MARTIN MARIETTA ENERGY SYSTEMS LIBRARIES
3 4456 0360673 8

CENTRAL RESEARCH LIBRARY
DOCUMENT COLLECTION

ORNL 1602
Instrumentation

Cy. 4

DEVELOPMENT OF THE PIPE-LINE GAS SEPARATOR

J. A. Hafford

CENTRAL RESEARCH LIBRARY
DOCUMENT COLLECTION

LIBRARY LOAN COPY

DO NOT TRANSFER TO ANOTHER PERSON

If you wish someone else to see this document, send in name with document and the library will arrange a loan.

OAK RIDGE NATIONAL LABORATORY
OPERATED BY
CARBIDE AND CARBON CHEMICALS COMPANY
A DIVISION OF UNION CARBIDE AND CARBON CORPORATION
UCC
POST OFFICE BOX P
OAK RIDGE, TENNESSEE

ORNL 1602

Copy No 4

C ntract No W 7405 eng 26

REACTOR EXPERIMENTAL ENGINEERING DIVISION

DEVELOPMENT OF THE PIPE-LINE GAS SEPARATOR

J A Hafford

DATE ISSUED

FEB 18 1954

OAK RIDGE NATIONAL LABORATORY
Operated by
CARBIDE AND CARBON CHEMICALS COMPANY
A Division of Union Carbide and Carbon Corporation
Post Office Box P
Oak Ridge Tennessee

3 4456 0360673 8

INTERNAL DISTRIBUTION

1 C E Center
2 Biology Library
3 Health Physics Library
4 5 Central Research Library
6 Reactor Experimental Engineering Library
7 11 Laboratory Records Department
12 Laboratory Records ORNL R C
13 C E Larson
14 L B Emlet (K 25)
15 J P Murray (Y 12)
16 A M Weinberg
17 E H Taylor
18 E D Shipley
19 C E Winters
20 F C VonderLage
21 R C Briant
22 J A Swartout
23 S C Lind
24 F L Culler
25 A H Snell
26 A Hollaender
27 M T Kelley
28 W J Fretague
29 G H Clewett
30 K Z Morgan
31 T A Lincoln
32 A S Householder
33 C S Harrill
34 D S Billington
35 D W Cardwell
36 E M King
37 R N Lyon
38 J A Lane
39 A J Miller
40 R B Briggs
41 A S Kitzes
42 O Sisman
43 R W Stoughton
44 C B Graham
45 W R Gall
46 H F Poppendiek
47 S E Beall
48 J P Gill
49 D D Cowen
50 R A Charpie
51 R E Aven
52 I Spiewak
53 L F Goode
54 56 F C Zapp
57-62 J A Hafford
63 P M Reyling
64 G C Williams
65 M J Skinner

EXTERNAL DISTRIBUTION

66 R H Wilson North American Aviation Inc
67 R F Bacher California Institute of Technology
68 316 Given distribution as shown in TID-4500 under Instrumentation Category

DISTRIBUTION PAGE TO BE REMOVED IF REPORT IS GIVEN PUBLIC DISTRIBUTION

NOMENCLATURE

C = constant of integration

E = energy of liquid stream ft

E_T = total energy of vortex ft

$\overline{E}$ = weighted energy ft

$\overline{E}$ = energy in vortex due to axial velocity ft

$\overline{E}_p$ = energy in vortex due to static pressure ft

$\overline{E}_t$ = energy in vortex due to tangential velocity ft

g = gravitational constant ft/sec^2

g_c = conversion constant lb of force ft/lb of mass sec^2

K = proportionality constant units not fixed

P_R = static pressure at radius R ft

P_s = static pressure ft

ΔP_s = change in P_s between positions 1 and 2 ft

Q = volumetric flow ft^3/sec

r = radius of pipe ft

r_v = radius of void ft

R = maximum inner pipe radius ft

u = radial bubble velocity at point r ft/sec

$\overline{u}$ = average radial bubble velocity ft/sec

V_1 = absolute velocity at position 1 (before vanes) ft/sec

V_2 = absolute velocity at position 2 (after vanes) ft/sec

V = axial velocity component ft/sec

V_0 = axial velocity before rotation ft/sec

V_R = tangential velocity component at R ft/sec

V_t = tangential velocity component ft/sec

$\overline{V}$ = average axial velocity ft/sec

γ = surface tension lb/ft

ρ_1 = liquid density lb/ft^3

ρ_2 = gas density lb/ft^3

DEVELOPMENT OF THE PIPE LINE GAS SEPARATOR

J A Hafford

The results of a series of tests on various components of a model of an axial gas separator are presented In particular the following information is included (1) the measured flow patterns through two sets of vanes compared with the expected theoretical flow patterns (2) the applicability of volutes and the experimental results obtained with two types of volutes (3) the most desirable means of gas removal and removal efficiencies and (4) a procedure for estimating required separator length and pressure loss

The problem of removing relatively small amounts of gas from a stream of liquid is usually solved by using a settling tank which permits bubbles of gas to rise to a free surface In applications in which the amount of liquid hold up is critical this approach has the serious drawback of requiring too much liquid However a chamber which imparts centrifugal force to the liquid and forces the gas to a free surface before the liquid leaves the chamber offers a possible solution There are several such mechanical separators which can be used The most promising is the pipe line or axial gas separator which occupies a straight section of pipe that in most applications would be available

The pipe line gas separator consists of a means of imparting rotation to the liquid that is vanes or a volute and a section of pipe with a void space in the center to provide the free surface for gas to travel to and from where it will be removed If it is desired the energy of rotation can be partially recovered with vanes or a volute at the discharge end of the separator

A model of the axial gas separator was built and several runs were made to test vanes and volutes of different types for energy conversion and recovery and for gas-removal efficiency Tests were also run on the most feasible means of controlling the size of the void

It is desired to acknowledge the assistance and encouragement given by I Spiewak of Oak Ridge National Laboratory and G F Wislicenus of Johns Hopkins University during the course of this work

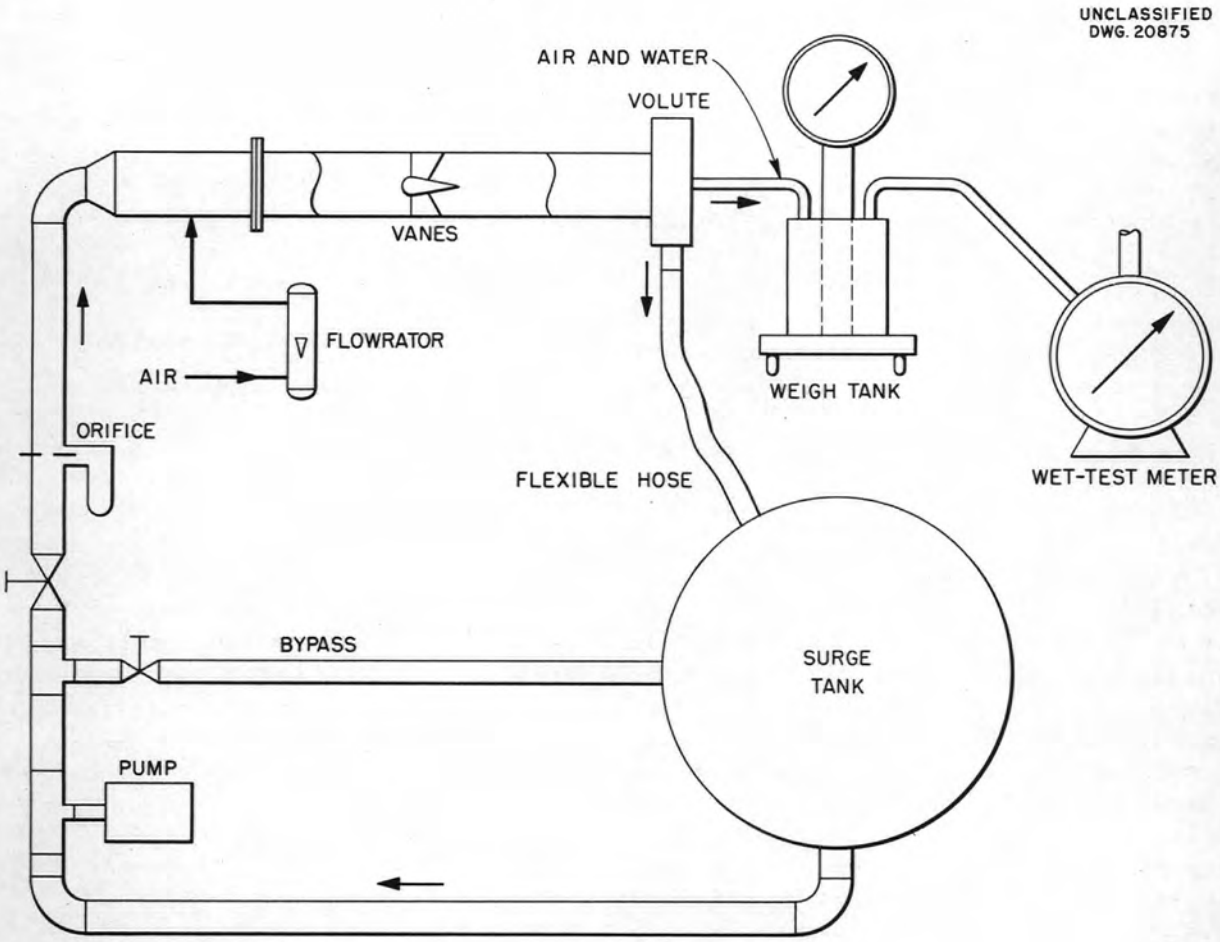
EXPERIMENTAL APPARATUS

The separator model consisted of 5 in lucite tubing of various lengths to test the effect of vortex length on the performance of the separator Two sets of vanes were built to provide rotation and one to test recovery One conventional volute was built to test both rotation and recovery and one modified volute was built to test rotation only These components are discussed in the section on Velocity Distribution

Figure 1*a* shows a schematic layout of the loop used during the runs Figure 1*b* shows the assembled separator model This photograph was taken before any runs had been made The pump had a maximum output of 900 gpm at a head of 75 ft of water The maximum flow rate used during the runs was 600 gpm Air was the only gas used and it was metered into the system through a Flowrater which had been calibrated against a wet test meter used to meter the separated gas

The separator discharged into a surge tank of 500 gal capacity Even though baffles were provided in the tank a small amount of gas was entrained which later was picked up in the void of the separator However other errors in measurements exceeded that introduced by the additional gas

When the volute was used at the discharge end of the separator it was possible to insert a rod into the void along its axis On the end of the rod was a small ball bearing with two pins welded radially to it Two adjustable



(*a*)

Fig. 1*a*. Schematic Diagram of Gas Separator.

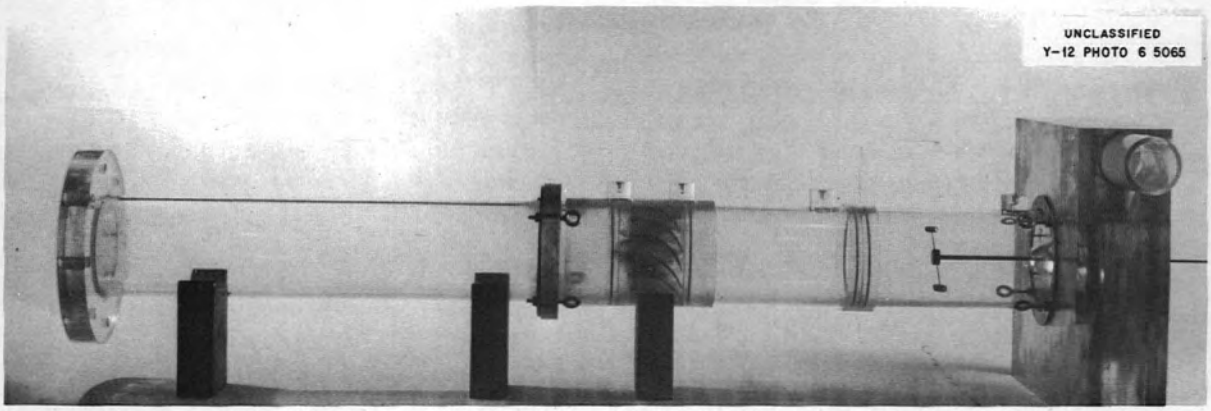


Fig. 1*b*. Assembled Separator Model with Model I Vanes in Place. Spinner used to measure flow patterns shown.

paddles on the pins rotated with the water and could be timed with a Strobotac In this manner the tangential component of the water velocity was measured over the length of the separator under a variety of conditions The absolute direction of flow was determined by inserting a streamer into the water and measuring the angle of flow with a protractor The axial component of flow could then be calculated

The gas could be introduced either directly into the void or into the liquid several diameters upstream from the separator Means for removal of gas included central take offs at both ends of the separators and various types of annular take-offs The pressure in the separator was usually about 20 to 30 psig

Provisions for measuring pressures at a number of points along the separator were included The over-all pressure drop averaged out for the various components to be about 2 to 3 velocity heads

VELOCITY DISTRIBUTION

Vanes Vanes possess several inherent advantages for gas separator design They are simpler to construct than volutes and they require less volume in the system In addition the velocity profiles can be regulated by the shape of the vane system However the problem of entry and exit from the void is somewhat more complicated and void shape and stability are less certain than in the case of a volute

In selecting a flow pattern a tangential velocity defined by the relation

$$V_t = K\sqrt{r} \tag{1}$$

was chosen This choice was based on the data presented in a report by Peters [1] which indicated that a vortex of this nature was quite stable the radial bubble velocity was theoretically adequate as calculated from equations taken from a report by Peebles [2]

To complete the calculations of vane discharge angle the axial velocity distribution must be known The expected axial velocity variation is shown to be (app A)

$$V_a \frac{dV_a}{dr} = -V_t\left(\frac{V_t}{r} + \frac{dV_t}{dr}\right) \tag{2}$$

and by substituting Eq 1 in Eq 2 and integrating

$$V_a^2(r) = V_R^2 + 3K^2R - 3K^2r$$

or

$$V(r) = \sqrt{V_R^2 + 3K^2(R - r)} \tag{3}$$

V_R is found by assuming a value for the axial velocity and then finding the average axial velocity from Eq 3 This trial and error procedure is repeated until a value is found for V_R that gives the correct average axial velocity With V and V_t known the direction of the absolute velocity which is the desired vane discharge angle can be found

During the actual fabrication of the first set of vanes a drawing was misinterpreted and the resultant vanes gave the theoretical distribution shown in Fig 2 The experimentally determined velocities are shown in Fig 3

The energy distributions were calculated from the experimental data by graphical methods that use the formulas developed in Appendix B There is apparently good conversion of energy from static pressure to velocity The over all efficiency of conversion is about 85%

The first set of vanes did a good job of separation of gas but the void created was a series of lumps and thus there was indication of an unexpected oscillating radial velocity in the vortex The reason for this is not clear The void although irregular was quite stable and the

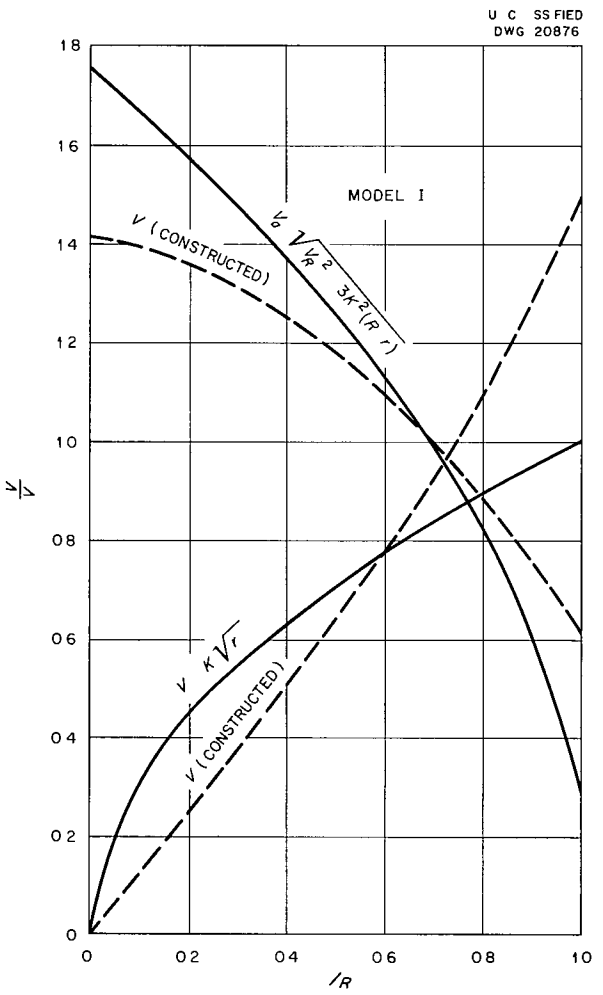


Fig 2 Theoretical Velocity Distribution Model I

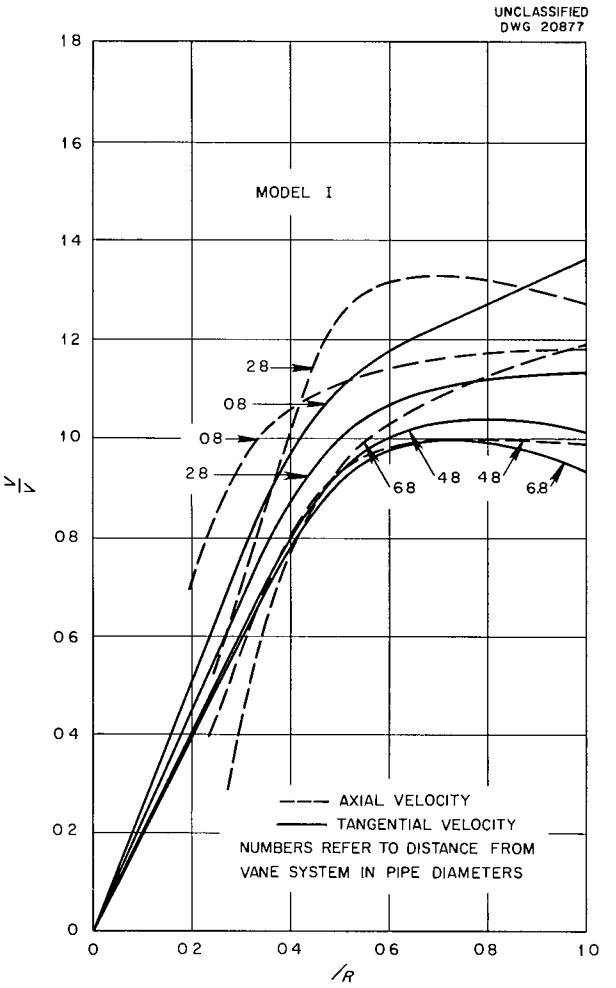


Fig 3 Measured Velocity Distribution Model I

tangential velocity decayed in an orderly manner but the profile of axial velocity was quite erratic Figure 3 shows the variations of V_t and V with distance from the vanes The irregular void appeared to cause some difficulty in gas removal the snozzle had to be carefully placed to get suitable efficiency In general the first set of vanes was not wholly satisfactory and a second set was constructed

The second set of vanes was designed to give a theoretical distribution of

$$V_t = \frac{K}{\sqrt{r}} \tag{4}$$

This distribution when substituted in Eq 2 gives

$$V^2 = \frac{K^2}{r} + C$$

For the case in question V was set equal to V_t for all radii (that is vane discharge angle set to 45 deg) thus

$$V = V_t = \frac{K}{\sqrt{r}}$$

(i e $C = 0$)

Here K is determined from conditions of continuity Figure 4 shows the theoretical velocity distribution and Fig 5 shows the experimental distribution

The energy relations show a vane efficiency of about 90% for a vortex of greater intensity than that of the

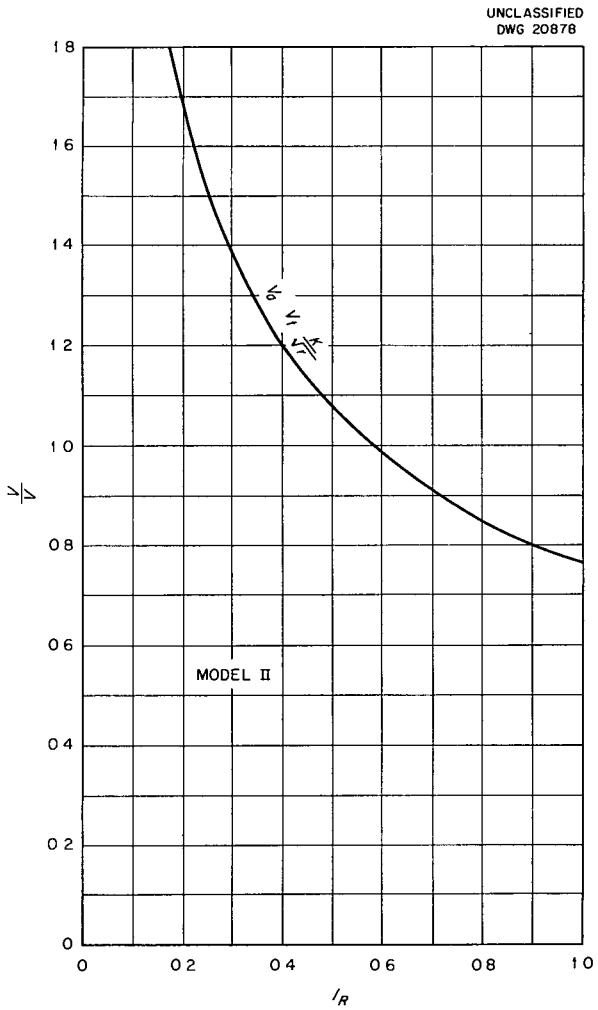


Fig 4 Theoretical Velocity Distribution Model II

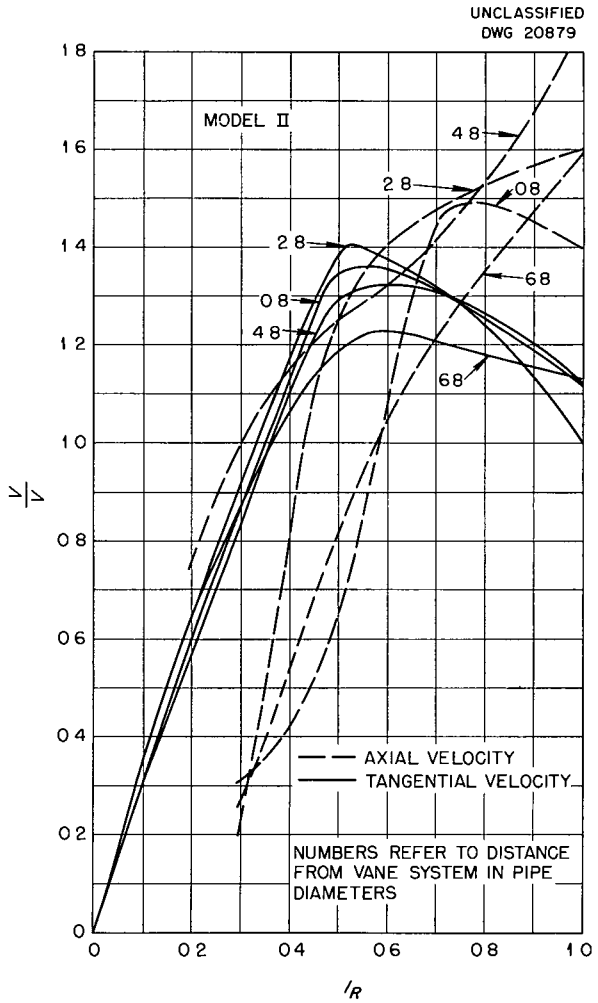


Fig 5 Measured Velocity Distribution Model II

previous model The gas separation was decidedly improved and the gas removal was simplified because the void was smooth and stable

Again the vortex was stable and decayed in a regular fashion however the axial velocity profile was more erratic than that in Model I The variation of V_a and V_t with distance from the vanes is shown in Fig 5

The theory used in vane design only roughly approximates the actual flow conditions At present there is no quantitative method of determining what the actual flow patterns will be for any given vane configuration A general qualitative feeling for the problem can be gained however by a study of the theory the deviations being in a generally expected direction

The method of fabrication of the vane system is of interest Vanes for Model I were cut from 5 in lucite tubing The varying discharge angle was achieved by cutting away the cylindrical section to the proper distance Vanes for Model II had a constant discharge angle and were cut from sheet copper bent to the desired parabolic shape and soldered to the supports Figures 6 and 7 show these vane systems

Recovery Vane System In the recovery vanes the leading edge of the vane should be parallel to the

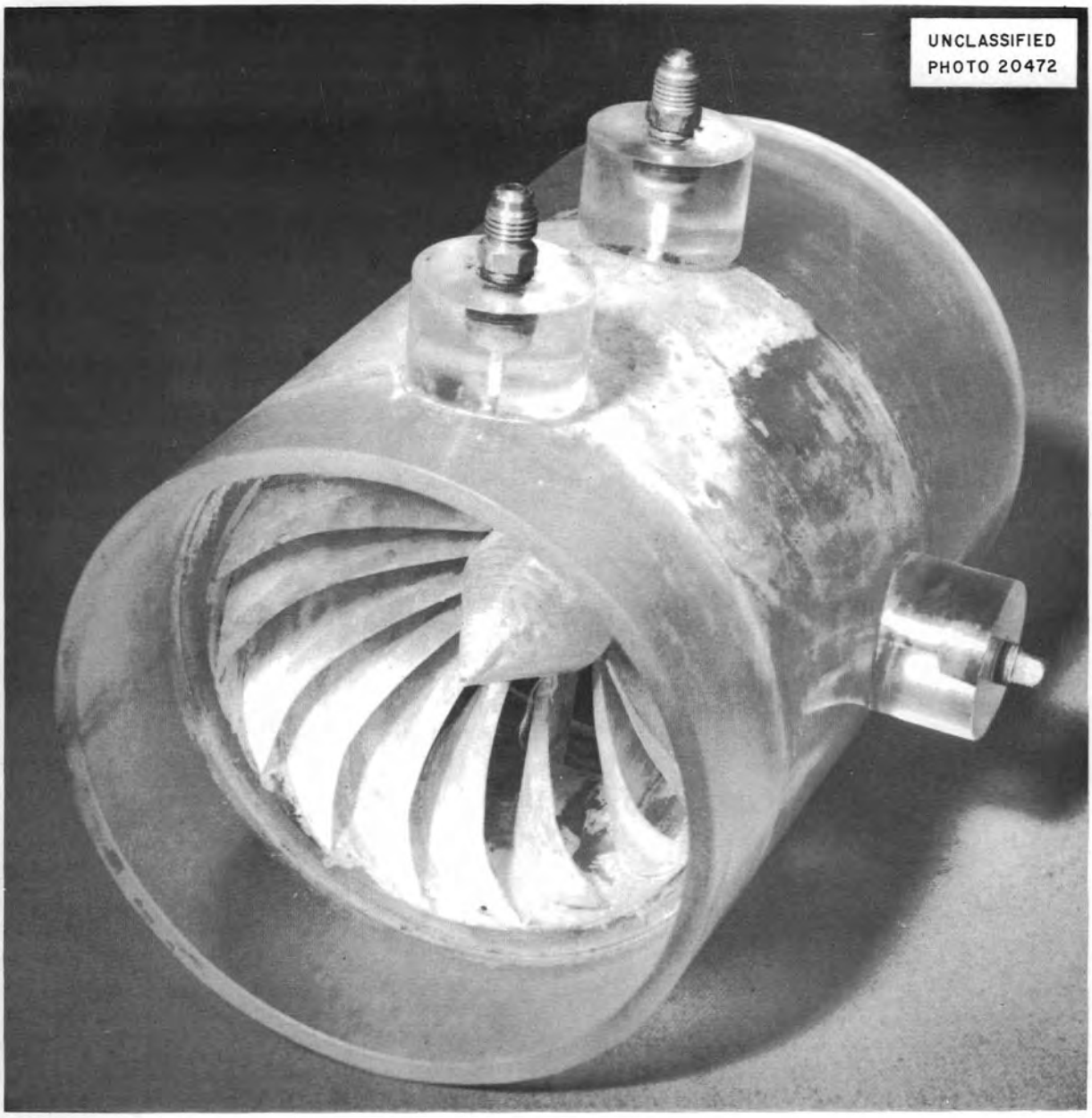


Fig. 6. Vanes for Model I.

absolute direction of flow, but the design of the trailing edge is dependent on the desired flow pattern following the vanes. An airfoil vane would provide greater latitude of design than the thin vanes used.

The design of recovery vanes is a difficult problem, since it is hard to predict the flow pattern in the vortex with any degree of accuracy, and the efficiency of the recovery system is highly dependent on the proper leading angle of the vanes. The best solution to the problem is to build the system, or a model of it, without the recovery vanes in place, and to measure the flow pattern before building the vanes. It is felt that about 80% efficiency of conversion of energy by the recovery vanes could be realized by careful

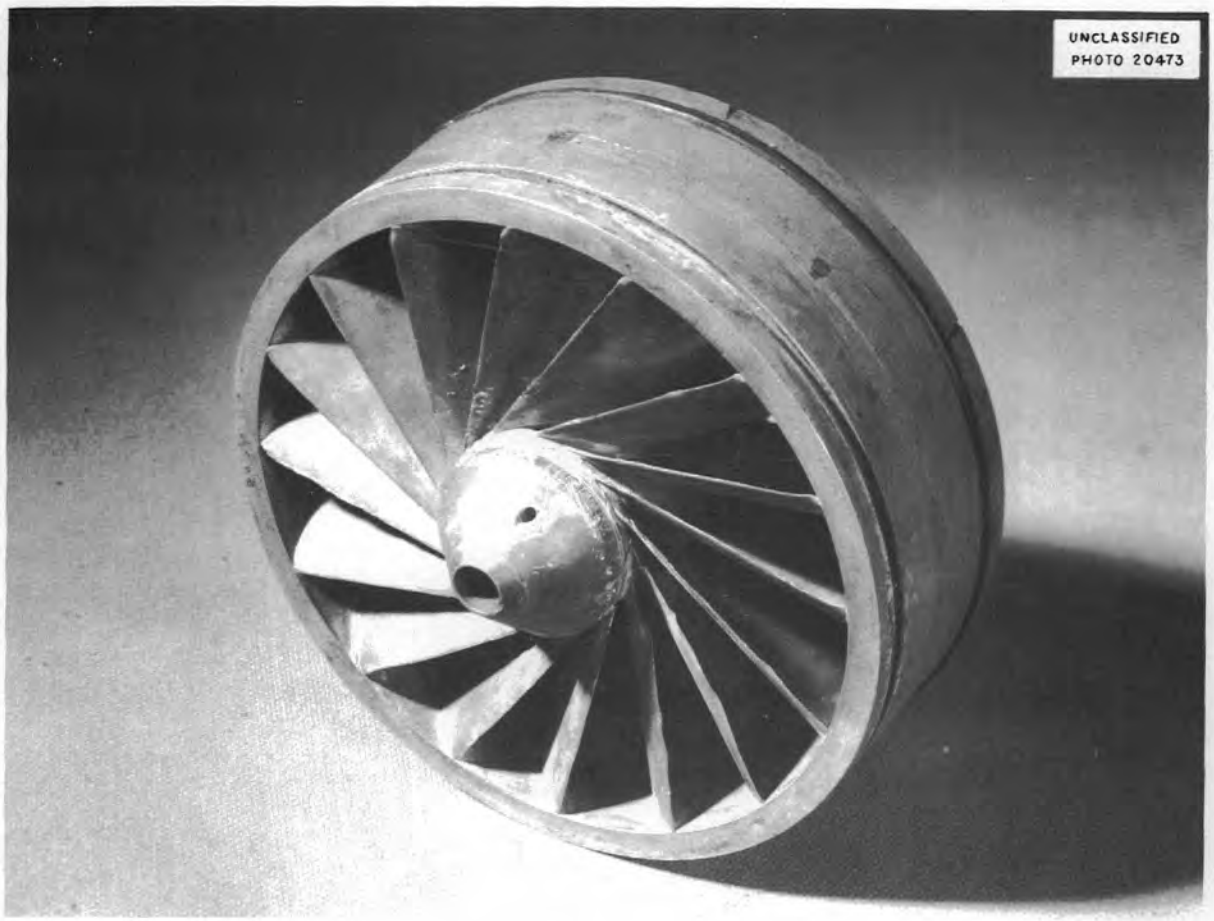


Fig. 7. Vanes for Model II.

design to utilize vanes that have proper airfoil cross sections.

The experimental recovery vanes tried in the model proved to be only about 15% efficient. (Efficiency is defined as percentage of kinetic energy converted to static pressure.) It is felt that this low efficiency was due to failure in determining the inlet angle of flow to the vanes. Figure 8 shows these recovery vanes.

Volute. In order to speed and simplify construction of a prototype volute, a compromise was made in the design of the volute used in the early model. Figure 9 is a photograph of the volute used in the experiments. The outlet was too small to give very good results, but even so, when the increased velocity is considered, the volute did a fair job of energy conversion, although instead of reducing velocity, it increased it. The volute casing was of triangular cross section and followed an Archimedean spiral. In the experiment, the velocity head was approximately doubled through the volute.

The volute was used principally at the outlet of the gas separator and provided a convenient means of entry for the spinner into the vortex section. The use of a volute to recover rotational energy loses value when the cost of construction of a large, high-pressure volute is compared

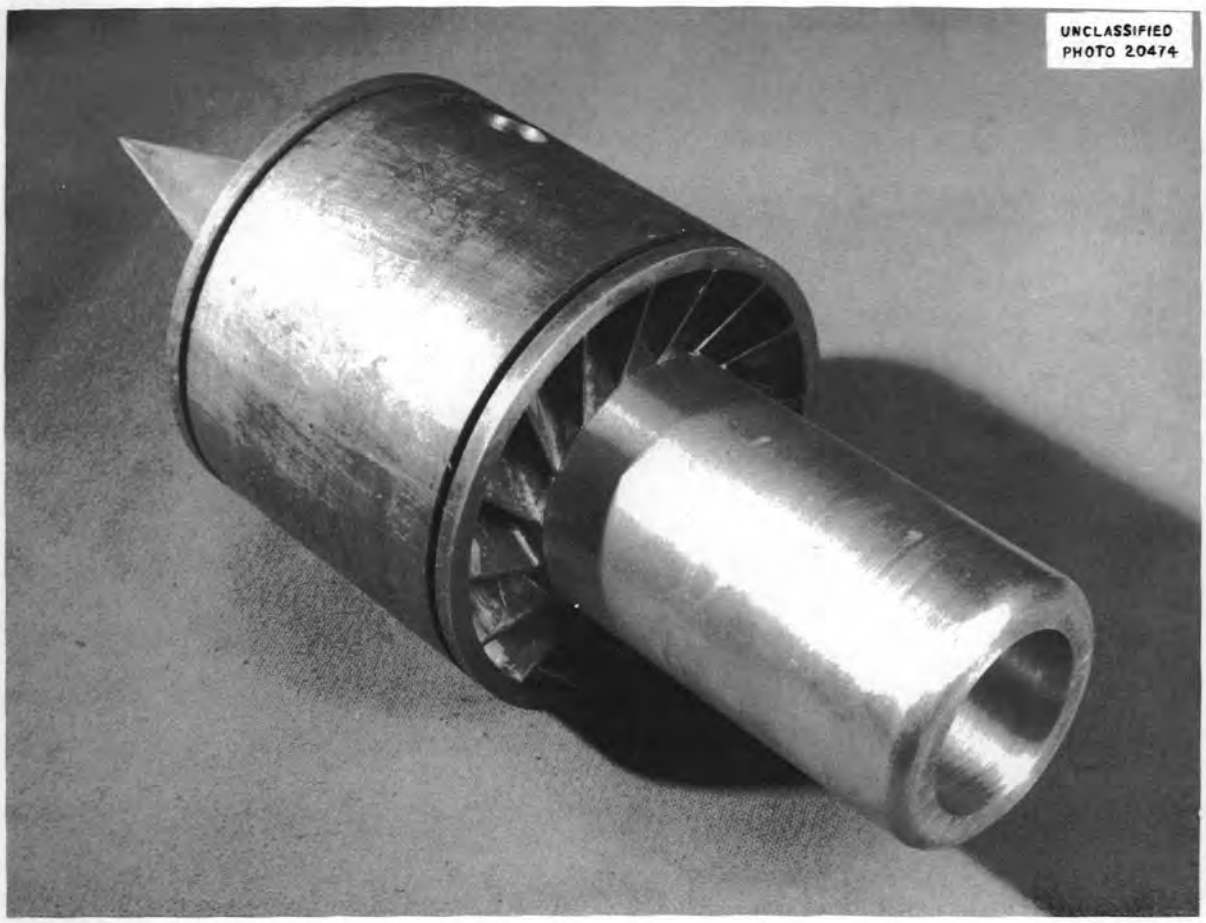


Fig. 8. Recovery Vanes and Large Void Take-Off.

with the cost of accepting the loss of energy without any attempt to recover it. It would take several years to pay for the volute with pumping power saved, not to mention the additional required hold-up of liquid when hold-up must be held to a minimum. However, if energy recovery is desired, it is not necessary to know the velocity distribution entering the volute and this troublesome design problem is eliminated.

The use of a volute as a rotating mechanism has definite advantages which may be of sufficient importance to override the disadvantages. A very high tangential velocity can be attained with a volute. Normally, such high velocities are unnecessary for separation, but they could be used to good advantage if an extremely short separator were required. The high velocities are attained with considerable sacrifice of pressure drop, and the pressure may be largely unrecoverable.

Helical Vane. A modified volute was built, Figs. 10 and 11, which consisted of a single helical vane that was entered on a chord and was perpendicular to the axis of the separator. In this case, there was no radial component of velocity and, consequently, no increase in velocity, since there was no requirement of conservation of angular momentum as there is in the standard volute.

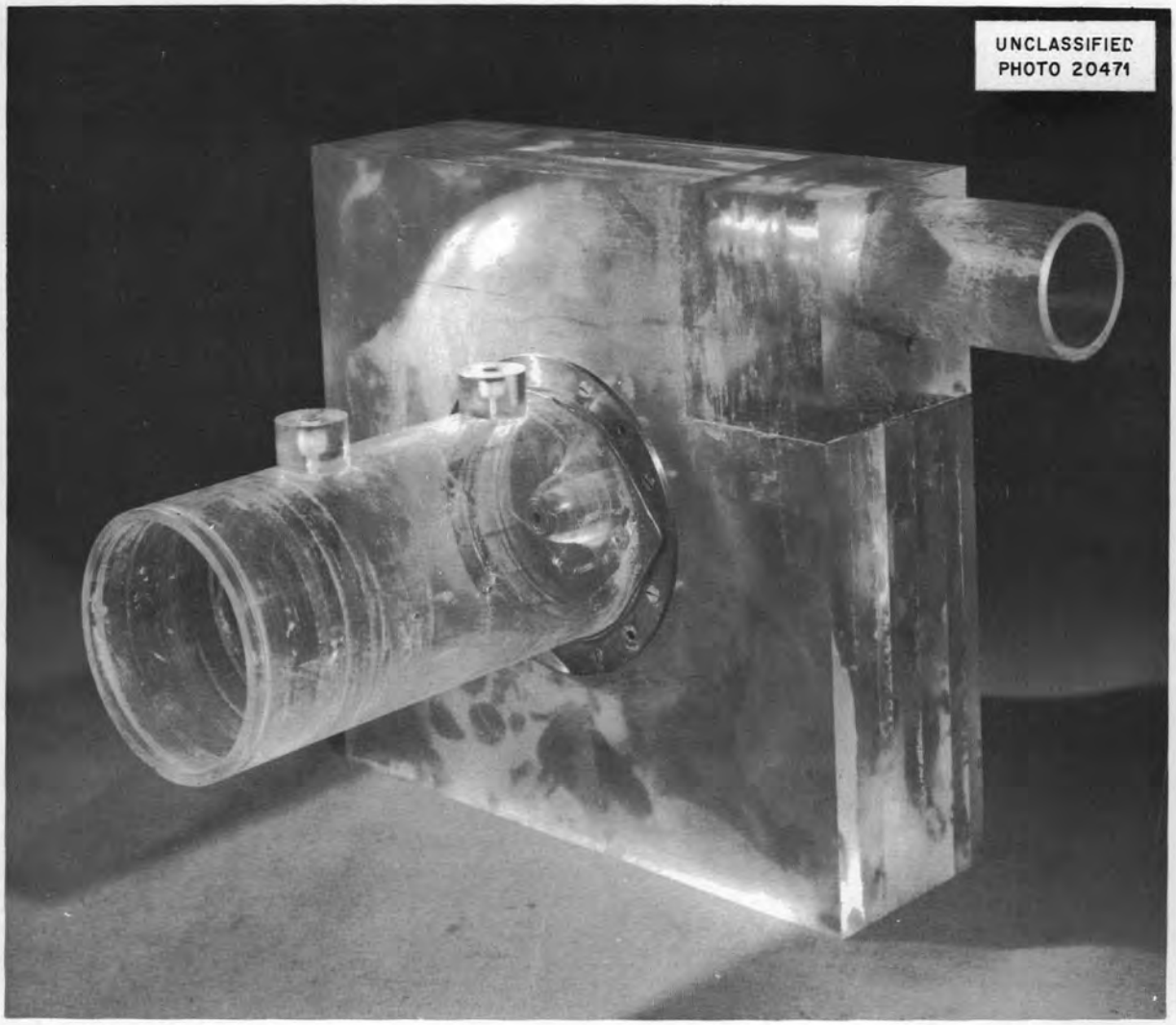


Fig. 9. Volute.

Test runs showed that there was little loss of energy in the helix, since there was good efficiency of conversion. The flow patterns were not determined, but they were calculated approximately by using a few assumptions. Appendix C shows the analysis of the helix.

The primary disadvantage of the helix is that it is difficult to construct a helix that will withstand high pressures. However, among the advantages of the helix is that it provides a convenient elbow if one is needed, and the elbow permits easy entry to the void, as in the volute; it has relatively low energy losses and low hold-up; the rotational velocities are ample for efficient removal of gas; and it is particularly adaptable to a large void. The possibility of use of the helix as a recovery system was not investigated, but it may be suitable under some conditions. However, no future work on this type of mechanism is being contemplated at this time.

GAS REMOVAL

The gas must, in general, be removed through a tube inserted on the axis of the separator, that is, along the center line of the void. This take-off may be at either end of the vortex

UNCLASSIFIED
DWG 20086A

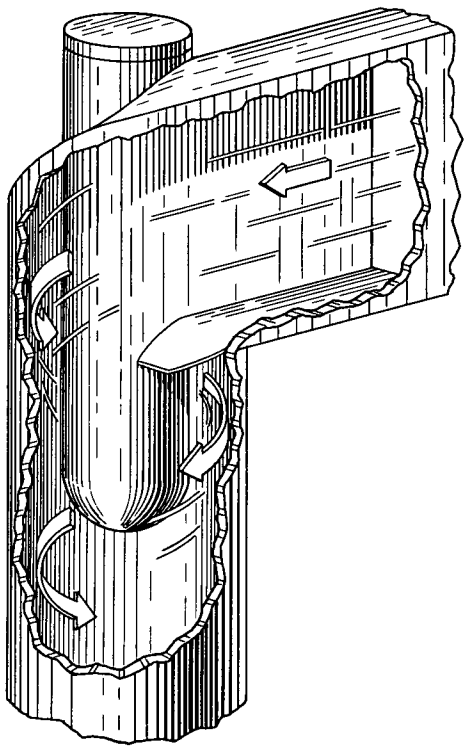


Fig 10 Principle of Operation of Helical Vane

section with the downstream take off being a little more reliable for complete removal

When no gas is flowing there will be little or no void existing in the separator and the liquid will be flowing in relatively large amounts through the take off If however it is desired that a gas space be maintained under all conditions of operation that is if the void were to be used as a pressurizer the take off must be an annulus whose outside diameter is larger than the minimum required void size A shaped knob with its maximum diameter equal to the minimum void size allowed is placed in the center of the take-off a short distance from the entrance as shown in Fig 12

Attempts to provide complete gas removal with a single annular take off were not successful in that many bubbles were swept past with the liquid stream However if the above restrictions are complied with essentially complete gas removal is experienced with all void sizes

In removing the gas a definite amount of entrainment is unavoidable The pressure drop in the gas take off line will be dependent on the amount of liquid entrained or conversely the amount of liquid entrained will be largely a function of available pressure drop and pipe size Several papers on calculation of pressure drop in two phase flow have been published [3 4 5] Reference 4 is the most complete publication and it refers to other work in the field Future tests are planned to determine the applicability of these calculations to larger scale operations

REQUIRED SEPARATOR LENGTH

The required length of a separator can be estimated by using the calculations given in ref 2 for bubble velocity (app D) The length of vortex section required for the model compared very favorably with the visually observed separation of bubbles The observation was aided by the use of a Strobotac to detect bubbles left in the liquid stream No bubbles could be seen beyond about two pipe diameters (10 in) from the vane systems It is well to state here that to make accurate predictions the velocity profiles must be known accurately The calculations in Appendix D were based on a relatively large bubble size If very small bubbles less than 0 010 in in diameter were present the length of required separator could be much greater

PRESSURE DROP THROUGH A SEPARATOR

The pressure drop through a gas separator can be approximated by assuming an efficiency of conversion

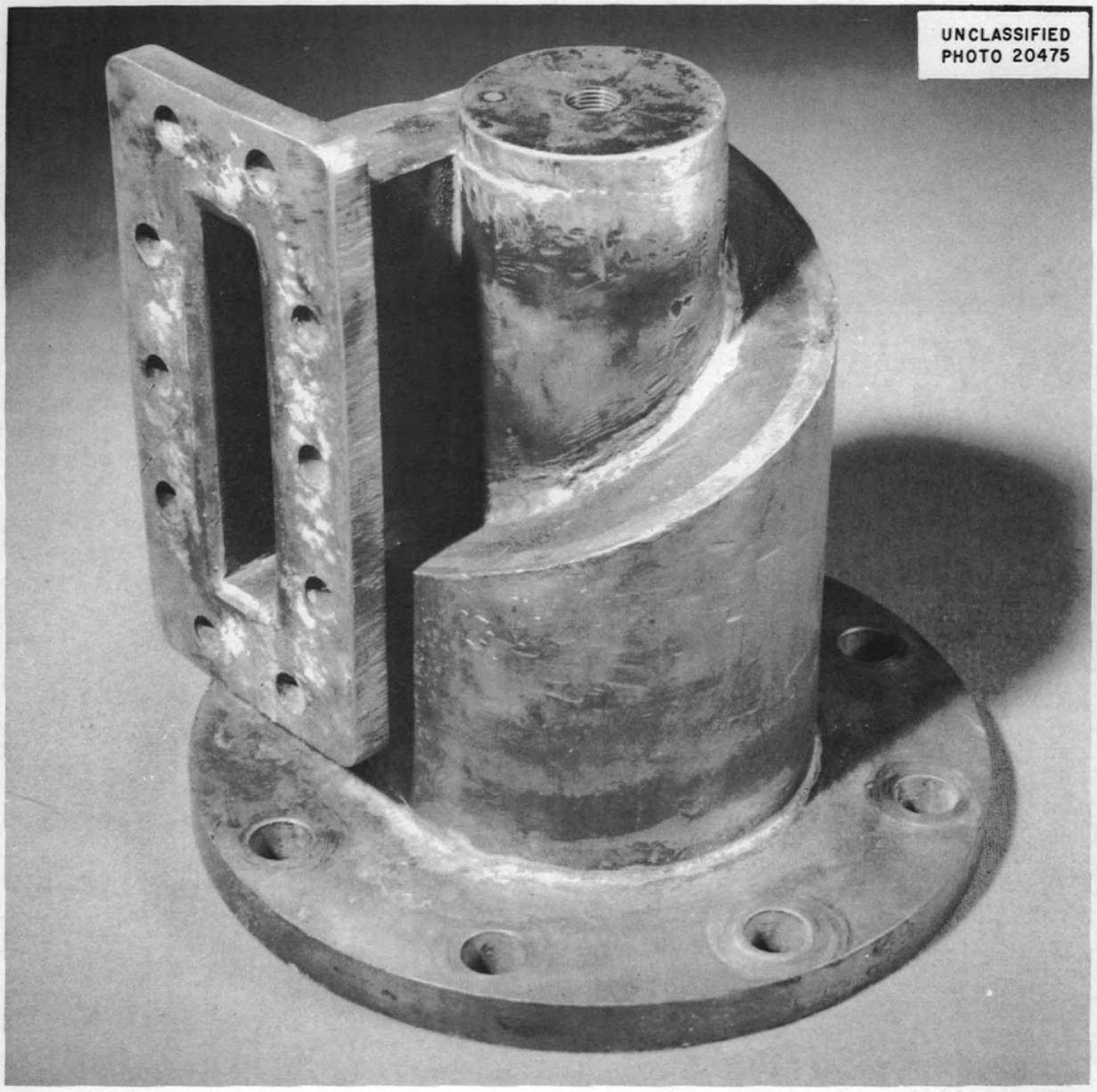


Fig. 11. Helical Vane.

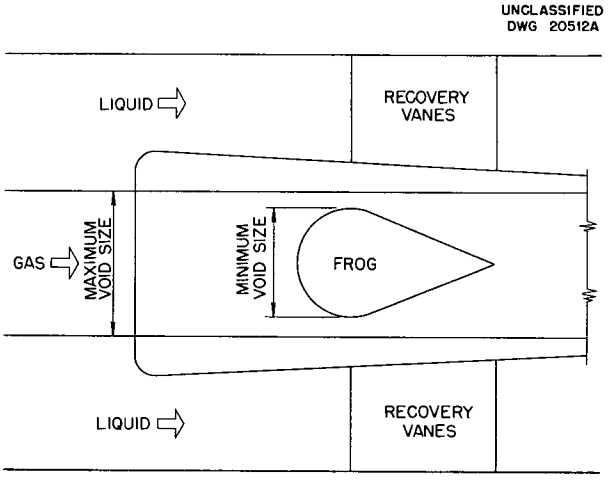


Fig 12 Gas Take Off for Maintenance of Large Void

of pressure to velocity in the rotation system of 90% and a recovery efficiency of 80% The pressure drop in the vortex will be about three times that which would exist without rotation

CONCLUSIONS

Either vanes or volutes may be used both as rotation systems or recovery systems with good results The choice must be made on the basis of the requirements of the whole system

Gas removal is very good if a centrally located take off is used The void size is controlled largely by the inside diameter of the gas take off line For the assured maintenance of a large void a take-off as large as the diameter of the void desired and having a recessed annulus must be used

The required length of the vortex as calculated is reasonably accurate if the velocity pattern is known

REFERENCES

1 H Peters *Conversion of Energy in Cross Sectional Divergences Under Different Conditions of Inflow* NACA TM 737
2 F N Peebles ORNL 1171 (classified)
3 O P Bergelin *Chem Eng* **56** 104 6 (1949)
4 R C Martinelli J A Putnam and R W Lockhart *Trans Am Inst Chem Eng* **42**, 681 705 (1946)
5 J A Hafford ORNL CF 53 5 112 (classified)
6 G F Wislicenus *Fluid Mechanics of Turbomachinery* chap 11 McCraw Hill New York 1947
7 J O Bradfute ORNL CF 52 5 235 (classified)

Appendix A

FLOW THROUGH STATIONARY VANES

The assumption is made that the entire flow field is at constant energy This assumption leads to

$$\Delta P_s = \frac{V_1^2}{2g} - \frac{V_2^2}{2g}$$

where

$$V_1^2 = V_{t_1}^2 + V_{a_1}^2$$

$$V_2^2 = V_{t_2}^2 + V_{a_2}^2$$

and therefore

$$\Delta P_s = \frac{V_{t_1}^2 + V_{a_1}^2 - V_{t_2}^2 - V_{a_2}^2}{2g}$$

Upon differentiating to get the radial pressure gradient

$$\frac{d(\Delta P_s)}{dr} = \frac{V_{t_1}}{g}\frac{dV_{t_1}}{dr} + \frac{V_{a_1}}{g}\frac{dV_{a_1}}{dr} - \frac{V_{t_2}}{g}\frac{dV_{t_2}}{dr} - \frac{V_{a_2}}{g}\frac{dV_{a_2}}{dr}$$

it can be shown that

$$\frac{dP_{s_1}}{dr} = \frac{V_{t_1}^2}{rg} \qquad \text{and} \qquad \frac{dP_{s_2}}{dr} = \frac{V_{t_2}^2}{rg}$$

and

$$\frac{d(P_{s_2} - P_{s_1})}{dr} = \frac{d(\Delta P_s)}{dr} = \frac{V_{t_2}^2 - V_{t_1}^2}{rg}$$

Hence

$$\frac{V_{t_2}}{g}\frac{dV_{t_2}}{dr} + \frac{V_{a_2}}{g}\frac{dV_{a_2}}{dr} + \frac{V_{t_2}^2}{rg} - \frac{V_{t_1}}{g}\frac{dV_{t_1}}{dr} - \frac{V_{a_1}}{g}\frac{dV_{a_1}}{dr} - \frac{V_{t_1}^2}{rg} = 0$$

or

$$V_{t_2}\left(\frac{V_{t_2}}{r} + \frac{dV_{t_2}}{dr}\right) + V_{a_2}\frac{dV_{a_2}}{dr} = V_{t_1}\left(\frac{V_{t_1}}{r} + \frac{dV_{t_1}}{dr}\right) - V_{a_1}\frac{dV_{a_1}}{dr}$$

In the case of no rotation before the vanes

$$V_{t_1} = 0 \qquad V_{a_1} = \text{constant}$$

Therefore by dropping subscript 2

$$V_t\left(\frac{V_t}{r} + \frac{dV_t}{dr}\right) + V_a\frac{dV_a}{dr} = 0$$

Appendix B

ENERGY BALANCE EQUATIONS

The energy before the vanes is

(1) $$E = \frac{V_{a_0}^2}{2g} + P_s$$

The total energy in the vortex at radius r is

(2) $$E(r) = \frac{V_a^2(r)}{2g} + \frac{V_t^2(r)}{2g} + P_s(r)$$

The axial and tangential energy components are self evident and

(3) $$P_s(r) = P_R - \frac{1}{g}\int_r^R \frac{V_t^2(r)}{r}\, dr = P_R - \frac{1}{g}\sum_r^R \frac{V_t^2(r)}{r}\,\Delta r$$

To find the energy passing through a section of the vortex $E(r)$ must be weighted for area and velocity that is

(4) $$\overline{E} = \frac{2\pi}{Q}\int_0^R E(r)\; V_a(r)\; r\; dr$$

where

$Q = \pi R^2\, \overline{V}_a$

$\overline{V}_a$ = average axial velocity ft/sec

Substituting Eq 2 in Eq 4 gives

(5) $$\overline{E} = \frac{1}{gR^2\overline{V}_a}\int_0^R r\overline{V}_a^3(r) + rV_a(r)\; V_t^2(r) + 2gr\; V_a(r)\; P_s(r)\; dr$$

Into summation form Eq 5 becomes

(6) $$\overline{E} = \frac{1}{gR^2\overline{V}_a}\sum_R\; rV_a^3(r) + rV_a(r)\; V_t^2(r) + 2gr\; V_a(r)\; P_s(r)\;\Delta r$$

or

$$\overline{E} = \overline{E}_a + \overline{E}_t + \overline{E}_p$$

The energy is evaluated from experimental measurements of V_a V_t and P_R

Appendix C

ANALYSIS OF HELICAL VANE

The assumptions are made that (1) the absolute velocity remains constant and is V_{a_0} and (2) the axial component of velocity varies inversely with the radius that is $V_a = Kr^{-1}$ = axial component of velocity (assuming there is no radial velocity) Then

$$(1) \qquad E_T = \underbrace{\frac{K^3\pi}{gQ}\int_{r_v}^{R} r^{-2}\,dr}_{E_a} + \underbrace{\frac{\pi K}{gQ}\int_{r_v}^{R} V_t^2\,dr}_{E_t} + \underbrace{P_R - \frac{2\pi K}{Q}\int_{r_v}^{R} P_s(r)\,dr}_{E_p}$$

Now from the assumptions

$$(2) \qquad V_t = V_{a_0}\cos\theta = \sqrt{V_{a_0}^2 - V_a^2}$$

where θ is the angle of absolute flow or

$$(3) \qquad V_t = \sqrt{V_{a_0}^2 - \frac{K^2}{r^2}}$$

Also to find $\int_{r_v}^{R} P_s\,dr$

$$P_s = \int_r^R \frac{1}{g}\frac{V_t^2}{r}\,dr = \frac{1}{g}\int_r^R \frac{\left(V_{a_0}^2 - \frac{K}{r}\right)^2}{r}\,dr = \frac{1}{g}\int_r^R \frac{V_{a_0}^2}{r}\,dr - \frac{1}{g}\int_r^R \frac{K^2}{r^3}\,dr$$

$$= \frac{1}{g}\left[V_{a_0}^2(\ln R - \ln r) + \frac{K^2}{2}(R^{-2} - r_v^{-2})\right]$$

or

$$(4) \qquad P_s = \frac{1}{g}\left(V_{a_0}^2 \ln R + \frac{K^2}{2R^2} - V_{a_0}^2 \ln r - \frac{K^2}{2r^2}\right)$$

So

$$(5) \qquad \int_{r_v}^{R} P_s\,dr = \frac{1}{g}\left(V_{a_0}^2 \ln R + \frac{K^2}{2R}\right)\int_r^R dr - \frac{V_{a_0}^2}{g}\int_{r_v}^{R} \ln r\,dr - \frac{K^2}{2g}\int_{r_v}^{R}\frac{dr}{r^2}$$

$$= \frac{1}{g}\left(V_{a_0}^2 \ln R + \frac{K^2}{2R}\right)(R - r_v) - \frac{V_{a_0}^2}{g}\Big[r\ln r - r\Big]_r^R + \frac{K^2}{2g}\frac{1}{r}\Big]_r^R$$

$$= M(R - r_v) - \frac{V_{a_0}^2}{g}\Big[(R\ln R - R) - (r_v \ln r_v - r_v)\Big] + \frac{K^2}{2g}\left(\frac{1}{R} - \frac{1}{r_v}\right)$$

Substituting Eqs 3 and 5 into Eq 1 gives

$$(6)\quad E_T = \frac{K^3\pi}{gQ}\int_r^R r^{-2}\,dr + \frac{\pi K}{gQ}\int_r^R \left(V_{a_0}^2 - \frac{K^2}{r^2}\right) dr$$

$$+ P_R - \frac{2\pi K}{Q}\left\{M(R - r_v) - \frac{V_{a_0}^2}{g}\left[(R \ln R - R) - (r_v \ln r_v - r_v)\right] + \frac{K^2}{2g}\left(\frac{1}{R} - \frac{1}{r}\right)\right\}$$

Equation 6 is evaluated in the following

$$(7)\quad E_T = \frac{K^3\pi}{gQ}(-1)\left(\frac{1}{R} - \frac{1}{r}\right) + \frac{\pi K}{gQ}\left[V_{a_0}^2 (R - r_v) + K^2\left(\frac{1}{R} - \frac{1}{r_v}\right)\right] + P_R$$

$$+ \frac{2\pi K}{Q}\left\{M(R - r_v) - \frac{V_{a_0}^2}{g}\left[(R \ln R - R) - (r_v \ln r_v - r_v)\right] + \frac{K^2}{2g}\left(\frac{1}{R} - \frac{1}{r_v}\right)\right\}$$

For the experimental model

Flow rate = 300 gpm of water = 0 669 ft^3/sec

Inside diameter of pipe = 5 in

Radius of helix hub = 3 in

Inlet area = 5 in^2

The solution for K is obtained by using Eq 2 and the second assumption to give

$$K = r\,V_0 \sin\theta$$

At R = 42 ft and θ = 17 deg

K = 0 208 (ft) × 19 3 (ft/sec) × 0 304

or

K = 1 24 (ft^2/sec)

Q = 0 669 ft^3/sec

R = 0 208 ft

r_v = 0 125 ft

Thus

$$\overline{E} = -\frac{K^3\pi}{gQ}\left(\frac{1}{R} - \frac{1}{r_v}\right) = \frac{-2\ 10\pi}{32\ 2 \times 0\ 67}\left(\frac{1}{0\ 208} - \frac{1}{0\ 125}\right) = 0\ 976 \text{ ft}$$

$$\overline{E}_t = \frac{\pi K}{gQ}\left[V^2{}_0 (R - r_v) + K^2\left(\frac{1}{R} - \frac{1}{r_v}\right)\right]$$

$$= \frac{3\ 14 \times 1\ 24}{32\ 2 \times 0\ 67}\left[368(0\ 083) + 1\ 64(-3\ 19)\right]$$

$$= 4\ 70 \text{ ft}$$

$$\overline{E}_p = P_R - \frac{2\pi K}{Q} P\ (r) = P_R - \left\{ M(R - r_v) - \frac{V^2_0}{g} [(R \ln R - R) - (r_v \ln r_v - r_v)] + \frac{K^2}{2g}\left(\frac{1}{R} - \frac{1}{r_v}\right)\right\}$$

where

$$M = \frac{1}{g}\left(V^2_0 \ln R + \frac{K^2}{2R}\right) = -\ 17\ 8$$

and thus

$$\overline{E}_p = P_R - \frac{2\pi K}{Q} \left\{-17\ 8(0\ 208 - 0\ 125) - \frac{368}{32\ 2} [(0\ 208 \ln 0\ 208 - 0\ 208) - (0\ 125 \ln 0\ 125 - 0\ 125)] + \frac{1\ 54}{64\ 4}\left(\frac{1}{0\ 208} - \frac{1}{0\ 125}\right)\right\}$$

$$= P_R - 1\ 74 \text{ ft}$$

For the energy balance

$$KE \text{ in inlet} = 5\ 72 \text{ ft}$$

Therefore

$$P_{(\ nl\ t)} + 5\ 72 - \text{losses} = E_T = 0\ 976 + 4\ 70 + P_R - 1\ 74$$

$$P_{(\ nlet)} - P_R = 5\ 66 - 1\ 74 - 5\ 72 + \text{losses}$$

$$P_{(\ nl\ t)} - P_R = -1\ 80 + \text{losses} = \Delta P \text{ across vane}$$

Experimental ΔP was found to be 0 55 ft of H_2O Thus the losses were 1 35 ft

Care must be exercised in any application of this analysis Obviously if r_v becomes small V gets very large and the calculation breaks down

The inlet velocity was 19 3 ft/sec so 0 23 velocity head was lost However the liquid must be greatly accelerated before it enters the helix and thus it introduces further losses

Appendix D

CALCULATION OF REQUIRED GAS SEPARATOR LENGTH

Equation 29*d* on p 25 of ref 2 that is

$$u = 0\ 825 \left(\frac{N_G^4}{g}\right)^{0\ 083} \left(\frac{\gamma g_c}{\rho_1}\right)^{0\ 250}$$

where

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

is used to obtain the bubble velocity

For the model

$$\gamma = 5\ 7 \times 10^{\ 3}\ \text{lb/ft}$$
$$\rho_1 = 62\ 4\ \text{lb/ft}^3$$
$$\rho_2 = 0\ 081\ \text{lb/ft}^3$$

Substituting these values in the above equation gives

$$u = 0\ 199 \left(\frac{V_t^2}{r}\right)^{0\ 332}$$

To get a conservative average bubble velocity

$$\bar{u} = \frac{\int_{r_v}^{R} u dr}{R - r} = \frac{\Delta r}{R - r_v} \sum_{r_v}^{R} \left(\frac{V_t^2}{r}\right)^{0\ 332}$$

the velocity profile at 14 in from the vanes was used and it was found that

$$\bar{u} = 1\ 84\ \text{ft/sec}$$

Now

$$R - r_v = 0\ 146\ \text{ft}$$

and thus the time for a bubble to get to the void is

$$\frac{0\ 146}{1\ 84} = 0\ 0794\ \text{sec}$$

If the axial velocity is assumed constant then

$$\bar{V} = \frac{Q}{\pi(R^2 - r_v^2)} = \frac{1\ 34}{\pi \times 0\ 0395} = 10\ 8\ \text{ft/sec}$$

The length of the separator is then

$$0\ 0794 \times 10\ 8 = 0\ 86\ \text{ft} = 10\ 3\ \text{in}$$

or about *two pipe diameters* It was visually observed that the last bubbles reached the void in about two pipe diameters