rate for various air flow rates. 2-ft static water level, No. 6 mesh screen, no vanes. (Test runs 170 through 194.)

43-024-949

B-9...Per cent carryunder vs water flow rate for various air flow rates. 4-ft static water level, No. 10 screen, no vanes. (Test runs 199 through 229.)

43-024-950

B-10...Per cent of supply air from separator vs water flow rate for various air supply flow rates. 4-ft static water level, No. 10 screen, no vanes. (Test runs 199 through 229.)

B-II...Per cent carryunder vs total air flow rate supplied to loop for various water flow rates. 4-ft and 2-ft water levels for comparison, No. 10 mesh screen, no vanes. (Test runs 199 through 229 for 4-ft, 234 through 269 for 2-ft.)

B-12...Per cent carryunder vs air flow rate from separator for various water flow rates. 4-ft and 2-ft static water levels for comparison, No. 10 screen, no vanes. (Test runs 199 through 229 for 4-ft, 234 through 269 for 2-ft.)

43-024-992

B-13...Total separator head loss vs water flow rate for various air supply flow rates. 4-ft static water level, No. 10 mesh screen, no vanes. (Test runs 199 through 229.)

B-14...Per cent carryunder vs water flow rate for various air supply flow rates. 2-ft static water level, No. 10 screen, no vanes. (Test runs 234 through 269.)

B-15...Per cent of supply air from separator vs water flow rate for various air supply flow rates. 2-ft static water level, No. 10 mesh screen, no vanes. (Test runs 234 through 269.)

43-024-993

B-16...Total separator head loss vs water flow rate for various air supply flow rates. 2-ft static water level, No. 10 mesh screen, no vanes. (Test runs 234 through 269.)

Appendix - Section C

Curves: C-1 through C-15 .

Tests with flow-straightening vanos
in downcomer region.

Test conditions:

Air-Water

Water Temperature - 175 F

Standpipe Pressure - 5 psig

Test runs:

C-1 through C-5...

Runs No. 358 through No. 396

C-6 through C-10...

Runs No. 482 through No. 535

C-11 through C-15...

Runs No. 536 through 549

Test File No. 2-4300-00364

C-1...Per cent by volume carryunder vs water flow rate for various air supply flow rates. 4-ft static water level no screen, one vane. (Test runs 358 through 396.)

43-024-957

C-2...Per cent of supply air from separator vs water flow rate for various air supply flow rates. 4-ft static water level, no screen, one vane. (Test runs 358 through 396.)

C-3...Per cent carryunder vs total air supplied to loop for various water flow rates. 4-ft static water level, no screen, one vane. (Test runs 358 through 396.)

C-4...Per cent carryunder vs air flow from separators for various water flow rates. 4-ft water level, no screen, one vane. (Test runs 358 through 396.)

43-C24-994

C-5...Total separator head loss vs water flow rate for various air supply flow rates. 4-ft water level, no screen, one vane. (Test runs 358 through 396.)

C-6...Per cent carryunder vs water flow rate for various air supply flow rates. 2-ft static water level, no screen, multi-vane. (Test runs 482 through 535.)

43-024-966

C-7...Per cent of supply air from separator vs water flow rate for various air supply flow rates. 2-ft static water level, no screen, multi-vane. (Test runs 482 through 535.)

43-024-967

C-8...Per cent carryunder vs total supply air flow rate for various water flow rates. 2-ft water level, no screen, multi-vane. (Test runs 482 through 535.)

43-024-968

C-9...Per cent carryunder vs air flow rate from separator for various water flow rates. 2-ft water level, no screen, multi-vane. (Test runs 482 through 535.)

43-024-995

C-10...Total separator head loss vs water flow rate for various air supply flow rates. 2-ft static water level, no screen, multi-vane. (Test runs 482 through 535.)

C-11...Per cent carryunder vs water flow rate for various air supply flow rates. 4-ft static water level, No. 16 mesh screen, multi-vane. (Test runs 536 through 549.)

C-12...Per cent of supply air flow from separator vs water flow rate for various air supply flow rates. 4-ft static water level, No. 16 mesh screen, multi-vane. (Test runs 536 through 549.)

C-13...Per cent carryunder vs total air supplied to loop for various water flow rates (two points on each curve for comparison). 4-ft static water level, No. 16 mesh screen, multi-vane. (Test runs 536 through 549.)

C-14...Per cent carryunder vs air flow rate from separator for various water flow rates (two points on each curve for comparison). 4-ft static water level, No. 16 mesh screen, multi-vane. (Test runs 536 through 549.)

43-024-996

C-15...Total separator head loss vs water flow rate for various water flow rates. 4-ft static water level, No. 16 mesh screen, multi-vane. (Test runs 536 through 549.)

Appendix - Section D

Sample Data and Calculation Sheets

- D-1 Sample Data Sheet
- D-2 Data Calculation Procedure
- D-3 Sample Calculation Sheet

D-1 SAMPLE DATA SHEET

File No. 2-4300-00364

Date: 5/16/60

Sheet No. 35

Personnel: G. Kutsch and H. Yant

(With multivane element in downcomer)

	Run Number						
	449	450	451	452	453	454	455
Air From Pool, In. H ₂ O #1 Man.	10.8-12.4 10.3	13.0	13.5	13.2	13.3	13.7	13.2
Air From Separator In. H ₂ O #2 Man.	3.1-1.6 3.4	1.0	0.75	0.5	0.4	0.2	0.15
Dome Pressure In. Hg. #3 Man.	10.0	9.9	9.8	9.2	9.1	8.8	8.3
Total Drop, In. Hg. #4 Man.	3.7-4.6	3.8-3.2	2.9-3.4	2.3-2.9	2.0-2.4	1.7-2.1	1.3-1.8
Inlet Drop, In. Hg. #5 Man.	1.6-2.3	1.4-2.0	1.2-1.7	1.0-1.5	.8-1.1	.7-1.0	.5-.7
Recirc. Water Orifice In. Hg. #9 Man.	8.1	7.0	6.2	5.05	4.2	3.5	2.8
Supply Air Orifice In. Hg. #10 Man.	5.8	5.8	5.8	5.7	5.6	5.65	5.6
Supply Air Pressure PSIG	85	85	85	86	86	86	83
Supply Air Temp. °F	71	71	71	72	72	72	72
Collect Drum Temp. °F	181	181	180	178	177	176	174
Water Temperature °F	191	190	189	186	185	185	183
Static Water Level In. Hg. #11 Man.	9.9	9.6	9.3	9.8	9.6	9.5	9.8
Carryunder Time Minutes: Seconds	2:05	2:45	2:50	4:05	6:00	6:30	12:00

- ① Air from pool surface - lb/min.
From calibration curve at Δh , in. H_2O , #1 Manometer
- ② Air from separator lb/min.
From calibration curve at Δh , in. H_2O , #2 Manometer
- ③ Air from Pool Surface -CFM Free Air =

$$\frac{\textcircled{1}}{\rho_{\text{Free Air}}} \left(\frac{\sqrt{\rho_{\text{Sat. Air at Standpipe press. and temp.}}}{\sqrt{\rho_{\text{Sat. Air at 5 psig and 170}^\circ\text{F}}} \right)$$

$$\rho_{\text{Free Air}} = .0764 \text{ \#/FT}^3$$

$$\rho_{\text{Sat. Air}} = .0747 \text{ \#/FT}^3 \text{ at 5 psig and 170}^\circ$$

$$\rho_{\text{Sat. Air}} = \rho_{\text{Vapor}} + \frac{\rho_{\text{Air}} = \rho_{\text{Vapor}} (144)}{85.58 (^\circ\text{R})} + \frac{\rho_{\text{Air}} (144)}{53.35 (^\circ\text{R})}$$
- ④ Air from Separator - CFM Free Air =

$$\frac{\textcircled{2}}{\rho_{\text{Free Air}}} \left(\frac{\sqrt{\rho_{\text{Sat. Air at Standpipe press. and temp.}}}{\sqrt{\rho_{\text{Sat. Air at 5 psig., 170}^\circ\text{F}}} \right)$$
- ⑤ Total air out - CFM Free Air

$$\textcircled{3} + \textcircled{4}$$
- ⑥ % of Total Air From Separator

$$100 \frac{\textcircled{4}}{\textcircled{3} + \textcircled{4}} = \frac{\textcircled{4}}{\textcircled{5}} (100)$$
- ⑦ Supply Air Press. PSIA
(Supply Air press. from data sheet + 14.7)
- ⑧ Supply Air - CFM
(Value from calibration curve at Δh in. H_2O , #10 Manometer) $\times (\sqrt{\textcircled{7}})$
- ⑨ Supply Air - CFM Free Air =

$$\frac{\textcircled{8} \sqrt{560}^\circ\text{R}}{\sqrt{^\circ\text{R Supply Air}}} \text{ at } 75^\circ\text{F Supply Air Temp: } \frac{\sqrt{560}}{\sqrt{^\circ\text{R}}} = 1.023$$

(10) Water Flow Rate - GPM

From Calibration Curve at Δh in. HG - #9 Manometer

(11) Water Flow Rate, CFM

$$\frac{(10)}{7.48 \text{ Gal./FT}}^3$$

(12) Static Water Level (9.8 inches Hg. on #11 manometer is 4 ft. above baffle sleeve outlet, equiv. water level)

(13) Carryunder time for 6 inch displacement on collection drum sight glass - Minutes

(14) Carryunder flow rate at Press. and Temp. in collection drum, CFM (Approximately at P and T at separator water outlet).

$$\frac{0.3925 \text{ FT}^3}{(13)}$$

(15) Carryunder rate, % by volume of mixture flow rate from separator water outlet

$$\frac{(14)}{(11)} (100)$$

(16) Total separator head loss, Ft. of test water

(inches Hg. on #4 manometer) $(1.046) \frac{\text{Density of Std. water}}{\text{Density of Test water}}$

(17) Separator Inlet Nozzle head loss, Ft. of test water

(inches Hg. on #5 manometer) $(1.046) \frac{\text{Density of Std. water}}{\text{Density of Test water}}$

(18) Separator Outlet Nozzle head loss, Ft. of test water

$$(16) - (17)$$

D-3 SAMPLE CALCULATION SHEET

File No. 2-4300-00364

Sheet No. 46

(4-ft. static water level - with multivane)

	Run Number						
	449	450	451	452	453	454	455
Pool Air Exhaust lb/min (1)	8.35 8.90 8.15	9.10	9.30	9.20	9.25	9.35	9.20
Sep. Air Exhaust lb/min (2)	2.85 2.05 3.00	1.60	1.40	1.15	1.00	.75	.65
Pool Air Exhaust lb/min (3)	107.0 114.0 104.4	116.6	119.2	117.9	118.5	118.0	116.4
Sep. Air Exhaust CFM (S) (4)	36.5 26.3 38.4	20.5	18.0	14.6	12.7	9.5	8.2
Total Air CFM (S) (5)	141.6	137.1	137.2	132.5	131.2	127.5	124.6
Per Cent from Sep. (6)	23.9	15.0	13.1	11.0	9.7	7.5	6.6
Air Supply Press. PSIA (7)	99.7	99.7	99.7	100.7	100.7	100.7	97.7
Air Supply CFM (8)	91.0	91.0	91.0	90.0	89.0	89.5	90.0
Air Supply CFM (F) (9)	93.4	93.4	93.4	92.3	91.2	91.8	92.3
Water Flow Rate GPM (10)	1520	1425	1330	1200	1100	1000	890
Water Flow Rate CFM (11)	204	191	178	161	147	134	119
Static Water Level, Ft (12)	9.9	9.6	9.3	9.8	9.6	9.5	9.8
Carryunder Time, Min. (13)	2.083	2.75	2.833	4.083	6.00	6.50	12.00
Carryunder Rate CFM (14)	.188	.143	.138	.096	.066	.060	.033
Carryunder Per Cent (15)	.092	.075	.078	.060	.045	.045	.028
Total Drop: Separator Ft. Reactor Water (16)	4.36	3.64	3.27	2.70	2.29	1.96	1.61
Drop: Inlet Nozzle Ft. Reactor Water (17)	2.02	1.76	1.50	1.30	.99	.88	.62
Drop: Outlet Nozzle Ft. Reactor Water (18)	2.34	1.88	1.77	1.40	1.30	1.08	.99