June 3, 1969 T. S. SPRAGUE 3,447,509

ONCE-THROUGH VAPOR GENERATOR

Original Filed Jan. 18, 1965

Fig. 1.

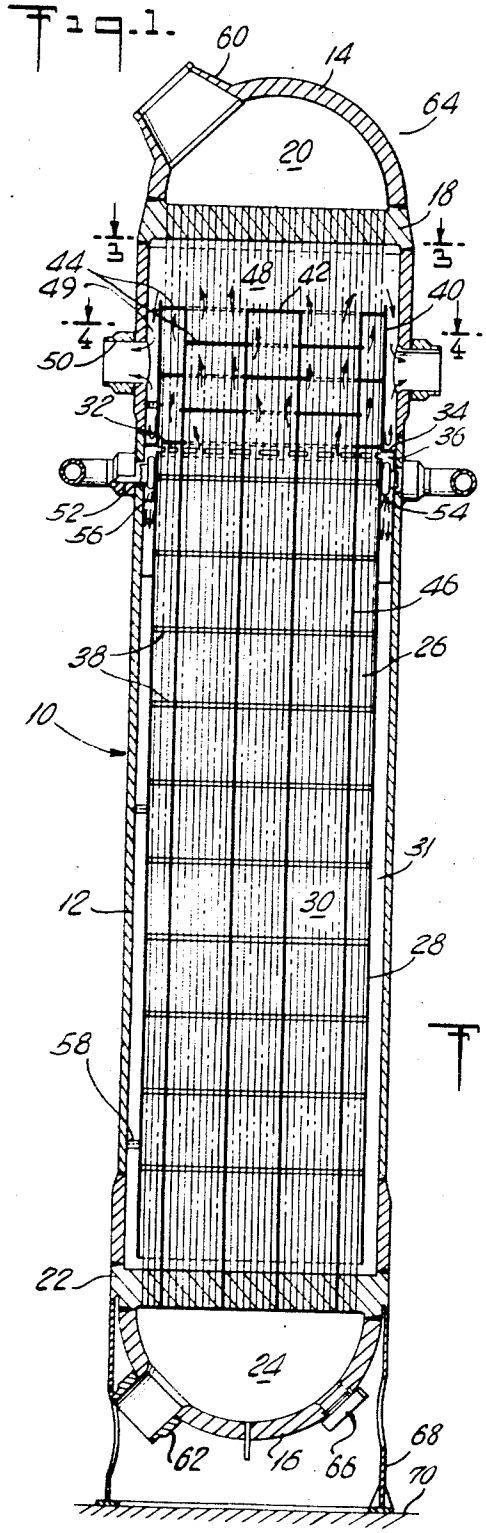



Fig. 2.

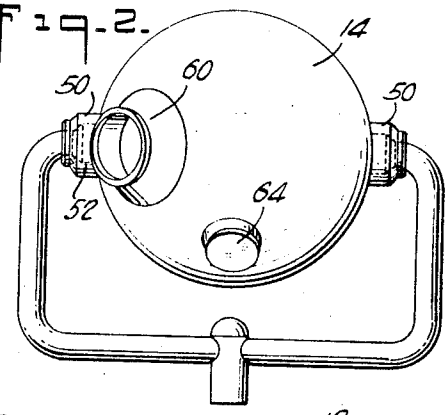



Fig. 3.

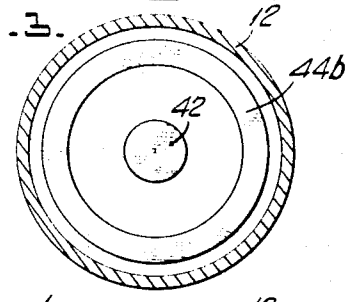



Fig. 4.

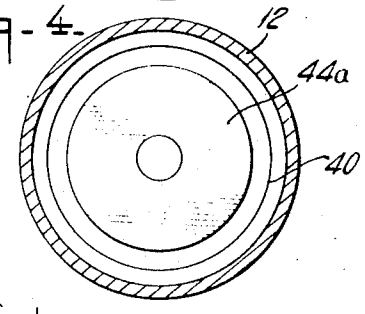
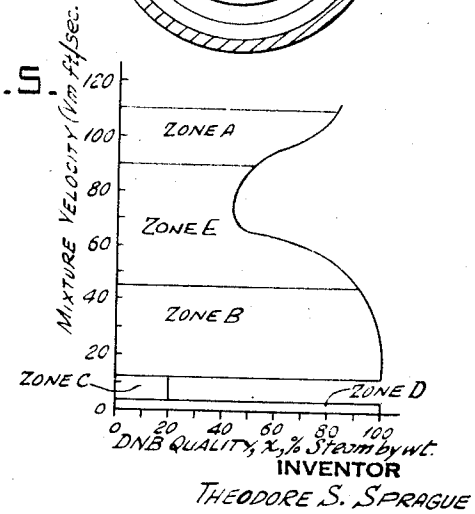



Fig. 5.




INVENTOR

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United States Patent Office

3,447,509

Patented June 3, 1969



3,447,509

# ONCE-THROUGH VAPOR GENERATOR

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Original application Jan. 18, 1965, Ser. No. 426,035, now Patent No. 3,385,268, dated May 28, 1968. Divided and this application Oct. 16, 1967, Ser. No. 675,538

Int. Cl. F22b *1/02, 9/02*

U.S. Cl. 122—32 6 Claims

## ABSTRACT OF THE DISCLOSURE

A heat exchanger comprising an upright cylindrical pressure shell divided by upper and lower horizontal tube sheets into primary fluid inflow and outflow chambers and a vapor generating and superheating chamber therebetween, with the latter chamber being occupied by an upright cylindrical shroud. The shroud encloses a bundle of tubes extending between and connected to the tube sheets, cooperates with the shell to form an annular fluid passage, and has its upper and lower ends respectively terminating adjacent the upper and lower tube sheets. Provisions are made for supplying heating fluid to the primary fluid inflow compartment for flow through the tubes to the primary outflow compartment; while vaporizable fluid is directed through the annular passage, then through the shroud and over the tubes in indirect heat absorbing relation with the primary fluid for conversion to vapor, and then to a point of use.

This is a division of application Ser. No. 426,035, filed Jan. 18, 1965, now Patent No. 3,385,268.

### *Background of the invention*

This invention relates in general to once-through vapor generators and superheaters and more particularly to such units operating at relatively low mass flows, that is, below 400,000 lbs./hr./ft.$^2$.

Mass flow is defined as the amount of fluid passing through a specific planar area in a given period of time. It was long believed that high rates of mass flow were needed in once-through vapor generators to increase the quality at which nucleate boiling breaks down and to improve the film heat transfer coefficient at and above the point of departure from nucleate boiling.

The point of departure from nucleate boiling, DNB, in a once-through vapor generator is particularly significant since it provides a sharp distinction between the high heat transfer rates usually associated with nucleate boiling conditions and the relatively very low heat transfer rates which distinguishes film boiling conditions. Nucleate boiling is characterized by the formation and release of vapor bubbles on the heat transmitting face of the heat transfer surface with the liquid still wetting the face, while in film boiling the heat transmitting face is coated with a film of vapor. To transfer heat from the transmitting face to the liquid, a temperature gradient is necessary. For a given set of operating conditions the magnitude of this gradient depends mainly on whether nucleate or film boiling is taking place.

In nucleate boiling the vapor bubbles generated at nucleation points or sites on the heat transfer surface rapidly detach themselves and move into the bulk liquid, the resulting agitation of the mixture producing an excellent heat transfer coefficient. In film boiling, a film of vapor forms over the heat transfer surface so that steam generation does not occur at the heat transfer surface but at the liquid-vapor interface. The vapor film prevents the liquid from wetting the surface and the resulting heat



transfer coefficients are poor. In effect the vapor film acts as a layer of insulation which retards the rate at which heat is transferred from the heat absorbing surface to the liquid. Therefore, the temperature of the heat absorbing surface is at a higher level than that resulting with nucleate boiling for the same mass flow conditions and the point at which nucleate boiling changes to film boiling is known as "departure from nucleate boiling" (DNB). It will be appreciated that while burnout will not occur in tubes when nucleate boiling prevails through the heat transfer domain there may be a serious problem of tube burnout in the film boiling range, depending on heat flux and mass flow within the heat transfer region.

To obtain maximum efficiency in a vapor generator, for example, a once-through type unit, it is important to maintain nucleate boiling water over as wide a range of steam qualities as possible. The optimum condition would be to maintain nucleate boiling during vapor generation from zero to 100 percent quality. However, achieving such range is not always possible. In the past it was the general belief that once-through boiler opeartion had to be carried out in a relatively high range of mass flow, that is above 600,000 lbs./hr./ft.$^2$. It had appeared that as mass flow dropped off the DNB quality limit would also continue to decrease. However, more recent research has indicated that a point exists where the DNB quality starts to increase with a further decrease in the mass flow. As a matter of fact, it appears that the DNB quality approaches 100 percent for extremely low values of mass flow.

Design of a once-through vapor generator is dependent upon a number of factors such as velocity, mass flow, heat flux, pressure and geometry or physical arrangement of the boiler. Previously, it was assumed that high flows were necessary for once-through vapor generators to maintain the necessary high DNB quality limit and also to provide the maximum cooling effect of the heat exchange surfaces in the film boiling range to avoid burnout. When it became apparent that once-through operation was possible in the relatively low ranges of mass flow it was recognized that such low mass flow conditions might advantageously be applied to once-through vapor generators utilized in combination with nuclear reactors, particularly of the pressurized water type. The primary coolant, outlet temperatures from such reactors are relatively low generally not exceeding 625° F. The coolant is pressurized to avoid boiling in the reactor and as a result it circulates from the reactor at high pressures, 2,000 to 2,5000 p.s.i., but as already stated at rather low temperatures.

### *Summary of the invention*

With the requirements of a heat exchanger for a nuclear reactor primary coolant system in mind, but without being limited to such a system, it is the primary object of the invention to provide a method of operating a once-through vapor generator using low mass flows and in the nucleate boiling domain.

Another object of the invention is to incorporate superheating with such a method of once-through vapor generation.

A further object of the invention is to incorporate a method of preheating the secondary feedfluid so that it will be at substantially saturation temperature corresponding to secondary vapor pressure operating at the commencement of its passage in heat transfer relationship with the primary heating fluid.

Still another object of the invention is to provide natural circulation induced flow of the vaporizable secondary fluid.

Yet another object of the invention is the utilization

of a portion of the vapor generated to heat the feedfluid substantially to saturation temperature.

Another main object of the invention is the provision of vertically disposed tubular heat exchanger arranged to carry out the above method of once-through vapor generation.

A further object of the invention is to provide means for heating feedfluid in the heat exchanger in an integral section, separated from the vapor generating space. Further, the heat exchanger is arranged so that vapor may be removed from the vapor generating space and introduced into the separate feedfluid inlet space for heating the feedfluid to saturation temperature.

Yet another object of the invention is to provide a special baffle construction for assuring that the vapor which is generated flows a tortuous path through the superheating space of the heat exchanger.

Accordingly, the invention comprises a method of vaporizing and superheating a fluid by circulating it through a heat exchanger in indirect heat transfer relationship with a heating fluid at mass flow rates below 400,000 lbs./hr./ft.$^2$. Prior to passing the vaporizable fluid through the heat exchanger, it is mixed with a portion of vaporized fluid which has absorbed heat, in the course of its passage about the tubes, to thereby heat the feedfluid to substantially saturation temperature.

Additionally, the invention comprises a vertically elongated heat exchanger having a bundle of heat exchanger tubes extending between a pair of spaced tube sheets. A cylindrically shaped shroud encloses the bundle of tubes for a substantial portion of its length, thereby providing a riser chamber about the tubes and an annular downcomer passage between the shroud and the heat exchanger shell. Openings are provided through the upper end of the shroud for admitting vapor, generated within the chamber, into the annular passageway wherein it mixes and heats the feedfluid by direct contact.

The various features of novelty which characterize the invention are pointed out with particularity in the claims annexed to and forming a part of this specification. For a better understanding of the invention, its operating advantages and specific objects attained by its use, reference should be had to the accompanying drawing and descriptive matter in which there is illustrated and described a preferred embodiment of the invention.

Of the drawings:

FIG. 1 is a vertical sectional view of a once-through vapor generator embodying the present invention;

FIG. 2 is a plan view of the vapor generator shown in FIG. 1;

FIG. 3 is a transverse section taken along line 3—3 in FIG. 1;

FIG. 4 is another transverse section taken along line 4—4 in FIG. 1; and

FIG. 5 is a plot illustrating a typical relationship of mixture velocity and DNB quality for a given vapor pressure, heat flux and geometry.

In FIG. 1 there is shown a once-through vapor generating and superheating unit **10** comprising a vertically elongated cylindrically shaped pressure shell **12** closed at its opposite ends by an upper head member **14** and a lower head member **16**. Within the unit a transversely arranged upper tube sheet **18** is integrally attached to the shell and upper head member **14** and forms in combination with the head member a fluid inlet chamber **20**. At the opposite end of the unit lower tube sheet **22**, integrally attached to the shell and lower head member, forms in combination with the head member a fluid outlet chamber **24**.

Vertically extending between the tube sheets **18** and **22** is a bank of straight tubes **26**. Disposed about the tubes is a cylindrically shaped lower shroud **28** which extends from a plane closely spaced above the lower tube sheet to another plane spaced a considerable distance below the upper tube sheet. This lower shroud forms a riser



chamber **30** including the bank of tubes and in combination with the shell **12** forms an annular shaped downcomer passageway **31**. Closely spaced below the upper end of the lower shroud **28** and passing through it are a number of openings **32** which afford communication between the riser chamber **30** and the annular downcomer passageway **31**. The upper end of the annular passageway is sealed by an annular plate **34** welded about its outer edge to the shell and around its inner edge to the shroud **28**. Within the riser chamber a number of lattice type tube supports **38** are spaced along the length of the bank of tubes **26**.

Extending upwardly from the lower shroud to a plane located below the upper tube sheet is cylindrically shaped upper shroud **40** which also encircles the bank of tubes **26**. Transversely positioned within this upper shroud **40** are a number of vertically spaced disk and doughnut flow directing baffles **42**, **44A** and **44B**, respectively, see FIGS. 1, 3 and 4, for imparting a tortuous flow path to the fluid flowing about the tubes. Support bars **46** anchored to and extending vertically upward from the lower tube sheet **22** through the tube bundle carry the disk and doughnut baffles. The space within the upper shroud **40** and extending upwardly to the lower face of the upper tube sheet provides a superheating chamber **48**. The space between the upper shroud and the shell forms an outlet passageway **49** from which vapor outlets **50** extend for delivering superheated vapor to a point of use.

At the upper end of the annular passageway **31** a pair of feedfluid inlets **52** extend through the shell **12** and are connected to a ring shaped feedfluid header **54** which circles the lower shroud **28** at its upper extremity. The heater has a plurality of small openings **56** in its lower surface. Between the shell and shroud at spaced locations are a number of transversely arranged centering pins **58** for properly locating the shroud.

In the upper head **14** a nozzle **60** provides a fluid inlet to the chamber **20**, see FIGS. 1 and 2, while in the lower head a nozzle **62** forms a fluid outlet from the chamber **24**. Further, in the upper and lower heads respectively manways **64**, **66** are provided for gaining admission to the chambers **20**, **24**.

At its lower end the vapor generator **10** is supported by a skirt **68** which extends downwardly from the lower end of the shell **12** to a support pad **70**.

During normal operation primary coolant received from a pressurized water reactor as similar heat source is supplied to the chamber **20** through nozzle **60**. From chamber **20** the primary coolant flows downwardly through tubes **26** and into the lower chamber **24** finally leaving the vapor generator through nozzle **62**.

The secondary coolant enters the heat exchanger through the inlets **52** flowing then into the feedfluid header **54**. From the header the secondary coolant or feedfluid at a temperature somewhat below its saturation temperature corresponding to the operating pressure of the secondary system is uniformly distributed into the upper end of the annular passageway **31**. Coincidentally vapor is drawn off from the riser chambers **30** through the openings **32** in the shroud **28**. At substantially 100 percent quality and upon passing through the space around the feedfluid header, it mixes with the atomized feedfluid. The vapor is condensed causing a slight reduction in pressure which provides an aspirating effect causing withdrawal of vapor from within the chamber **30** into the annular passageway **31**.

Within the passageway the vapor gives up its latent heat of vaporization to the feedfluid with the mixture being heated substantially to saturation temperature. From the annular passageway the secondary fluid passes into riser chamber **30** and since it is at substantially saturation temperature vapor generation immediately commences. The secondary coolant flows upwardly about the tubes and nucleate boiling is maintained as it passes in counter flow and indirect heat transfer relationship with the primary

heating liquid within the tubes. The low rate and temperature of the secondary coolant supplied to and flowing within the annular passageway is proportioned to assure natural or thermosyphonic circulation of the secondary coolant in its upward flow about the tubes **26**.

As the secondary coolant flows upwardly through the riser chamber **30** vapor is generated ranging from zero quality at the lower tube sheet **22** to substantially 100 percent quality adjacent the upper end of the lower shroud **28**. From this point that portion of the vapor which is to be superheated passes in a tortuous or sinuous path about the disk and doughnut baffles **42**, **44**A and **44**B and is thus superheated in chamber **48** before it reverses direction about the upper shroud **40**, flows downwardly into the outlet passageway **49** between the upper shroud and the shell and finally exits from the unit through the vapor outlet **50**.

As indicated, the primary coolant is preferably in a liquid state and typical operating parameters for both the primary and secondary coolants in such a unit are as follows:

TABLE 1

| | Primary | Secondary |
|---|---|---|
| Design pressure | 2,515 p.s.i.a. | 995 p.s.i.a. |
| Operating pressure | 2,215 p.s.i.a. | 825 p.s.i.a. |
| Coolant flow | 63,500,000 lb./hr. | 4,230,000 lb./hr. |
| Inlet temperature | 582° F. | 440° F. (feed). |
| Outlet temperature | 540° F. | 550° F. (33° F. S.H.). |

It will be noted that an extremely large quantity of heating liquid passes through the vapor generator and that while the temperature differential between the inlet and outlet is only 42 degrees there is a large reservoir of heat availability. In a typical unit such as the one shown in FIG. 1 there would be more than 15,000 ⅝" O.D. tubes in the tube bank.

In FIG. 5 the DNB curve illustrates qualitatively a typical relationship between DNB quality and mixture velocity. At the lower end of the mixture velocity scale between zero and approximately 3 feet per second is the zone D in which natural liberation and separation of steam from a body of water occurs in the course of vapor generating. This phenomenon is characterized as "pot type" boiling because of its similarity to boiling in a pot of water on a stove. As the pot is heated nucleation points develop and the steam bubbles break away from the surface of the pot to pass upwardly through the water, finally leaving the liquid. It will be noted that in this zone there is an upper DNB quality limit of 100 percent.

Again in FIG. 5 the area identified as zone C, extending roughly between velocities of from 3 to 12 ft./second and having a DNB quality limit of 20 percent is the normal design range of operation for natural circulation boilers. Natural circulation may be achieved by effective utilization of the force produced by differences in the density of the heated fluid mixture and the feedfluid supply. As heat is imparted to a fluid it becomes less dense than the fluid supply and it is this differential density phenomenon which, in properly proportioned downcomer and riser systems, promotes natural circulation of the fluid being heated.

A typical natural circulation boiler is disclosed in the Blaser et al. Patent No. 2,862,479 wherein the fluid to be heated flows first in an annular downcomer and then upwardly and is heated as it passes over the bank of shaped tubes. The weight of the column of vaporizable fluid in the downcomer passageway is greater than the weight of steam-water mixture within the riser passageway because the mixture, upon being heated, becomes less dense, thereby providing a fluid density differential which promotes natural circulation flow through the unit. However, from experience natural circulation boilers have generally been designed for a mixture quality maximum of 20 percent by weight leaving the vapor generating section. Since the design exit quality is thus far below the limiting DNB quality of 100 percent from FIG. 5, zone C, it is apparent that for normal operating conditions of such units nucleate boiling exists throughout the entire vapor generating flow path.

In order to provide exceptionally dry, saturated steam in units of the Blaser et al. type, the steam-water mixture must pass through mechanical separating equipment to separate the saturated steam from the water, the latter being then available for recirculation through the unit. When the mixture velocity through the boiler circuitry exceeds the upper limit defined by zone C as shown in the graph in FIG. 5 the resultant high circulation rate to produce so high a fluid pressure drop (resistant to flow) that natural circulation of fluid within the boiler unit is seriously impaired.

The upper portion of the plotted curve designated as zone A identifies the region of mixture velocity generally associated with forced flow, once-through boiler design. As previously indicated units of this type employ a high temperature heat source so that high velocities and correspondingly high mass flows over or through the tubular conduits are required to avoid burnout. As illustrated by the curve in FIG. 5 the limiting DNB quality in this domain reduces rapidly, viz from 80 percent to 50 percent quality, for a corresponding mixture velocity reduction from 110 feet per second to 90 feet per second. Heretofore, it has generally been presumed that this distinctive characteristic would persist as the mixture velocity was further reduced and it was concluded that once-through vapor generators could not be designed for satisfactory operation below this relatively high mixture velocity. However, subsequent research has demonstrated that the limiting DNB quality curve reverses itself with decreasing mixture velocities. In FIG. 5 it will be noted in zone E that at approximately 70 ft./sec. the curve starts to show increasing limiting values of DNB quality with decreasing velocity of mixture. The DNB quality rises quite rapidly from 45 percent to 85 percent for mixture velocities from 70 to 50 ft./sec. respectively. In zone B, on FIG. 5 between 12 and 45 ft./sec. the limiting DNB quality ranges from almost 100 percent down to 90 percent.

Besides mass flow and mixture velocity, however, the design of once-through boilers is dependent upon a number of other design criteria, such as heat flux, pressure and geometry of the flow channel. Thus, it is not possible to employ this lower region identified as zone B in FIG. 5 for the design of units having high temperature heat sources because of the incipient danger of the tube failure or high heat flux because of thermal fatigue in the zone of rapidly changing metal temperatures due to cyclical changes from nucleate to film boiling and back. As indicated earlier for the unit illustrated the operating parameters which characterize the primary coolant from a pressurized water reactor as a heat source make it especially adaptable for use in a once-through boiler operating in the low mixture velocity, high quality region. The temperature of the primary coolant is relatively low and at the same time large quantities of it are available from the reactor. By employing a comparatively low pressure secondary coolant system, with relatively low mass flows through the unit it is possible to achieve very high DNB quality provided the temperature of the heat source and heat flux are limited in accordance with the conditions previously discussed. Further, the change in temperature of the primary coolant as indicated previously in Table 1 is quite small, thereby limiting the deleterious effect of temperature differential within the unit as a result limiting thermal stresses particularly on the tube walls in the region where DNB occurs. However, the most important feature of this invention is the ability to maintain nucleate boiling throughout the entire range of vapor generation i.e. from zero to substantially 100 percent quality.

If during the operation of a once-through vapor generator film boiling occurs in a portion of the unit, the heat transfer from the heating fluid to the heated fluid

is substantially lessened and the absorption efficiency of the unit reduced. During nucleate boiling in the unit shown in FIG. 1, vapor bubbles from the secondary fluid form on the exterior of the tube walls, break away, and travel upwardly as an integral part of a vapor-liquid mixture. Moreover, since the secondary coolant or feedfluid is introduced at substantially saturation temperature, steam generation commences immediately. As the vapor-liquid mixture flows upwardly along and around each individual tube a boundary layer of liquid is maintained on the outside surface of the tubes to provide nucleation sites for the generation of steam bubbles. As the steam bubbles break away from the sites, they travel upwardly through the body of liquid at a velocity greater than that of the liquid. This phenomenon, known as "steam slip" characterizes vapor flow within a two phase fluid. Its effect is particularly significant at relatively low fluid velocities and operating pressures, and this is of especial importance as regards the steam generating system here disclosed. Since the vapor friction of the mixture is less dense than the liquid fraction, the effect of gravity on the more dense liquid will retard its upward velocity relative to that of the less dense vapor and thus will cause the vapor to travel upwardly faster than the liquid. This tendency of the vapor to flow at a higher relative velocity promotes the maintenance of the liquid film upon the tubes.

It is readily recognized that "steam slip" is a function of the pressure and corresponding relative specific weight or density of the vaporous and liquid constituents comprising the two phase fluid mixture. Thus as the pressure approaches the critical value, i.e. 3206 p.s.i.a., the differential density of the constituents is continually reducing so that at 3206 p.s.i.a. there is no differential and "steam slip" is non-existent. Obviously, the optimal relationship of these phase densities with respect to the corresponding liquid density and considering fluid flow friction characteristics, will produce the highest natural circulation head. Experience has indicated that this optimum condition generally prevails in the 800 to 1400 p.s.i. range and that natural circulation units operating within this range benefit substantially from "steam slip" both from optimization of heat transfer in the nucleate boiling domain and in economy of overall heating surface requirements for a specified performance condition.

By properly coordinating the sizing of the unit and its geometric arrangement and by maintaining the operating conditions within established design limits, it is possible to assure that the vaporizable fluid would be in a liquid state at substantially saturation temperature at its point of introduction into riser chamber **30** for flow about the tubes. In the heat exchanger construction shown in FIG. 1, the vaporizable fluid is transformed from a liquid at saturation temperature into a vaporous state in the course of its upward flow from the lower end of tube bank **26** to the upper portion of the tubes adjacent the upper end of the lower shroud **28**. Thus, at the upper end of the shroud the vaporizable fluid will exist as vapor at substantially 100 percent quality.

From the upper end of the lower shroud a certain amount of vaporized fluid flows from the chamber into the passageway due to the aspirating effect produced as feedfluid spray collapses the vapor. The vaporizable feedfluid is sprayed into the passageway at a subcooled temperature and condenses the vapor withdrawn from the chamber. This condensation results in a sudden decrease in pressure at the top of the passageway **31** and as a consequence vapor is continuously withdrawn from within the chamber **30**.

By establishing the amount and temperature of the vaporizable feedfluid supplied to the unit through the header **54** and by properly selecting the size of the openings **32** in the lower shroud **28** it is possible to admit sufficient vapor into the passageway so that the feedfluid can be heated to substantially saturation temperature within the passageway **31**. Accordingly, the rate of feedfluid supply to the unit will be equivalent to the amount of superheated vapor taken from the outlet **50**, while the total flow rate in the downcomer passageway **31** will be the sum of the feedfluid and the saturated steam withdrawn from the chamber **30** through the openings **32**.

Earlier this unit has been designated as a once-through unit utilizing natural circulation to effect fluid flow therethrough. It will also be observed that there is some degree of recirculation since a portion of the vapor generated is utilized to attain the saturation temperature of the feedfluid, which contributes to the uniqueness of this arrangement and the distinctive method of operation. Ordinarily, with natural circulation it is a liquid fraction of the heated fluid which is recirculated. In the present invention a portion of the vapor is withdrawn for recirculation first in vaporous state and then as a liquid. In the Blaser et al. patent previously mentioned, especial care is taken to prevent the admission of any vapor to the down flowing separated coolant and this is common of all such units, however, in the present invention the complete opposite is desired, i.e., the vapor is purposely mixed with the feedfluid to heat it to saturation temperature to serve as supply for the steam generating portion of the unit. If the output demand on the steam generator changes, the temperature and quantity of the feedfluid may be varied to maintain the desired operating characteristics.

Accordingly, by maintaining the proper limits of temperature, velocity, heat flux, pressure and geometry, it is possible to operate a once-through vapor generator and superheater in the mass flow range which had not previously been considered possible.

While in accordance with the provisions of the statutes, I have illustrated and described herein a specific form of the invention now known to me, those skilled in the art will understand that changes may be made in the form of the apparatus disclosed without departing from the spirit of the invention covered by my claims, and that certain features of the invention may sometimes be used to advantage without a corresponding use of the other features.

What is claimed is:

**1.** A heat exchanger comprising

an upright cylindrical pressure vessel,

upper and lower tube sheets dividing the vessel into fluid inflow and outflow chambers and a tube bank chamber therebetween,

a bank of straight vertical tubes in the tube bank chamber connected to the tube sheets,

an upright shroud in the tube bank chamber surrounding the tubes, cooperating with the vessel to provide a fluid flow passage therebetween, and having its upper and lower ends respectively terminating adjacent the upper and lower tube sheets,

means dividing the flow passage into inlet and outlet compartments,

means for directing a heating fluid to the inflow chamber through the tubes to the outflow chamber,

means for directing a heat absorbing fluid to and through the inlet compartment, through the shroud and over and along the tubes in indirect heat exchange relation with the heating fluid, to the outlet compartment, and

means for withdrawing a portion of the heat absorbing fluid at a point in the course of its passage along the tubes and directing it in mixing relation with the heat absorbing fluid entering the inlet compartment.

**2.** A heat exchanger according to claim **1**, wherein said last named means comprises a series of closely spaced ports formed in the shroud and opening to the inlet compartment.

**3.** A heat exchanger according to claim **1**, wherein the tube bank chamber comprises a vapor generating section and a vapor superheating section, and horizontal baffles are provided within the shroud and outside the tubes in

the vapor generating section and are arranged to direct the heat absorbing fluid through a tortuous path over and along the tube portions therein.

4. A heat exchanger according to claim 1, wherein the inflow and outflow chambers are respectively at the upper and lower ends of the chamber, the inlet compartment opens to the lower end of the shroud, and the outlet compartment opens to the upper end of the shroud.

5. A heat exchanger according to claim 1, wherein the vessel is formed with an outlet nozzle opening to the outlet compartment.



6. A heat exchanger according to claim 1, wherein the shroud is cylindrical in form.

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KENNETH W. SPRAGUE, *Primary Examiner.*