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Paul et al.

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[54] **ROTARY HEAT EXCHANGER AND APPARATUS**

[75] Inventors: **Richard S. Paul**, Grand Island; **David W. Weiler**, Lancaster, both of N.Y.

[73] Assignee: **Union Carbide Corporation**, New York, N.Y.

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[52] **U.S. Cl.** **165/1**, 165/86, 165/88, 165/110
[51] **Int. Cl.** **F28b 1/04**
[58] **Field of Search** 165/1, 86, 88, 89, 133, 165/90, 8, 165, 166, 164; 65/110

[56] **References Cited**

UNITED STATES PATENTS

2,799,259	7/1957	Farny et al.	165/86 X
3,170,512	2/1965	Smith	165/110
3,363,676	1/1968	Hunter	165/86
3,523,577	8/1970	Milton	165/133 X

FOREIGN PATENTS OR APPLICATIONS

35,421	1/1906	Switzerland	165/86
87,654	1/1921	Switzerland	165/86

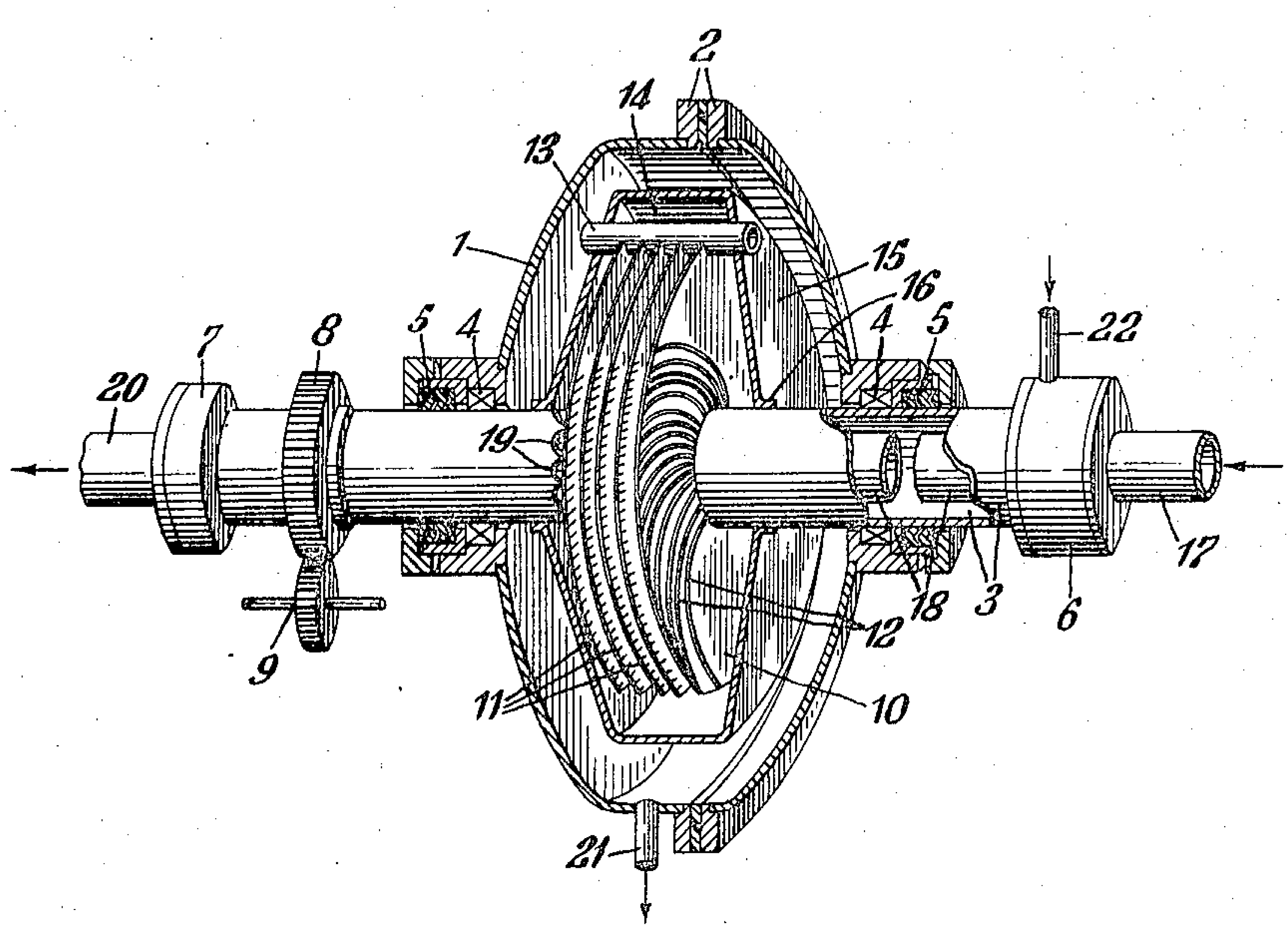
Primary Examiner—Samuel W. Engle
Attorney, Agent, or Firm—Harrie M. Humphreys

[57] **ABSTRACT**

The invention described herein may be manufactured and used by or for the Government for governmental purposes without payment of any royalty thereon.

Heat transfer to a condensing vapor is conducted in a rotating exchanger comprising a tightly-nested array of involute-shaped passages coated internally with a thin porous layer of bonded metal particles.

4 Claims, 10 Drawing Figures



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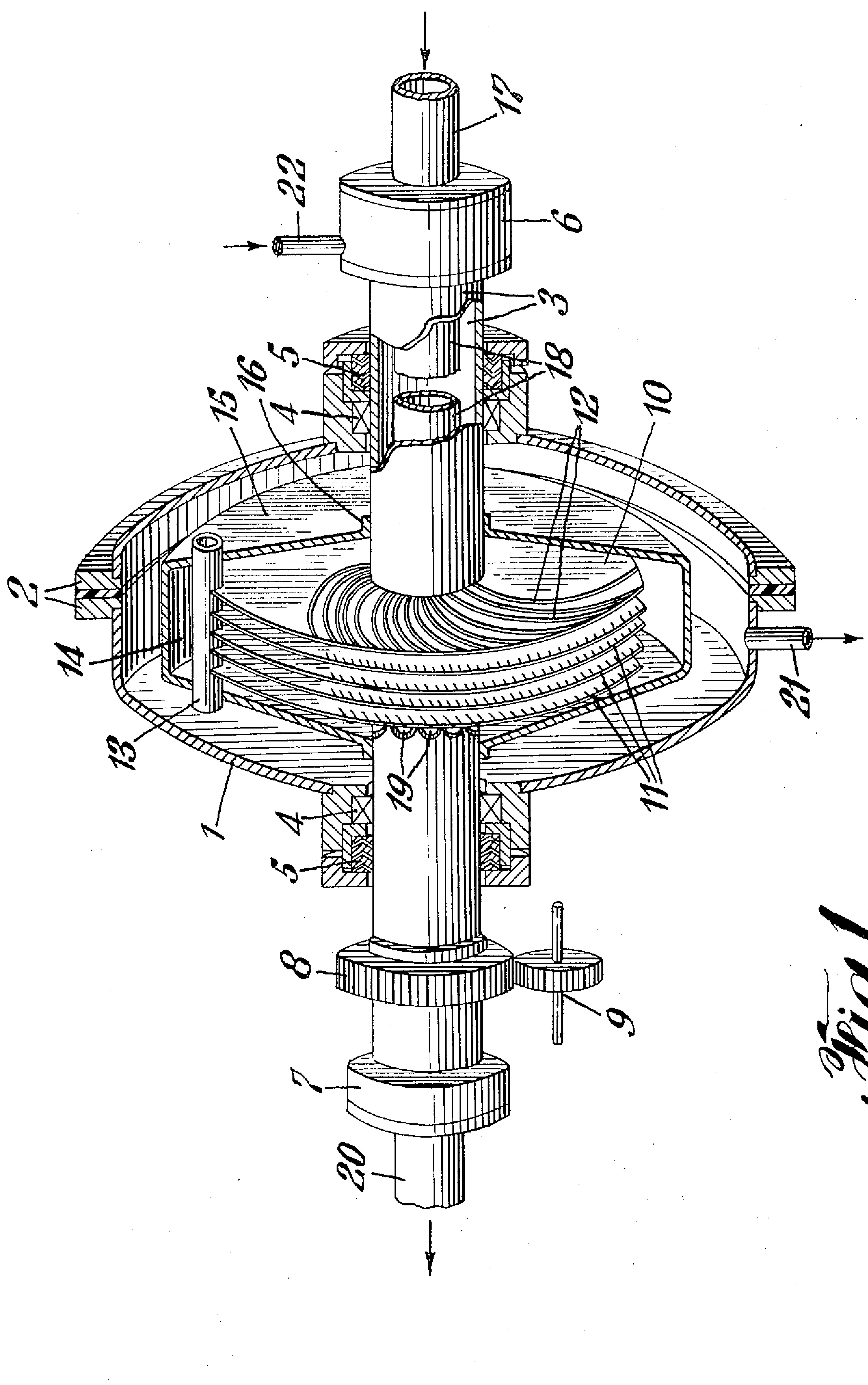

Fig. 1.

INVENTORS
DAVID W. WEILER
RICHARD S. PAUL
BY Harris M. Humphreys
ATTORNEY

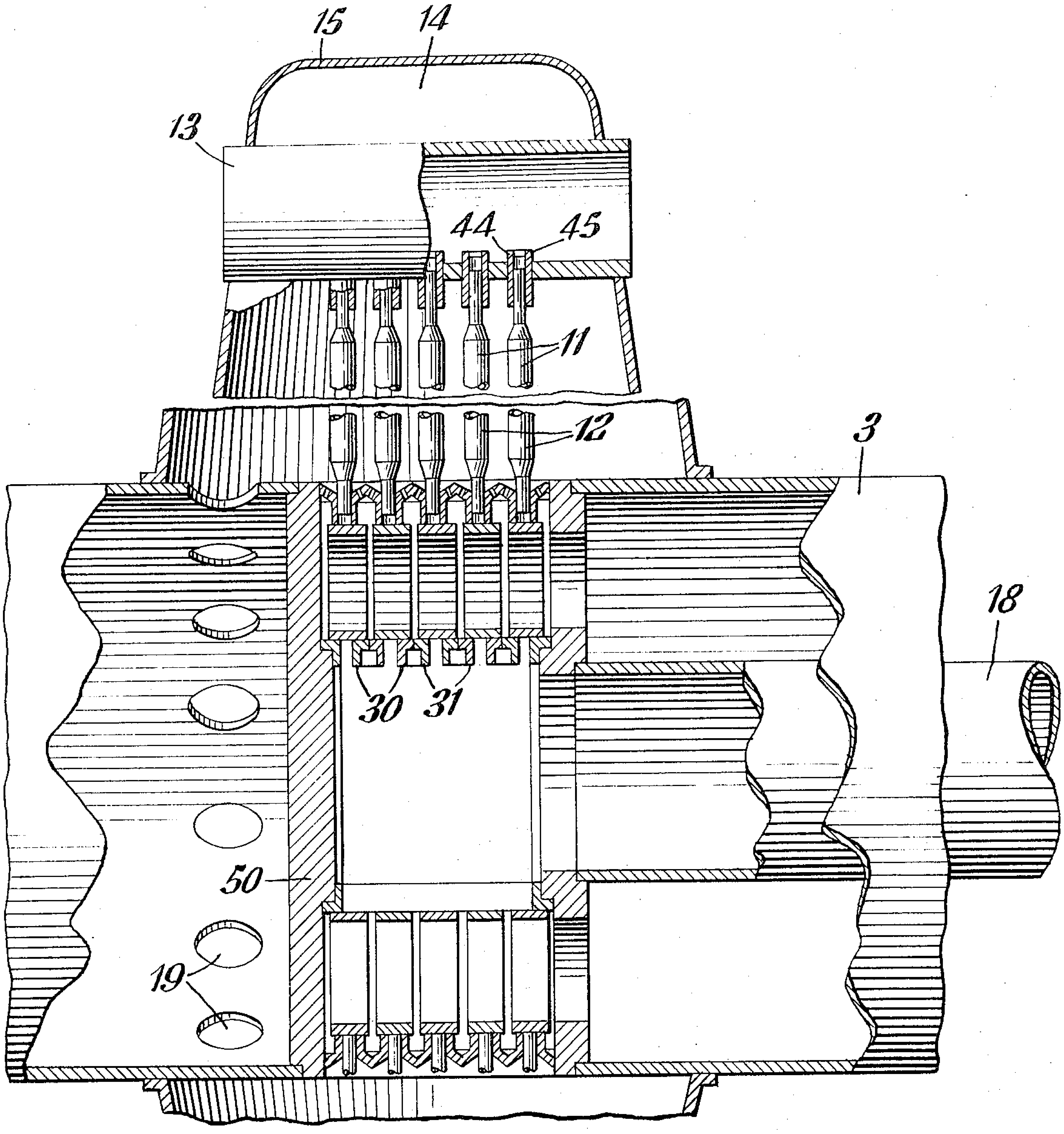

Fig.2

INVENTORS
DAVID W. WEILER
RICHARD S. PAUL
BY Harris G. Humphreys
ATTORNEY

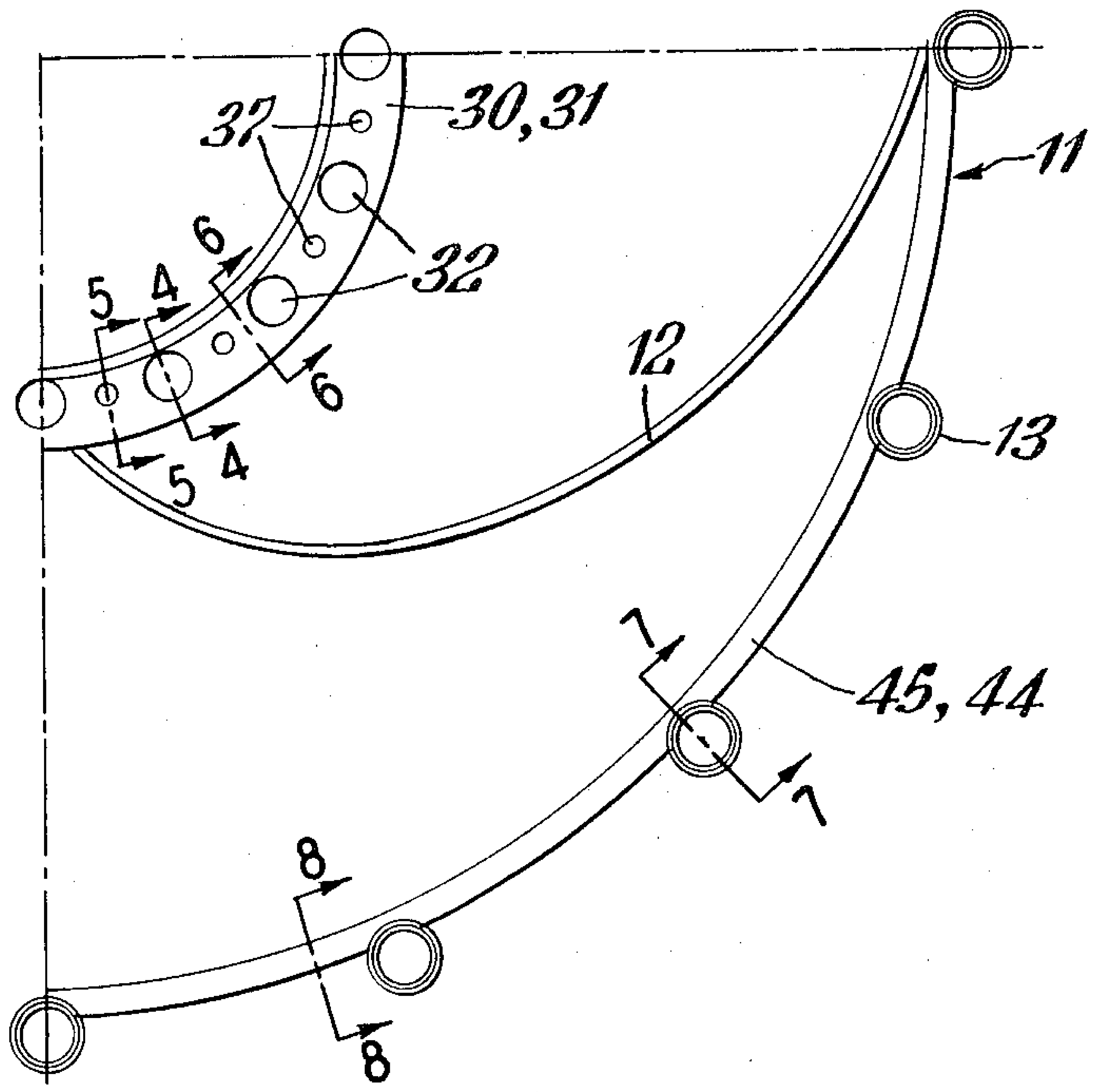

Fig.3.

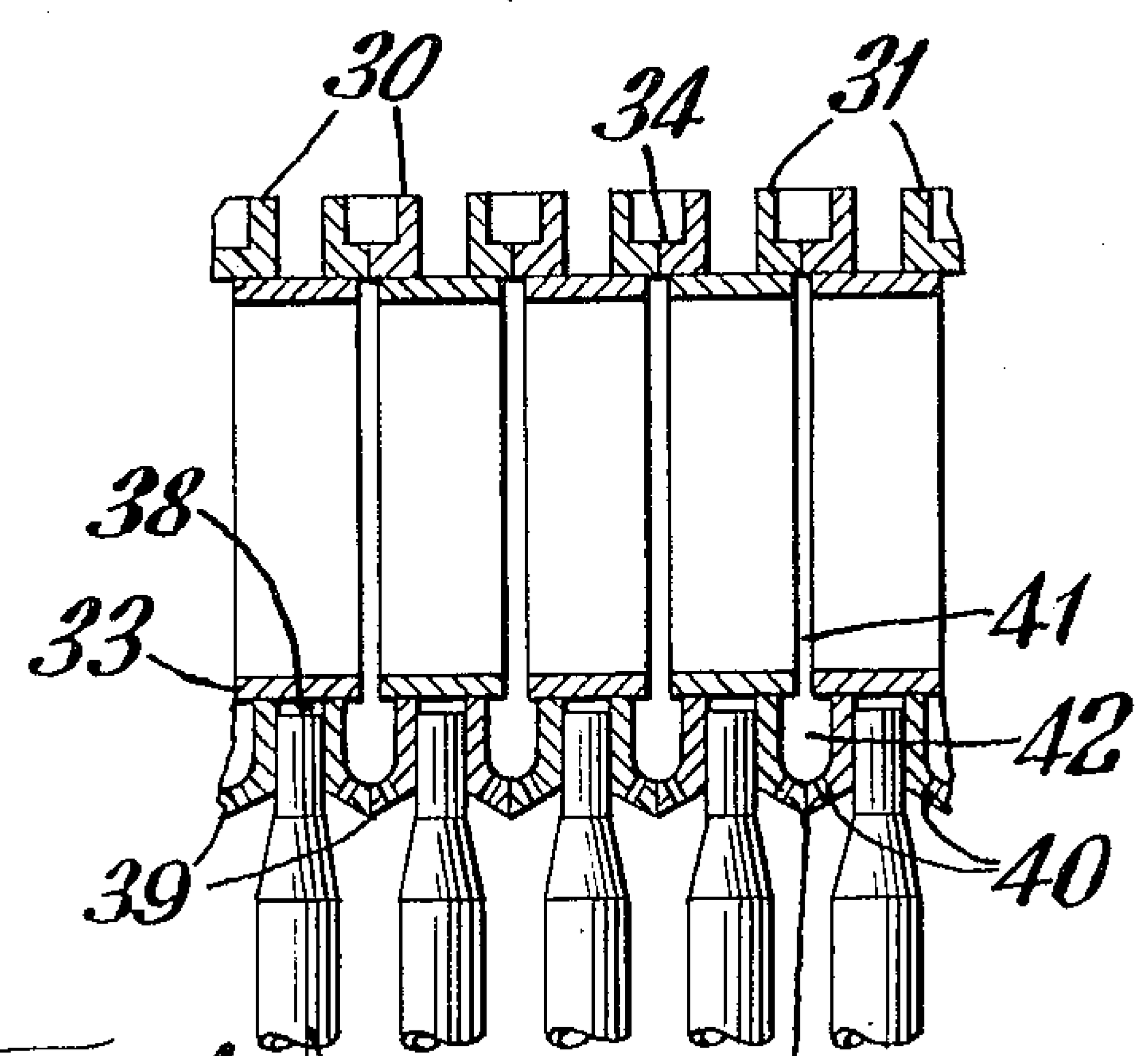

Fig.4.

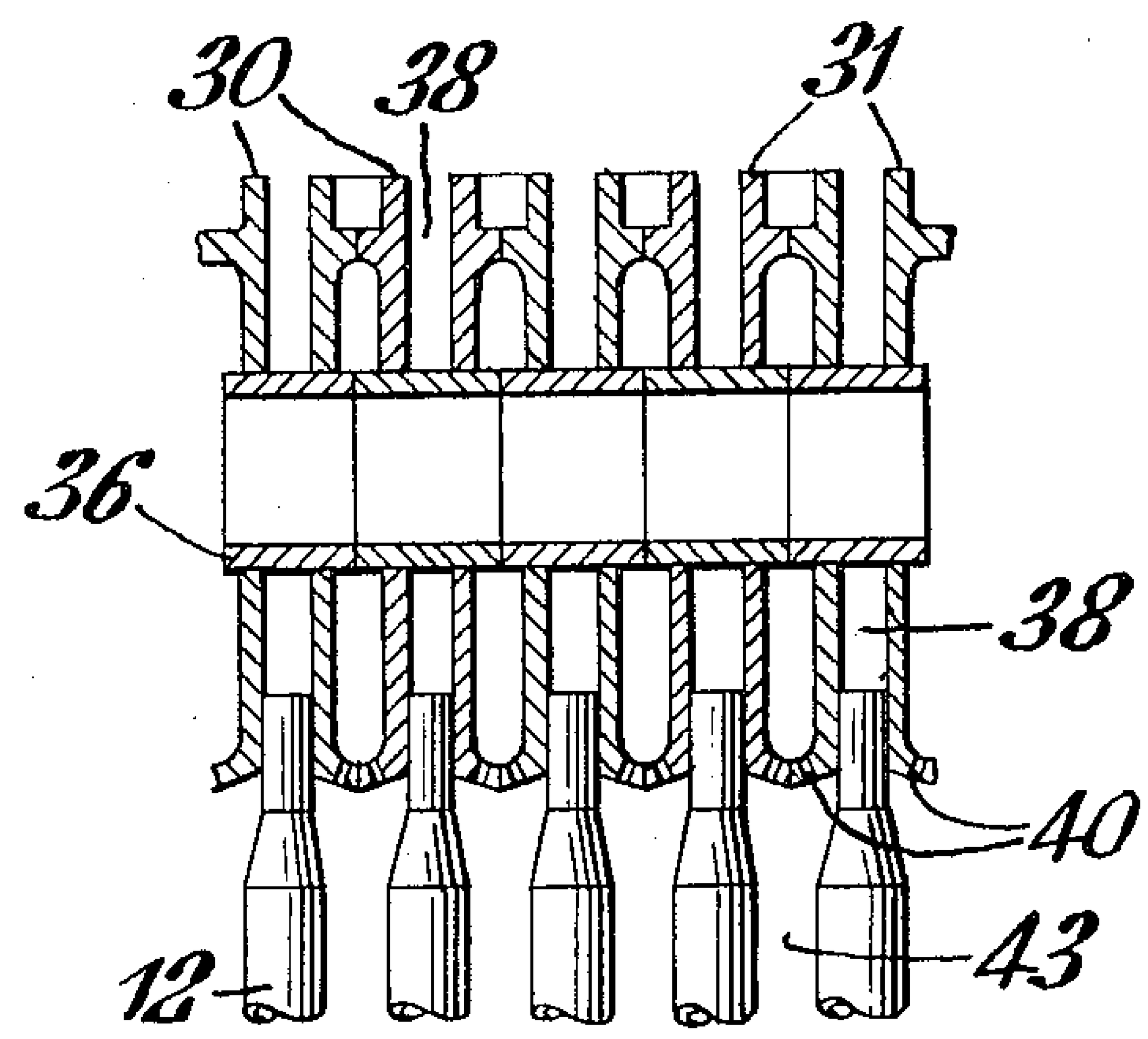

Fig.5.

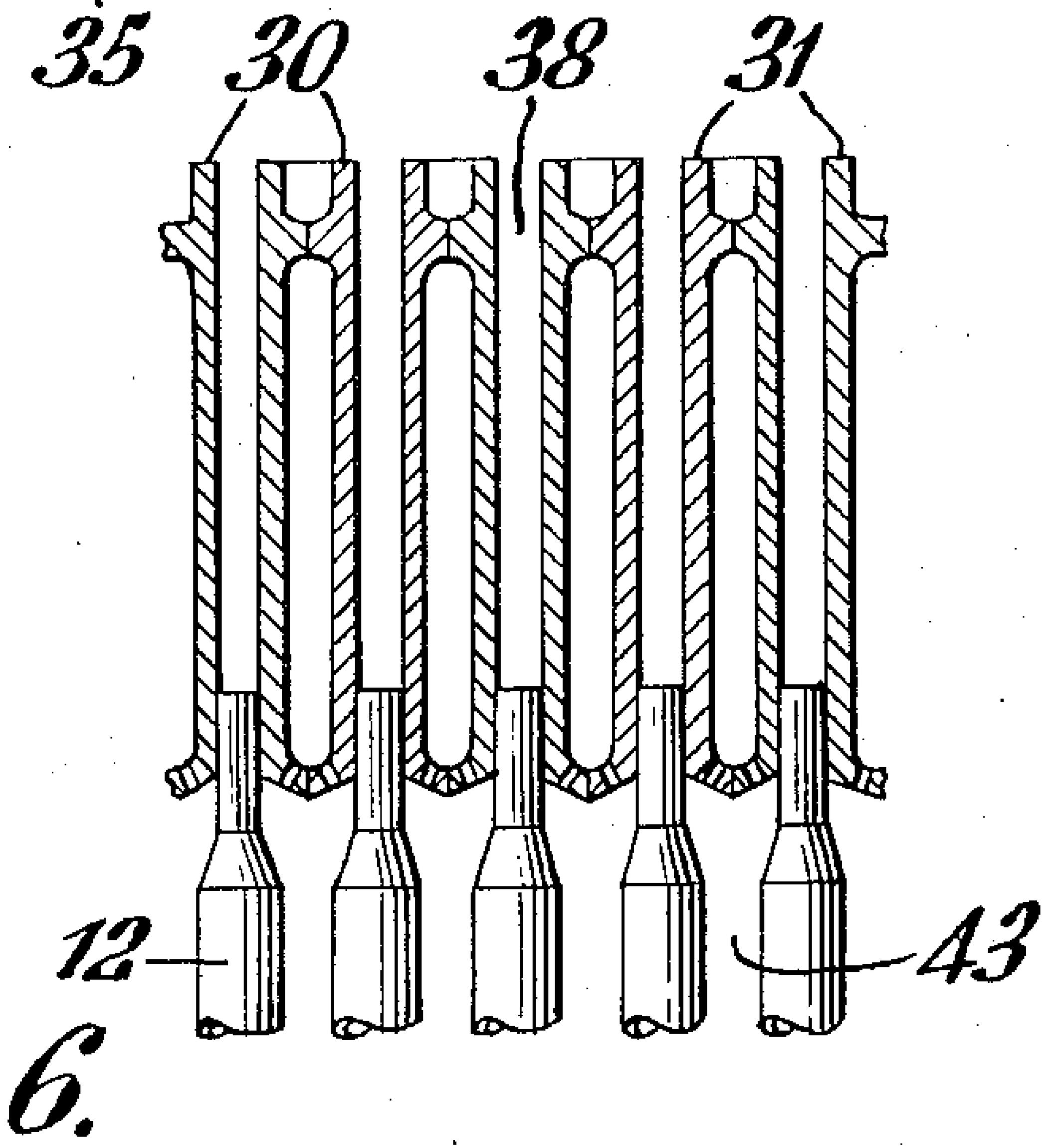

Fig.6.

INVENTORS
DAVID W. WEILER
RICHARD S. PAUL
BY Harris M. Humphreys
ATTORNEY

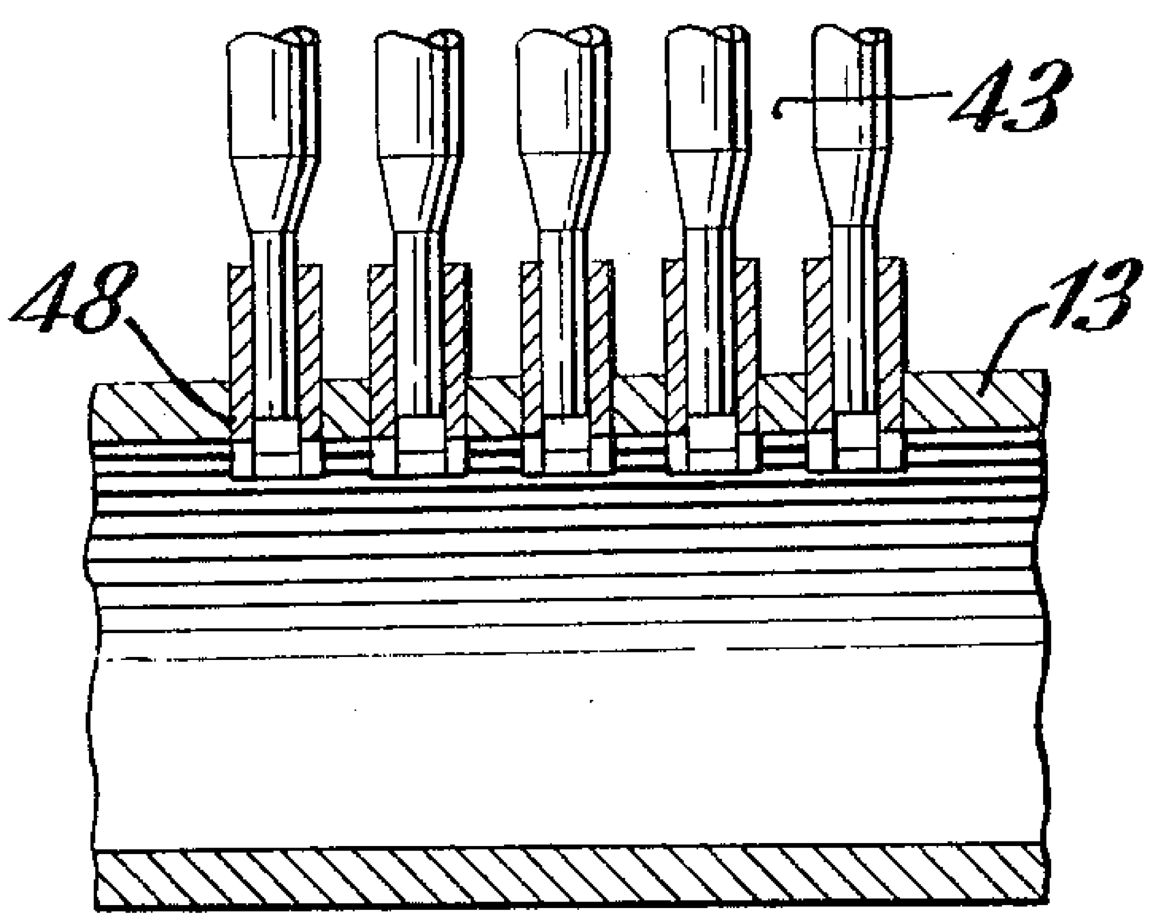

Fig. 7.

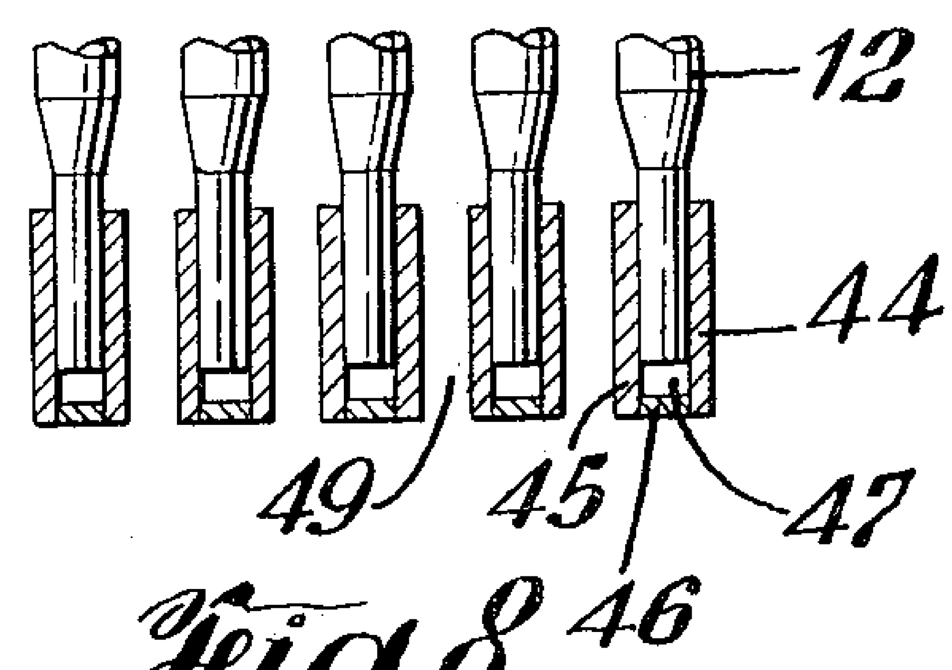

Fig. 8.

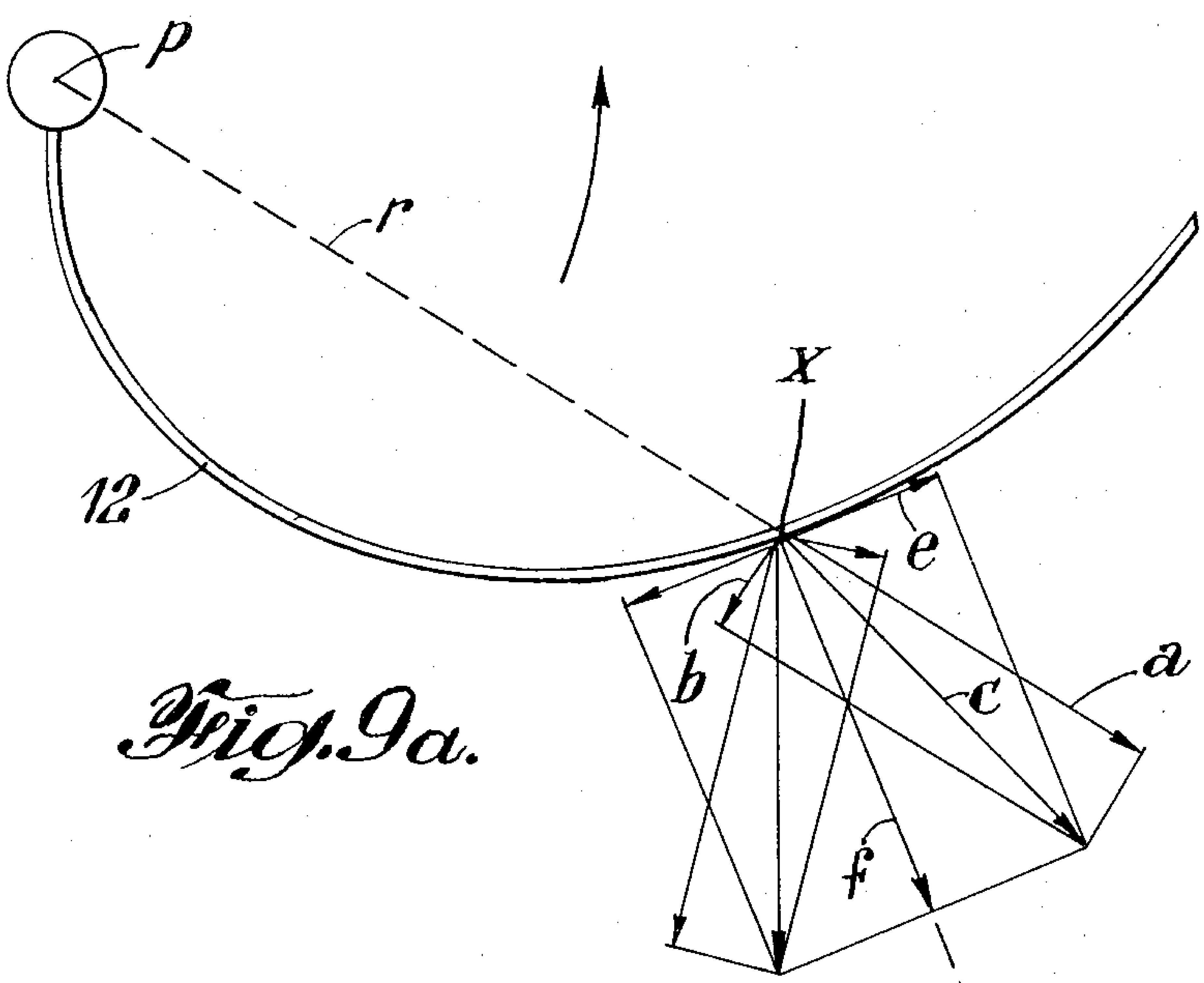

Fig. 9a.

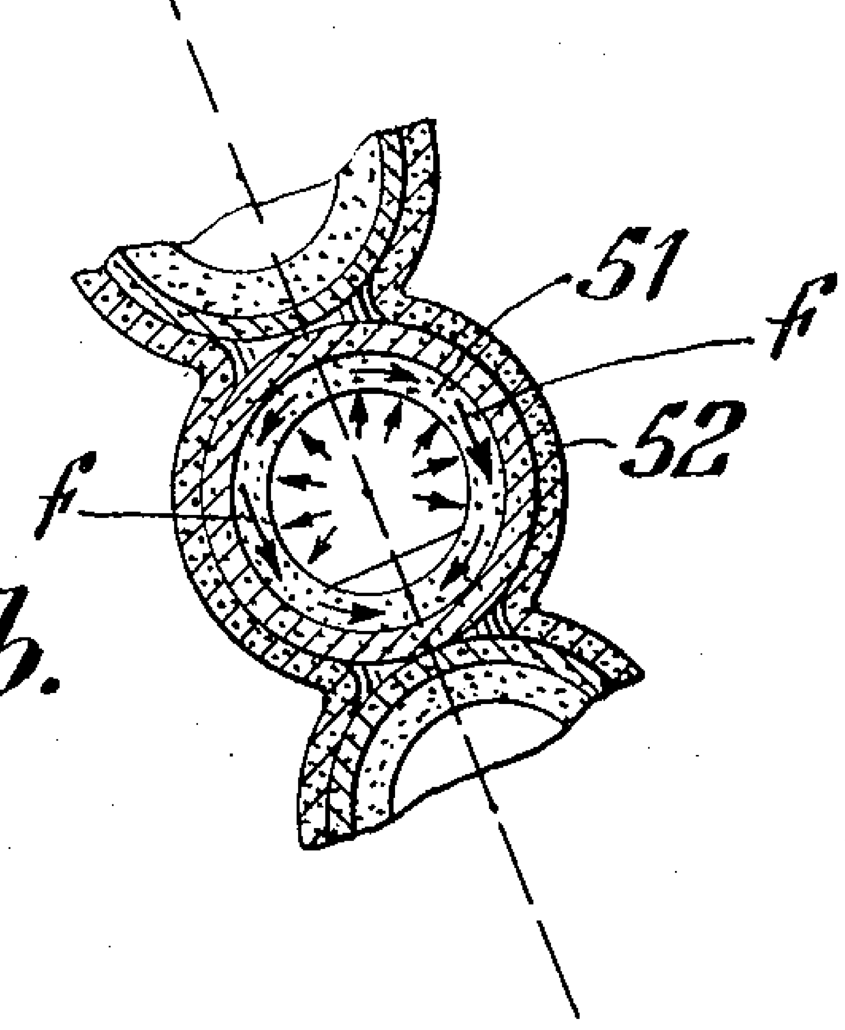

Fig. 9b.

INVENTORS
DAVID W. WEILER
RICHARD S. PAUL
BY
Harris M. Humphreys
ATTORNEY

ROTARY HEAT EXCHANGER AND APPARATUS

BACKGROUND OF THE INVENTION

It has long been an objective of heat exchange technology to increase substantially the coefficients of condensation. Heat transfer improvements in a condensing system encounter special problems because the condensed fluid adheres to the heat transfer wall, accumulates in heavy layers and seriously reduces the rate of heat transport from the condensible vapor to the wall.

Improvements in condensing coefficients are especially needed when a heat exchanger couples a boiling process and a condensing process. Substantial strides have been made in improving boiling coefficients with the result that the condensing coefficient now becomes the limiting factor in overall performance. With light volatile fluids for example, boiling coefficients in excess of 10,000 Btu/hr $\times$ ft^2 $\times$ °F can be obtained, while the usual condensing coefficients for such fluids are lower by an order of magnitude. The several resistances to heat flow between the fluids are added reciprocally and the overall coefficient tends to approach the lowest coefficient involved in the heat flow path. Therefore, even though a boiler-condenser is provided with an advanced boiling promoter, the condensing side will become the overwhelming resistance, and only marginal advantage can be realized from the high boiling coefficient.

The accumulation of condensate on the heat exchange wall of a condenser-boiler creates yet another dilemma in that, changing the heat flux (total heat transport Q per unit heat transfer area A) exerts opposed influences on the two respective coefficients. Increasing the heat flux promotes higher boiling coefficients but degrades the condensing coefficient. Thus, the condensing coefficient has inevitably become an obstacle in the path of improving overall performance of such exchangers.

Hydrophobic materials have been applied to the heat transfer wall so that the fluid will condense in droplets ("beads") rather than being distributed in a thick continuous film. A fraction of the wall area therefore remains essentially bare of liquid and is constantly exposed to the vapor without presence of a thick insulating film. Examples of such hydrophobic materials are noble metal platings (U.S. Pat. Nos. 3,289,753 and 3,289,754), plastic coatings (U.S. Pat. No. 3,207,209), and oils (McAdams "Heat Transmission," McGraw-Hill, Second Edition, 1942, page 276). Condensing coefficients can be materially improved by this means to 10,000 Btu/hr $\times$ ft^2 $\times$ °F or higher, but the procedure leaves much to be desired. Such coatings and additives are expensive and are often intolerable because they contaminate the condensing fluid. Thin coatings of such materials applied "permanently" to the wall soon erode away and the condensing performance degrades seriously. Finally, if high heat fluxes are obtained in long, narrow passages, the passages may flood with condensate despite the use of hydrophobic materials.

It has been proposed to "flute" the surface of a heat transfer wall, thereby providing closely spaced grooves of small dimension adapted to draw the condensate into the grooves by surface tension. The condensate then flows by gravity along the groove to a point of removal from the wall. The condensate is stripped from the ridges lying between the grooves by the surface tension forces, and the ridges thus become "dry" low resistance areas for heat transport from the vapor. (Ref: Gregorig, R., "An Analysis of Film Condensation On Wavy Surfaces Including Surface Tension Effects," ZAMP, Vol. 4, pages 40–49) Condensing coefficients of 10,000 Btu/hr $\times$ ft^2 $\times$ °F at $\Delta T = 1.5$ °F have been reported. Again, this is often not an ideal method for promoting condensation. Such grooves are not only tiny but the profile of the groove is critical and they are quite expensive to produce. The cost problem is especially acute when the condensing surface is intricate or relatively inaccessible, e.g., the inner surface of a small tube. Moreover, a groove must collect and transport all the condensate produced on a crest throughout the full length of the groove without flooding. The liquid handling capacity of the tiny groove is quite limited and only a short groove will operate effectively without flooding at high heat flux.

It has also been proposed to rotate a cylindrical or conical heat transfer wall about its axis of symmetry and to condense a vapor on the outer surface. The condensate film is thus disposed on a plane normal to the field, and the high gravity (high-G) field in the radial direction induces instability in the film. The layer of liquid is expelled from the surface leaving a film of much reduced thickness and resistance. (Ref: Birt, D.C.P. et al., Transactions of the Institution of Chemical Engineers, Vol. 37, 1959, page 289) Values of condensing coefficients up to 8000 Btu/hr $\times$ ft^2 $\times$ °F have been reported at 22-G's on a 1-inch diameter tube. Unless the diameter is small, the surface area of a cylinder is low in relation to its volume, and the complexity of spinning a multitude of small cyclinders individually about their axis is impractical. Despite the higher coefficients achievable by the procedure, the overall effectiveness per unit volume of heat transfer device is severely limited.

It is an object of this invention to provide method and apparatus for condensing vapor at extremely high rates.

Another object is to provide method and apparatus for achieving high heat transfer rates between a condensing fluid and a boiling liquid.

Still another object is to provide an improved rotating heat exchanger for condensing a vapor, which exchanger is compact and low in weight.

Other objects and advantages of this invention will be apparent from the ensuing disclosure and the appended claims.

SUMMARY

In the present invention, a condensing process is conducted on the surfaces of a porous layer within a high-G field. The interstices of the porous layer are interconnected in all directions and the porous layer is bonded, e.g., by thermal process to the walls of a plurality of small passageways comprising a heat exchanger. The passageways are oriented within the high-G field such that the condensate which forms on the surfaces of the porous layer flows in a generally involute path parallel to the axis of the passageways and is discharged from the ends thereof. The condensate flows into the porous layer in a direction normal to its plane, flows in a direction lateral to the high-G field toward a zone of condensate accumulation and flows in a direction parallel to the high-G field toward a point of discharge from the heat exchanger walls.

In preferred practice, the condensing process is conducted inside the tubes and the porous layer is bonded substantially coextensively to the inner surface of the tube. Each tube is curved in involute-shape and with a linear profile such that a plurality of tubes in a circular array will nest tube-against-tube along essentially the full length of each tube. A solid disk composed of tubes is thereby formed and is arranged to be spun about an axis normal to its plane to create the high-G field.

In the preferred boiler-condenser embodiment of the invention, the external surface of the disk of tubes is coated substantially coextensively with a second porous layer. A boiling process is conducted on and within the second porous layer, thereby absorbing the heat transmitted to the heat exchanger wall by the condensing process.

The method of the invention comprises the steps of providing a first fluid in vapor form at near its condensing temperature, flowing the first fluid along an involute-shaped channel enclosed by a gas-impervious wall from an inlet end toward a discharge end, in indirect heat exchange with a colder fluid, rotating the involute-shaped channel with the inlet end nearest the center of rotation and the discharge end more remote from the center of rotation, thereby producing a high radial gravitational force on the first fluid having a component aligned parallel to the axis of the flow channel, and condensing the first fluid along the involute-shaped channel on the surfaces of a porous heat conductive matrix layer interposed across the flow of heat from the first fluid to the wall. The liquid resulting from condensation flows into the porous layer by capillary force, and flows generally parallel to the plane of the porous layer toward a liquid collecting zone along the flow channel under the influence of the normal force component. At least a portion of the liquid in the collecting zone is freed of the capillary force and flows toward the discharge end of the involute-shaped channel under the influence of the parallel force component.

The apparatus of the invention comprises a fluid-tight, involute-shaped conduit having an inlet end of greatest curvature and an outlet end of least curvature and being provided with a layer of inter-bonded metal particles afixed substantially coextensively to the inner wall. Means are provided for rotating the conduit in the plane of the conduit with the inlet end nearer the center of rotation. Further means are provided for introducing a fluid to the conduit inlet and for withdrawing the fluid from the outlet.

In the embodiments, an envelope is provided surrounding and spaced apart from the conduit and rotating therewith, thereby defining a first fluid flow passage within the conduit and a second fluid flow passage without the conduit. Means are provided for introducing first and second fluids separately to the inlet ends of the first and second flow passages, and for withdrawing the fluids separately from the flow passages.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. 1 is an isometric view in partial section of a rotating heat exchanger suitable for practicing this invention.

FIG. 2 is a sectional view through the rotating envelope of the apparatus of FIG. 1 showing method and means for conducting fluids into, through and from the rotating envelope.

FIG. 3 is a partial plan view of a heat exchanger disk element employed in the apparatus of FIGS. 1 and 2.

FIGS. 4–8 are partial cross-sectional views through a multiple-disk exchanger showing details of manifolding the fluid passages thereof.

FIG. 9*a* is a schematic representation of a typical fluid passage in the rotating heat exchanger showing forces exerted by the fluid in the passage due to rotation.

FIG. 9*b* is an enlarged cross-section through the passage of FIG. 9*a* showing fluid movement and disposition within the passage.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIG. 1, a reboiler-condenser embodiment of the invention is illustrated in a partial cross-section of an isometric view. The exchanger 10 is enclosed within fixed casing 1 which is separable at flanges 2 and is suitably supported on a rigid frame and base (not shown). A hollow shaft 3 extends through axially aligned openings in casing 1 and is adapted for fluid-tight rotation therein by means of bearings 4 and seals 5. Additional rotary fluid-tight seals 6 and 7 at the ends of shaft 3 permit the introduction and withdrawal of fluids through the rotating shaft, to-and-from the exchanger. Driving means such as gear 8 fixed to the rotating shaft is coupled to a suitable shaft power source 9.

A fluid may be introduced through fixed conduit 22 into rotary seal 6 and thereby gains entry into hollow rotating shaft 3. Another fluid may be introduced through fixed conduit 17 into rotary seal 6 and thereby gains entry into rotating conduit 18 concentric within rotating shaft 3. Thence, the separate fluids flow to the heat exchanger and are distributed uniformly over the heat transfer surfaces as hereinafter described.

The heat exhanger 10 consists of a plurality of disk-shaped elements 11, later to be described in detail. The disk-shaped elements each contain many involute-contoured passages 12 extending from an inner hub manifold of the disk to an outer rim manifold of the disk — these manifolds to be identified and described later herein. The rim manifold connects into transverse fluid collectors 13 spaced at frequent intervals around the disk circumference (only one such collector being shown in FIG. 1 for simplicity). Thus, a fluid may be introduced via hollow shaft 18 into the inner hub manifold, distributed among and through the involute passages, discharged from the passages into the outer rim manifold, and then discharged into collectors 13.

The plurality of disk-shaped elements 11 are sealed within rotating envelope 15 attached to the rotating shaft at 16. Transverse fluid collectors 13 pass sealingly through envelope 15 and are open-ended so that fluid received by collectors 13 can flow laterally from the open ends thereof and be collected in fixed casing 1. Fluid thus collected in casing 1 may be withdrawn via conduit 21.

The disk-shaped elements 11 are spaced axially apart from one another with means provided near the center for distributing a fluid over external surfaces on both sides of each disk. This fluid travels generally radially over the disk outer surfaces and is discharged at the outer periphery into space 14 within rotating envelope 15. From space 14, the fluid may flow inwardly between the heat exchanger 10 and the wall of rotating envelope 15 to openings 19 provided in hollow shaft 3.

Thence, the fluid flows axially through rotary seal 7 and is withdrawn via fixed conduit 20.

The method of introducing and withdrawing fluids to and from the appropriate passages of the heat exchanger is illustrated in FIG. 2 supported by FIGS. 3–8. Involute tubes 12 comprising a disk 11 terminate at their inner ends between a pair of rings 30 and 31, which together with the ends of tubes 12 comprise the inner hub manifold. A quadrant of a disk with rings in place is shown in plan view in FIG. 3. Holes 32 are drilled through the inner ring at uniform intervals around the circle and a short tube nipple 33 (FIG. 4) is inserted through each hole and bonded in place. The resulting disk and ring sub-assemblies are stacked one upon the other with surfaces 34 and 35 between adjacent inner rings in smooth contact. Additional short tube nipples 36 (FIG. 5), slightly longer than nipples 33, are bonded in other holes 37 spaced uniformly around the inner ring. With tubes 36 aligned through the stack of disks, bolts (not shown) are inserted through the tubes 36, and nuts are applied to clamp the stack tightly together.

It will be noticed that tubes 12 are not inserted between rings 30, 31 deeply enough to contact tube inserts 33 or 36. Therefore, the gap or hub manifold 38 formed by rings 30, 31 extends all around the tube inserts, as shown clearly in FIGS. 4, 5 and 6. Thus, fluid flowing through the inner hub manifold 38 and into the tubes 12 is not obstructed by tube inserts 33 and 36.

It will also be noted that the outer edges 39 of rings 30, 31 are flared away from their associated disk and are provided with orifices 40 at frequent uniform intervals around and through the outer lip of the rings. Thus, fluid flowing axially through tube inserts 33 may pass through the gaps 41 between the ends of tube inserts 33, into space 42, thence through orifices 40 into the space 43 between disks 11.

The outer ends of involute tubes 12 comprising a disk 11 terminate between another pair of rings 44, 45 as shown in FIGS. 7 and 8. Rim 46 is bonded between the outer edges of rings 44, 45. Tubes 12 are not inserted between rings 44, 45 deeply enough to contact rim 46, thereby leaving annular space 47 for circumferential flow of fluid. The ends of tubes 12 together with rings 44, 45 and rim 46 comprise the outer rim manifold for disk 11.

FIG. 7 is a section taken through the fluid collector 13 showing its juncture with the outer rim manifolds of disks 11. At the intersection with each disk, the collector tube has a milled slot whose width is equal to the disk thickness and whose depth is somewhat greater than the wall thickness of collector 13. The edge of the disk is "notched" to the contour of the inner surface of collector 13, thereby removing a short section of rim 46 and opening annulus 47 into collector 13. The interfitting surfaces 48 of disks 44, 45 and collector 13 are bonded together, thereby preventing the leakage of fluid from collector 13 into space 43 between the disks.

To summarize the flow paths through the system, it is seen that a first fluid entering fixed conduit 22, rotary seal 6 and rotating shaft 3 may pass axially through tube inserts 33 and radially through gaps 41, space 42 and orifices 40. Here the first fluid is distributed across the external surfaces of disks 11 and thereafter flows radially through spaces 43 and 49 into space 14 within envelope 15. Finally, it flows through apertures 19 into rotating shaft 3 and is withdrawn through rotary seal 7 and fixed conduit 20.

A second fluid entering fixed conduit 17, rotary seal 6 and rotating conduit 18 may pass through inner hub manifold 38 and be distributed among tubes 12 comprising each disk 11. Thereafter, the second fluid flows through tubes 12 in contact with the internal surfaces thereof, discharges into rim manifold space 47 and flows circumferentially to the point of intersection with a collector tube 13. Finally, it flows laterally out the ends of collector 13 into casing 1 and is withdrawn through conduit 21.

It will be noted (FIG. 2) that hollow shaft 3 is not continuous through exchanger 10, but terminates on one side thereof and continues from the other side. Fluids entering through rotating shaft 3 and concentric conduit 18 are prevented from mixing with fluid discharging through apertures 19 by solid wall 50 sealed transversely across the continuation of shaft 3.

As described previously, each disk of the heat exchanger consists of an array of involute-shaped tubes, formed to a contour such that each tube is approximately in line-contact throughout its length with the adjacent tubes on either side thereof. The tubes of the array are bonded together at their lines of contact to form a solid, rigid disk. All tubes terminate between, and are sealed to, ring members which comprise a hub manifold at the inner ends of the tubes and a rim manifold at the outer ends. Whereas individual tubes may be circular in cross-section throughout most of their length, it is preferable to form the ends of the tubes covered by the ring members into rectangular shape so that all sealing surfaces in the manifold zones will be flat and in smooth contact. This will produce a stronger structure, less likely to develop leaks.

According to generally accepted definition, an involute is a curve generated from an origin located on a reference circle such that the distance from any point "X" on the involute along a straight line to the point of tangency with the reference circle is equal to the arc length from the origin of the involute around the reference circle to the point of tangency. A common example of an involute is the locus of a point on a taut thread as it is unwound from a cylindrical spool.

The involute-shape appropriate for the tubes of this invention reflects the requirement that the tubes nest one against the other in line-contact along the tube length, thereby producing a solid disk. The equation for the curve which fits this requirement is:

$$\theta = \cos^{-1} (r_i/r) - \sqrt{(r/r_i)^2 - 1}$$

where

r_i = radius of the inner circle formed by the ends of the nested tubes

r = radius (from the center of the inner circle) to any point on the curve

θ = angle between radii r and r_i (radians)

The radius of curvature to points along a tube formed according to the foregoing requirements will increase proceeding from the end adjacent the inner circle toward the remote end falling on the outer circle (circumference of the disk). Hence, the "curvature" of the tube at any point, defined as the reciprocal of the radius of curvature, is greatest at the end adjacent the inner circle and is least at the end falling on the outer circumference of the disk.

The internal surfaces of a disk (and if desired, the external surfaces thereof, with exception of surfaces in the manifold zones) are provided with a thin porous matrix layer comprised of metal particles which serve to promote heat transfer. (See items **51**, **52** of FIG. 9*b*.) By "porous" is meant a network of interconnected pores, cavities or interstices forming capillary channels for fluid flow into and through the porous layer in all directions. The porous layer may be deposited or coated on the surfaces from a slurry containing the metal particles. Thereafter, the coating is dried and the particles are bonded at particle contact points with other particles and with the disk surfaces, as by sintering, brazing or soldering.

The metal of the tubes and of the porous layer is preferably highly heat conductive. Aluminum and its alloys are preferred materials because they combine high heat conductivity and low weight. However, copper and its alloys are also excellent materials. Where other factors such as corrosion resistance are important, metals such as stainless steel and Hastelloy may be dictated. The metal should be capable of producing and maintaining a sound bond, and if different metals are chosen for the tubes and the porous layer, such metals must be compatible in this respect.

If the heat transfer wall and porous layer are of aluminum, the porous surface may be prepared using a slurry containing special fluxes to remove aluminum oxide from the surfaces to be bonded. For example, a suitable slurry may be prepared from the following solid ingredients:

	grams
Aluminum powder (270–115 mesh)	100
Lithium Chloride	13½
Sodium Chloride	22½
Potassium Chloride	27
Lithium Fluoride	6
Zinc Chloride	36

The above solids in powder form are mixed with about 75 ml. methyl alcohol, the exact amount of liquid being chosen or adjusted to provide a slurry consistency which is easy to handle and which produces a uniform, non-sagging coat of the desired thickness. The aluminum powder should be preliminarily cleaned and a preferred procedure is to (1) degrease with acetone, (2) dry, (3) rinse at room temperature with a solution of one part concentrated H_3PO_4 in four parts water, (4) rinse in water, and dehydrate with methyl alcohol. The slurry should be used soon after preparation as it degrades upon standing.

In the foregoing recipe, zinc bromide may be substituted in whole or part for the zinc chloride and will usually reduce the melting temperature of the flux. These zinc compounds decompose during the brazing process — the halide assisting in the removal of aluminum oxide and the elemental zinc alloying with the aluminum.

A small amount of fine powder (e.g., 10 gms) is preferably included with the dry ingredients of the foregoing recipes to increase the quantity of low melting bonding metal available for alloying the particles together.

All surfaces of the disk to be coated should be thoroughly cleaned immediately prior to application of the slurry. A preferred procedure is to (1) degrease in a solvent such as trichloroethane, (2) dry, (3) dip in a 5% Na OH solution for 30 seconds at 130°F, (4) rinse in cold water, (5) dip in a 30% HNO_3 solution for 60 seconds at room temperature, (6) rinse in hot water, and (7) air dry.

A suitable sequence of steps for fabricating the heat exchanger disk is to first assemble and bond the tubes together, then apply and bond the porous layer to all surfaces, next refinish the flat faces of the disk at the tube ends in the manifold zone, and finally bond the rings in place which form the hub and rim manifolds.

For the first of the above steps, all the tubes of a disk are positioned and clamped together in contact, and aluminum brazing alloy in the form of powder or preferably wire is laid over and along the lines of contact between tubes. The assembly is then immersed in a molten salt bath using procedures well known in the art. The brazing alloy should be chosen with a relatively high melting point, e.g., 1100°F to 1140°F. After brazing, the disk is thoroughly washed to remove residual flux.

For the second step, the bonded disk-of-tubes is precleaned as described above and then dipped in a tank of the aforementioned slurry, whereby both inside and outside surfaces are coated in one operation. While immersed in the slurry, the disk is rotated slowly about its axis in such direction that the outer ends of the tubes scoop the slurry into the tubes. The rotation is continued while withdrawing the disk from the bath so that excess slurry drains freely and uniformly from the surfaces. The slurry coating sets quickly into a self-supporting layer which should preferably be between 0.005 inch and 0.015 inch thick. The coating is dried in circulating air at 200°F for 30 minutes and is then brazed in a furnace preheated to 1000°F to 1050°F for 45 to 60 minutes. After removal, the disk is washed thoroughly, first by impingement of strong jets of hot water, then by 5-minute immersion in a hot 20% HNO_3 solution, and finally with water. The disk is finally dried in air.

For the third step, the flat faces of the disk in the manifold zones are refinished by machining away the porous layer. The depth of machining should be sufficient to expose solid metal and to remove any distortion which may have resulted during brazing.

For the fourth step, rings **30**, **31**, **44** and **45** are bonded against the refinished surfaces of the disk. A low temperature bond should be chosen, such as a low melting aluminum solder. For ambient or cryogenic service, a strong organic resin such as epoxy is adequate. Rim **46** should also be assembled between rings **44**, **45** and bonded during this step. Fasteners such as bolts with nuts may be applied in holes **37**, and when these are employed, the ring-to-disk bond serves primarily as a leak sealant.

It will be noted that the three bonding steps used in disk fabrication occur at progressively lower temperatures. Thus, the temperature of a bonding step does not damage or weaken a bonded joint previously produced.

For copper or copper alloy construction, essentially the same fabrication procedure can be followed as for aluminum except for the formation of the porous layer. Appropriate choices of bonding methods are required for tube joining, powder bonding and manifold ring bonding so that progressively lower temperatures are employed in successive bonding operations.

Porous layers of copper or copper alloy may be formed by sintering. A preferred procedure is described in U.S. Pat. No. 3,384,154 to R. M. Milton and employs metal particles, 1 to 50 microns in size, mixed with a plastic binder and a solvent to form a slurry which is then applied to the heat transfer wall in the desired thickness of coating. A suitable plastic binder is an isobutylene polymer having a molecular weight of about 140,000 and known commercially as Vistanex. The solvent must be compatible with the binder and suitable solvents for Vistanex are kerosene or carbon tetrachloride. The components are mixed to provide a uniform viscous slurry with a metal-plastic weight ratio of about 92 to 1. The base metal must be free of grease, oil and oxide coating to obtain proper bonding. Immediately before coating, the surface may be flushed with the plastic solution to facilitate wetting by the slurry.

The disk may be coated with the above slurry using a procedure as described for aluminum, after which excess slurry is drained away from the surface and the adherent coating is air-dried to remove the bulk of the solvent by evaporation. This leaves a solid self-supporting layer held in place by the binder. The base metal and the coating are blanketed by a mildly reducing atmosphere, and the temperature is raised for a sufficient time to sinter the particles together and to the base metal. The circulating reducing gas removes the thin oxide film and also purges away the products of decomposition of the binder. In the case of copper, the coating is sintered at about 180°F below its melting point or about 1760°F. Copper alloys which melt at lower temperatures should obviously be sintered at temperatures appropriately below their melting points. It is also apparent that the metal of the powder should be chosen to be sinterable at a temperature safely below the melting point of previously bonded joints between the tubes.

While the disks have been described as being fabricated of an assembly of separate, involute tubes, it is within the scope of the invention to form the array of involute passages from a pair of sheet metal disks which are bonded together, face-to-face. Thus, two sheet metal disks are stamped or otherwise formed with patterns of involute grooves which are identical except that one pattern is a mirror image of the other. Then the disks are bonded together along the ridges which separate the grooves, thus producing an array of tubular involute passages internal of the disk.

The hydraulic effect of spinning the involute tube is shown in FIGS. **9**a and **9**b. Item **12** represents a tube and point p its center of rotation. Radius r can be drawn to any point x along the tube. The forces which the liquid at this point exerts on its supports are represented by vector a aligned with the radius, and vector b normal to the radius and opposite to the direction of rotation. The resultant c can be resolved into components e and f, parallel and normal respectively to the tube tangent at point x.

Component f drives the liquid as rapidly as it is condensed, around the inner circumference of the tube to the sector of the cross-section most remote from the center of rotation. This flow is within the porous layer **51** — the liquid being held within the porous layer by capillary forces. Component e drives the liquid along the tube toward the discharge end. In the enlarged cross-section of the tube, FIG. **9**b, component e acts normal to the plane of the section.

The force f required to drive the flow of condensate circumferentially around the tube is substantially greater than the force e required to drive the condensate axially toward the discharge end. This is because the axial flow of condensate accumulated in the outermost sector of the tube occurs primarily over the porous layer. In contrast, circumferential flow of condensate must occur through the porous layer at a rapid rate despite the high flow resistance of the fine interconnected pores. Rapid, circumferential flow through the porous layer is required in order to avoid flooding the porous layer.

The involute tube should function satisfactorily regardless of direction of rotation. The Coriolis or "drag" force b normal to the radius of rotation is usually quite small relative to the centrifugal force a parallel to the radius. In other words, resultant c is actually almost coincident with vector a. It is usually preferable to rotate the tube in the direction of the expanding "sweep" of the involute, i.e., counterclockwise for the tube orientation shown in FIG. **9**a. This direction is preferred because the Coriolis force, even though small, tends to benefit performance, whereas for reversed rotation, the Coriolis force tends to lower performance. In the case of very large diameter disks, it may be advantageous to rotate the involute in the opposite direction, i.e., clockwise for FIG. **9**a, so that the Coriolis force aids in promoting flow of liquid axially along the tube.

While the involute tubes have been described as lying wholly in a plane normal to the axis of rotation, it would be feasible to shape the tubes so that the resulting disk is cupped or dished. For example, the tube of FIG. **9**a might possess an additional curvature out of the plane of the illustration. Such shapes are complicated and expensive to fabricate and in operation tend to produce uneven distribution of fluid on the external surface of the disks. However, the term "plane" as applied herein to the tube or disk-of-tubes is construed to include planes of either 2- or 3-dimension generated by the centerline of a tube as it revolves about its axis of rotation.

In order to evaluate the condensing heat transfer coefficient, tests were conducted on short sections of tube rotating at varying speeds and inclined about 5° from the normal to the gravitational field. The tests included both a smooth tube and tubes coated with porous metal on the inside surface. A condensible gas (high pressure nitrogen) was fed to the inner surface at measured rate and heat was removed from the outside surface by a boiling liquid (low pressure nitrogen). Results of tests on the porous coated tubes are given in Table I and are compared with the performance of the smooth, uncoated tube.

TABLE I

		Copper		Aluminum
Tube inside diameter—in.		0.250		0.305
Tube outside diameter—in.		0.375		0.375
Thickness of porous coating—mils		5-15		8-15
Coating particle size —mesh/inch		200-325		140-270
Porosity of Coating—%		50		60
Heat flux—Btu/hr×ft²	15,000	20,000	15,000	20,000
Condensation improvement factor for inside porous coated tube vs. smooth tube at:				
80 G	1.0	1.05	2.7	2.7
150 G	1.4	1.45	3.0	3.0
230 G	1.85	1.9	3.7	3.7
325 G	2.5	2.45	4.5	4.4
Evaluated condensing heat transfer coefficient—Btu/hr×ft²×°F at:				

80 G	1950	1840	5000	4600
150 G	3350	3200	6900	6300
230 G	5200	4750	9800	8900
325 G	8000	7000	13000	11900

For smooth tubes, the condensing coefficient was found to increase at higher values of G. At 325 G, smooth-tube condensing coefficients were observed up to 4.25 times the coefficient in normal one-G environment.

Table I shows that the porous coated condensing surface achieves still a further factor of improvement over the smooth tube which, for the aluminum tube is 4.5 at 325 G. Thus, the combined effect on the condensing coefficient of the high-G field and the porous coating is an improvement factor of 19 over a stationary smooth surface.

The coefficients obtained with the aluminum porous coated tube are higher than those obtained with the copper porous coated tube. This is attributed to the larger particle size and higher porosity of the aluminum porous coating. The more highly porous coating was better able to transport the circumferential flow of condensate within its thickness, thereby presenting more bare metal to the condensing vapor.

Based upon the evaluated condensing coefficients, a multiple-disk condenser-reboiler was designed and constructed using the involute tube array as heretofore described. The disks were 20 inches inside diameter and 48 inches outside diameter and each was composed of 183 aluminum tubes 0.250-inch I.D. × 0.16-inch wall thickness. 45 disks were provided in the assembly and were manifolded and coated inside and out with a porous aluminum layer as described previously. Gaseous nitrogen precooled to condensing temperature was fed to the inner tube manifold and liquid oxygen at its boiling point was distributed over the outer disk surfaces.

The reboiler-condenser was operated at varying rotational speeds and fluid pressures. Average gravity levels varied between 40 and 96 G, heat flux between 15,000 and 22,000 Btu/hr × ft², and overall heat transfer coefficients between 2000 and 3500 Btu/hr × ft² × °F. Data selected from the tests is shown in Table II.

TABLE II

Test No.	1	2	3	4
Heat exchanger disk I.D.—in.	20	20	20	20
Heat exchanger disk O.D.—in.	48	48	48	48
Rotary speed—rpm	350	350	360	350
Avg. gravity level—G	60	60	63	60
Nitrogen pressure—psia	167	199	200	200
Oxygen pressure—psia	45	45	55	55
Heat flux—Btu/hr×ft²	17,200	19,700	18,300	17,000
Avg. total ΔT—°F	6.7	8.5	6.8	5.7
Overall heat transfer coefficient Btu/hr×ft²×°F				
Experimental	2600	2300	2700	3000
Predicted	2080	1970	2120	2150

The boiling coefficients using the porous coated surfaces were known to be about 6000 to 7000 Btu/hr × ft² × °F so that the overall coefficients were controlled primarily by the condensing side. Thus, it was possible to separate the condensing coefficients from the overall coefficients with confidence and with low error. The coefficients thus separated are shown in Table II as "Experimental" and are compared with the coefficients "Predicted" from the foregoing short tube tests.

The preferred range of tube diameters for use in this invention is between 0.125 and 0.500 inch. Tubes smaller than 0.125 inch pose problems in applying the porous coating uniformly and without plugging. Moreover, the internal volume of the tube relative to its surface area diminishes to the point that a traffic problem is created for the condensate and results in the immersion of an excessively large sector of the tube wall in the flowing stream. With diameters larger than 0.500 inch, the surface area of the tube relative to its volume diminishes to the point that heat transport is seriously curtailed.

The preferred rotational speeds for this invention are those which produce force fields at the rotor periphery between 50 G and 400 G. Force fields less than 50 G cannot move the large flow of condensate rapidly enough through the porous layer to avoid flooding, with the result that performance is not significantly better than that achieved with smooth tubes. Force fields greater than 500 G create severe mechanical stresses in the rotor and porous layer. The preferred heat fluxes for this invention are between 5,000 and 40,000 Btu/hr × ft². In general, the need for the invention diminishes considerably below a heat flux of 5000 Btu/hr × ft², since lower cost stationary heat exchangers fitted with advanced boiling and condensing surfaces can approach this value without severe penalty in weight and space. Heat flux values above 40,000 Btu/hr × ft² tend to create fluid traffic problems within the exchanger. Moreover, if such high heat flux is the result of high ΔT generated by pressurizing the condensing fluid relative to the boiling fluid, then excessive mechanical stresses may occur. Excessive heat fluxes must also be avoided which approach the critical, "burn-out" value where boiling in the porous surface converts from the nucleate to the film mode.

The heat exchanger of this invention is useful for applications which require high capacity in very compact, low mass equipment. An illustrative process is air separation by distillation, wherein nitrogen is condensed as reflux against boiling oxygen. Another process is water purification by distillation wherein impure water is vaporized in heat exchange with the resultant vapor which is condensed at higher pressure. Yet another process is the vaporization of liquefied natural gas or methane by heat exchange with another fluid, the latter being condensed to recover the refrigeration of the vaporizing liquid.

Although certain embodiments have been described in detail, it will be appreciated that other embodiments are contemplated along with modifications of the disclosed features, as being within the scope of the invention.

What is claimed is:

1. A condensing heat transfer process comprising the steps of:

a. providing first fluid in a vapor form at near its condensing temperature,

b. flowing the first vapor fluid along an involute-shaped channel enclosed by a gas-impervious wall from an inlet end of greatest curvature toward a discharge end of least curvature in indirect heat exchange with a colder fluid,

c. rotating the involute-shaped channel with the end of greatest curvature nearer the center of rotation, thereby producing a high radial gravitational force on said first fluid, and

d. condensing first vapor fluid flowing along the involute-shaped channel on the surfaces of a porous heat conductive matrix layer interposed across the flow of heat from said first vapor fluid to said wall.

2. A condensing heat transfer process comprising the steps of:

a. providing first fluid in a vapor form at near its condensing temperature,

b. flowing the first vapor fluid along an involute-shaped channel enclosed by a gas-impervious wall from an inet end of greatest curvature toward a discharge end of least curvature in indirect heat exchange with a colder fluid,

c. rotating the involute-shaped channel with the end of greatest curvature nearer the center of rotation, thereby producing a high radial gravitational force on said first fluid having a component aligned parallel to the axis of the flow channel and a component aligned normal to the axis of the flow channel,

d. condensing first vapor fluid flowing along the involute-shaped channel on the surfaces of a porous heat conductive matrix layer interposed across the flow of heat from said first vapor fluid to said wall,

e. flowing liquid of said condensation into said porous matrix layer by capillary force,

f. flowing liquid of said condensation within the thickness of said porous matrix layer along a course generally parallel to the plane of said layer toward a liquid collecting zone within and along said flow channel, under the influence of said normal force component, at least a portion of the liquid collected in said zone being freed of said capillary force, and

g. flowing liquid of said condensation along said liquid collecting zone toward the end of said channel of least curvature remote from the center of rotation, under the influence of said parallel force component.

3. A process of heat transfer between a condensing vapor and a boiling liquid comprising the steps of:

a. providing a first vapor fluid at near its condensing temperature,

b. flowing the first vapor fluid along one gas-impervious wall enclosing an elongated involute-shaped channel from an inlet end of greatest curvature toward a discharge end of least curvature,

c. providing a second fluid in liquid form at a temperature near its boiling point and colder than the condensing temperature of said first fluid,

d. flowing the second, liquid fluid along the opposite side of said gas-impervious wall enclosing an involute-shaped channel from the end of greatest curvature toward a discharge end of least curvature,

e. rotating the involute-shaped channel with the end of greatest curvature nearest the center of rotation, thereby producing a high radial gravitational force on said fluids,

f. condensing first vapor fluid flowing along the involute-shaped channel on the surfaces of a first porous heat conductive matrix layer interposed between said first fluid and said wall, said layer containing pores of capillary size which are interconnected in all directions within the matrix layer, and

g. boiling the second, liquid fluid within the pores of a second porous heat conductive matrix layer interposed between said second fluid and said wall, said pores being of capillary size and being interconnected within the matrix.

4. A process of heat transfer between a condensing vapor and a boiling liquid comprising the steps of:

a. providing a first vapor fluid at near its condensing temperature,

b. introducing the first vapor fluid into an elongated involute-shaped channel enclosed by a gas-impervious wall having internal and external surfaces,

c. flowing the first vapor fluid along said internal surface and within the involute-shaped channel from an inlet end of greatest curvature toward a discharge end of least curvature,

d. rotating the involute-shaped channel with the end of greatest curvature nearest the center of rotation, thereby producing a high radial gravitational force on said first fluid having a component aligned parallel to the axis of the channel and a component aligned normal to the axis of the channel,

e. providing a second fluid in liquid form at a temperature near its boiling point and colder than the condensing temperature of said first fluid,

f. flowing the second, liquid fluid against the external surface of said involute-shaped channel from the end of greatest curvature toward a discharge end of least curvature under the influence of the high radial gravitational force,

g. condensing first vapor fluid flowing within the involute-shaped channel on the surfaces of a first porous heat conductive matrix layer interposed between said first fluid and said walls, said layer containing pores of capillary size which are interconnected in all directions within the matrix layer,

h. boiling the second, liquid fluid within the pores of a second porous heat conductive matrix layer interposed between said second fluid and said wall, said pores being of capillary size and being interconnected within the matrix,

i. flowing liquid of said condensation into said first porous matrix layer by capillary force,

j. flowing liquid of said condensation within the thickness of said first porous matrix layer along a course generally parallel to the plane of said first layer toward a liquid collecting zone within and along said channel under the influence of said normal force component, at least a portion of the liquid collected in said zone being freed of said capillary force, and

k. flowing liquid of said condensation along said liquid collecting zone toward the end of said channel of least curvature remote from the center of rotation under the influence of said parallel force component.

* * * * *

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,797,559 Issue Date March 19, 1974

Inventor(s) Richard S. Paul and David W. Weiler

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Col. 3, line 25: After "channel" insert -- and a component aligned normal to the axis of the flow channel --.

Col. 7, line 58: After "fine" insert -- zinc --.

Col.13, line 11: "inet" should read -- inlet --.

Signed and sealed this 30th day of July 1974.

(SEAL)
Attest:

McCOY M. GIBSON, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents