FORCED FLOW VAPOR GENERATOR

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The present invention relates in general to the construction and operation of a forced flow fluid heating unit and more particularly to improvements in the construction and arrangement of fluid heating circuits especially adapted for use in a forced circulation once-through vapor generating and superheating unit.

The invention herein disclosed is an improvement over the vapor generating unit described in U.S. Patent No. 3,125,993, issued March 24, 1964 in the name of P. H. Fox. The general object of the present invention is the provision of a fluid heating unit of the character described so constructed and arranged as to produce superheated vapor from a vaporizable fluid over a desired range of high pressures and temperatures; to assure an optimum quantitative and qualitative distribution of fluid to all fluid flow paths; to assure an optimum relation of fluid velocity within the tubes to heat input into the tube walls to effect adequate cooling, thereby maintaining the tube walls at a safe operating temperature.

The construction of forced circulation once-through steam generators requires the use of a large number of parallel circuits connected between inlet and outlet headers. One of the fundamental problems involved with such a steam generator is the control of the flow through the various parallel circuits in order that the flow in each circuit will be stable and the enthalpy of the fluid discharged from any individual circuit will be close to the average of that from all circuits, in which case the circuits will be in a balanced flow condition. Unbalanced flow may be caused by unequal heat absorption in parallel circuits due to unsymmetrical arrangement of heating surfaces, slag accumulation, or part-load operation. The more severe causes of unbalance may be due to unequal resistances caused by different lengths of circuits. When steam or water, or mixtures thereof, is heated in parallel flow paths provided by the furnace wall tubes or tubular panels disposed within the furnace, unbalanced heat and/or fluid flow distribution may lead to excessive localized tube metal temperatures and/or to excessive temperature differentials between adjacent furnace wall tubes, resulting in excessive thermal stresses in the furnace wall-forming components. The problem of unequal fluid flow distribution is accentuated when the fluid supplied to the high heat absorbing parallel flow circuits of the furnace is a mixture of steam and water. Whenever a steam-water mixture must be distributed to many tubes of parallel circuits, the possibility of separation of the steam-water mixture exists. Thus, one tube may receive saturated steam and another tube may receive saturated water or any combination of the two components.

Such a condition implies a limit on the rate of heat absorption that the tubes of the parallel flow circuits can tolerate without exceeding allowable metal temperatures or allowable temperature differences between adjacent tubes.

The fluid distribution system described in the above identified U.S. patent has proven to be successful for its intended purpose of uniformly distributing circulating fluid both quantitatively and qualitatively, to a plurality of parallel-flow fluid heating circuits. However, it has been found that, due to uneven heat absorption in the individual fluid heating circuits, uniform quantitative distribution of fluid to the circuits tends to produce exorbitant localized tube metal temperatures and exhorbitant temperature differentials between adjacent furnace wall tubes under certain operating conditions. For example, it has been found that the heat absorption in the circuit having the highest heat absorption rate is about fifty percent higher than in the circuit having the lowest heat absorption rate when both circuits have equal amounts of fluid flowing therethrough. To remedy this situation and to effect balanced conditions whereby the enthalpy of the fluid discharged from any individual circuit will be close to the average of that from all circuits, it has become necessary to provide flow restrictors in the individual circuits so that the flow through each may be adjusted.

It should be recognized that the adjustment of flow in this instance refers only to the qualitative and not the quantitative characteristics of the flow, i.e., it is still desirable that the enthalpy or quality of the fluid flowing into all the circuits be uniform.

To obtain the necessary flow adjustment, a variable orifice valve has been installed in each circuit and set to effect the necessary pressure drop. A pressure drop of about 150 p.s.i. was found to be required across the valve in the circuit requiring maximum restriction, i.e., the circuit having the lowest heat absorption rate. Over a short period of operation with the valves in service, it was noted that the flow in the circuits was gradually changing, and that temperature limitations in some tubes were being exceeded. It was also noted that the changes in flow were most notable in those circuits having the greatest restriction, and that the flow in the unrestricted circuits was relatively unchanged. Inspection of the valves revealed a deposition on the seats and discs of the variable orifice valves projecting into the flow stream and restricting flow therethrough.

It was known from experience that plate-type or sharp edge orifices, when used in boiler fluid flow circuits, tend to accumulate deposits on the upstream side of the orifice. In time, these deposits extend over the orifice edge and restrict the free flow area. Analysis of these deposits indicates that they are composed primarily of metal oxides and that they are of such a nature as to be extremely resistant to mechanical or chemical removal efforts. It has also been found that the amount of deposit was generally proportional to the pressure drop through the orifice and the abruptness with which the pressure drop is experienced. Those restrictors having the greatest pressure drop and the most abrupt pressure change were found to be the most susceptible to the formation of the above described depositions. These phenomena previously experienced with depositions on plate-type orifices were obviously the cause of the deposition problem in the variable orifice valves.

So a further and more specific object of this invention is to provide a device and system for selectively restricting the flow in the individual furnace wall heating circuits so that balanced enthalpy conditions may be obtained at the discharge ends thereof. This aspect of the invention is embodied in a forced circulation fluid heating unit having a furnace chamber supplied with high temperature heating gases, with each wall of the furnace including a row of upwardly extending contiguous upflow tubes arranged for parallel flow of fluid therethrough relative to each other and supplied at their lower inlet ends with a vaporizable fluid from a common source. These tubes are grouped into a plurality of parallel flow fluid heating circuits, some of which are arranged to absorb less heat than others. Provisions are made for distributing the fluid from the common source into a plurality of subcircuits or conduits for flow into the upflow tubes of each circuit. The conduits serving those circuits having rela-
tively low heat absorption rates are provided with flow inhibitors to compensate for the inequality of the heat absorption in the circuits and thereby promote substantially equal enthalpy conditions at the outlet ends of all the upflow tubes. Each flow inhibitor includes a venturi-shaped flow restrictor constructed to avoid abrupt changes and to effect gradual reduction of pressure.

The various features of novelty which characterize my 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 drawings and descriptive matter in which I have illustrated and described a preferred embodiment of the invention.

Of the drawings:

FIG. 1 is a partially diagrammatic sectional elevation of a forced circulation once-through steam generating unit constructed and operable in accordance with the invention;

FIG. 2 is an enlarged representation of the lower portion of the furnace of FIG. 1 taken along line 2--2 of FIG. 4;

FIG. 3 is a fragmentary front view taken along line 3--3 of FIG. 4;

FIG. 4 is a fragmentary plan section taken along line 4--4 of FIG. 2;

FIG. 5 is a sectional view of a preferred type of flow restrictor element;

FIG. 6 is a plan section taken along line 6--6 of FIG. 1;

FIG. 7 is a plan section taken along line 7--7 of FIG. 1; and

FIG. 8 is a plan section taken along line 8--8 of FIG. 1.

In the drawings the invention has been illustrated as embodied in a top-supported forced flow once-through vapor generating and superheating unit designed for the production of superheated steam at pressures below the critical pressure of 3206 p.s.i. The unit herein shown is of the same general construction and arrangement as the unit disclosed and described in detail in the above-mentioned U.S. Patent No. 3,125,995. The details of the unit will therefore be described herein only so far as is necessary to recognize the import and applicability of the invention, and reference should be made to the aforesaid U.S. patent for further detail.

The description of the invention in terms of this particular unit is not meant to limit the invention and it should be recognized that the invention may also be used in conjunction with any type of forced flow once-through vapor generator operating either above or below the critical pressure.

The main portions of the unit illustrated include an upright furnace chamber 10 of substantially rectangular horizontal cross-section defined by a front wall 12, a rear wall 14, side walls 16 and a roof 18 and having a gas outlet 20 at its upper end opening to a horizontally extending gas pass 22 of rectangular vertical cross-section formed by a floor 23 and extensions of the furnace roof 18 and side walls 16. The gas pass 22 communicates at its rear end with the upper end of an upright gas passage 24 of rectangular horizontal cross-section formed by a front wall 26, a rear wall 28, side walls 30 and a roof 32. The lower portions of the front and rear walls of the furnace slope inwardly downwardly and cooperate with the furnace side walls to form a hopper 40 and a rectangular throat passage 42 for discharging ash into an ash pit, not shown.

A secondary superheater 44 is disposed in part in the upper portion of the furnace 10 adjacent the gas outlet 20 thereof, with the remainder occupying the furnace end of the gas pass 22. An intermediate primary superheater section 60 and a secondary reheater (not shown) are disposed in the end of the gas pass 22 nearest the entrance to the upright gas pass 24. The upright gas pass 24 is occupied by a primary superheater 59 and a primary reheater (not shown), each of which comprises horizontally extending nested multiple-looped tubes arranged in laterally spaced panels. The lower end of the upright gas passages 24 is occupied by a horizontally arranged multiple-looped return bend tubular economizer 66 disposed downstream gas-wise of the primary superheater 59 and the reheater. It should be recognized that the above described superheater and reheater sections are interconnected by appropriate piping (not specifically identified) and that vapor flows serially through these sections in accordance with the description in the aforesaid U.S. Patent No. 3,125,995.

Combustion air and fuel are supplied to the lower portion of the furnace. The resulting heating gases flow from the furnace chamber 10 over the radiant heat absorbing portion of the secondary superheater 44, and then through the gas outlet 20 and over the convection heat absorbing portion of the secondary superheater 44 in the gas pass 22. Continuing through the gas pass 22, the heating gases flow over the superheater 60 and the secondary reheater (not shown) and enter the upright gas passage 24 at its upper end and flow downwardly therethrough over the primary superheater 59 and primary reheater and thence through the economizer 66 to outlet 20 where the gases are discharged into an air heater (not shown).

The furnace chamber 10 is fired by horizontally extending burners 68 arranged to direct fuel and air in mixing relationship into the chamber through corresponding burner ports in the boundary walls of the furnace. Preheated air is supplied to the burners by a forced draft fan (not shown) which discharges air under pressure through the air heater and a duct 70 to a vertically extending windbox 75 enclosing the burners 68 and the lower portion of the boundary walls of the furnace 10. The front, rear and side walls of the furnace 10 and the side walls of the gas pass 22 are formed in most part by Insulation covered fluid heating tubes. In accordance with the present invention, the furnace boundary wall fluid heating surface is so arranged that the distribution of flow to all fluid flow paths is substantially proportional to the heat absorption rates of the particular fluid flow paths; that the maximum temperature differential between the adjacent tubes is below a predetermined critical limit, thereby maintaining differential expansion in the walls within safe limits; that the tube surfaces in different portions of the furnace are sufficient in quantity to carry away the heat at a rate adequate to prevent overheating of the tubes; and that the tubes are of sufficient inside diameter along their lengths to provide adequate fluid circulation velocities. Accordingly, each of the boundary walls of the furnace 10 is lined by a row of upwardly extending parallel tubes arranged in groups to form coplanar laterally contiguous radiant heat absorbing tubular panels, or circuits, the front wall 12 having a row of tubes 80, the rear wall 14 including a row of tubes 87, and each side wall having a row of tubes 80. The rows of tubes 80 are disposed in the end of the side walls of the gas pass 22 opposite the secondary superheater 44. Some of the rear wall tubes 87 have their upper portions bent inwardly and upwardly to form a nose arch 96; then rearwardly and upwardly to form the floor 23 of the gas pass 22; and then vertically to form part of a screen 98 disposed at the rear end of the gas pass 22. Intermediate portions of some of the tubes of the front, rear and side walls in the furnace are suitably bent to form the openings or ports for the burners 68. As shown in FIGS. 1 and 7, the front wall 12 includes a mating portion 100 of a front pass flow tube 80 disposed in parallel spaced relation and for parallel flow of fluid therethrough, spaced about a tube diameter apart to provide intertube spaces, and extending throughout the high heat intensity burner zone of the furnace from
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the level of the top of the hopper to a point about half way between the top row of burners and the nose arch 96. The tubes 80A have their opposite ends connected to horizontal outlet and inlet headers 82 and 84, respectively, disposed outside of the wall 12, while the corresponding tubes 80B, disposed outside of the wall 24, have their lower and upper ends connected to horizontal outlet and inlet headers 85 and 86, respectively. Since the tube lengths of the burner zone are exposed to gases of higher heat intensity than those above and below this zone, their total absorption is higher and the quantity of heating surface presented by the tube lengths in the burner zone must be greater than that and below the burner zone to carry away the heat at a rate sufficient to prevent overheating of the tubes. Accordingly, the front end of the furnace as well as the other furnace boundary walls, also includes a multiplicity of second upflow tubes, designated as tubes 80B for the front wall, extending throughout the height of the furnace, arranged in parallel spaced relation and for parallel flow of fluid through them, and disposed in the spaces between and contiguous to the initial upflow tubes, tubes 80A in the case of the front wall, along the height thereof, so that the number of tubes presented to the gases in the zone of high heat intensity is double that above and below this zone, as shown in FIGS. 6 and 8. Thus the tubes 80B are spaced about a tube diameter apart and connect the tubes 80A through the height of the burner zone. The same relation applies to the corresponding tubes of the other upright furnace boundary walls. The tubes 80B have their upper and lower ends connected respectively to horizontal outlet and inlet headers 85 and 83 situated outside of the furnace, while the corresponding tubes 90B of each side wall 16, as shown in FIGS. 1 and 2, have their opposite ends connected to horizontal outlet and inlet headers 95 and 93, and the corresponding tubes of the rear wall 14 have their lower and upper ends connected to horizontal headers 81 and 89, respectively. The furnace boundary walls fluid heating tube portions in the zone of high heat intensity are covered by metallic casing suitably secured thereto, while the furnace boundary wall tube portions above and below the zone of high heat intensity, these being second upflow tube portions, have their tube spaces closed by metallic webs 79, rigidly secured to the tubes, as shown in FIGS. 6 and 8.

For the unit shown, feedwater at a pressure of 3070 p.s.i.g. is supplied by a feed pump (not shown), to the economizer 66 wherein it is partially heated, then flows through the economizer 66 and outlet headers 84, 88 and 94, supplying the alternate initial upflow tubes of the radiant heat absorbing fluid heating tubes lining the front, rear and side walls of the furnace chamber. The water flowing to these headers is in a sub-cooled condition, i.e., at a temperature below the saturation temperature corresponding to the pressure. Having passed through the initial upflow tubes, the fluid is collected in the outlet headers 82, 86 and 92 and passes through tube inlets connected to the header 112 disposed along the perimeter of the gas pass 24 at a position intermediate the economizer 66 and the primary superheater 58. The peripheral header 112 is arranged to supply fluid in parallel flow paths to the tubes forming the baffle walls (not shown) of the gas pass 22 and the upright gas passageway 24 and to the tubes lining the side walls of the gas pass 22 and 24. The fluid flowing through these baffles and boundary walls, is collected, through a system of headers (not specifically identified) at the top of the unit, in the transverse horizontal header 113 above the front wall 12. While the fluid heating surface of the unit are proportioned and arranged so that at full load and designed operating pressure the portion of the heated fluid circuit in which the transition of the fluid from a water condition to a steam-water condition will be located in the second upflow tubes of the furnace, it is expected that under certain load and pressure conditions the transition will take place before the fluid reaches the second upflow tubes. Thus the fluid supply system for the second upflow tubes of the furnace is so constructed and arranged as to inhibit or prevent separation of the steam from the water when the fluid consists of a mixture of both. In addition, the system must promote mixing of the fluid streams whether they be in a water or a steam-water condition, as they pass from the initial upflow tubes to the second upflow tubes of the furnace and by providing a substantially uniform fluid enthalpy upon discharge to the second upflow tubes. To compensate for the inequality of heat absorption in the various portions of the water circuits of the furnace wall, and to promote substantially equal enthalpy conditions at the outlet ends of the second upflow tubes, it is also required that the fluid supplied to the second upflow tubes be so proportioned that those tubes or flow circuits having a relatively low heat absorption rate are supplied with less fluid than those tubes or flow circuits having higher heat absorption rates.

Accordingly, the fluids collected in the header 113 are passed without further heating downwardly through the upright downcomer 105 extending along and outside of the front wall 12 to the spherical fluid mixing and distributing vessel 107. This vessel 107, as described in detail in said U.S. Patent No. 3,975,955, tends to qualitatively and quantitatively distribute the fluid to the conduits or pipes 109 which supply fluid to the headers 83, 81 and 95 serving the second upflow tubes of the furnace chamber walls. The fluid supply system for each of the fluid supply headers 83, 81 and 95 comprises one or more supply tubes 109 each opening at one end to the vessel 107 and arranged to discharge fluid in parallel flow relation to a group of tubes 115 having their outlet ends connected to the corresponding fluid supply header at uniformly spaced positions along the length thereof. The second upflow tubes 109 having its upper end closed by a corresponding nipple 117. Tubes 115 of each group lead radially from the corresponding nipple in a common horizontal plane, then extend vertically for distribution of fluid to the tubes 115, with the vertical portion of each tube 109 having its upper end closed by a nipple 117. Tubes 115 of each group lead radially from a corresponding nipple in a common horizontal plane, then extend vertically for connection to one of the fluid supply headers. Outlet headers 93, 85 and 89 of the second upflow tubes of the furnace side, front and rear walls are connected by pipes (not shown) for series flow of the vapor-liquid mixtures generated in the second upflow tubes to a collecting header 121 from which the fluids pass through a conduit 123 to the primary superheater 58.

The particular form of the vessel 107 and the routing and arrangement of the conduits leading to and from the vessel assure that a completely homogeneous fluid of uniform enthalpy will be discharged to the second upflow tubes because the fluid streams are intimately mixed as they pass from the initial to the second upflow tubes of the furnace and because separation of steam from water is inhibited or prevented whenever the fluid is so constituted. Assuming that the back pressure is equal in all of the circuits served by pipes 109, the distributor vessel 107 would effect uniform quantitative as well as qualitative (equal enthalpy) distribution among the supply pipes 109. However, as discussed above, some of the portions of the furnace wall may be subject to more radiant heat than others so that some of the flow circuits supplied through the inlet headers 81, 83 and 95 may have greater heat absorption capacities than others. In order that uniform enthalpy conditions may be maintained at the outlet ends of the second upflow tubes, and
in order to avoid exceeding the tube metal temperatures in the tubes of the flow circuits exposed to the greatest amount of radiant heat, it is desirable to proportion the flow of fluid to the various furnace wall circuits so that the fluid to a given circuit is substantially directly proportional to the heat input to that circuit.

By way of example, and not of limitation, the downcomer 105 at 9° I.D. and has a vertical height of about 114 ft.; the vessel 107 is 30° I.D. and supplies fluid to thirty-six of the tubes 109, with each tube 109 in turn supplying fluid to four of the tubes 115 which distribute fluid to the seven hundred and thirty-eight second upflow tubes of the furnace by way of the inlet headers thereof. In the preferred construction and arrangement of the fluid mixing and apparatus, tubes 109 should have at least the equivalent of ten diameters of vertical straight length ahead of the nipples 117 and should have a cross-sectional area sufficient to provide a minimum fluid velocity of 10 ft. per second. For obvious reasons of economy and simplicity, the proportioning of fluid to the various portions of the furnace wall should be effected in the supply conduits or pipes 109, each of which serves a fluid circuit consisting of a panel of approximately thirty second upflow wall tubes.

The present invention is primarily concerned with the provision of flow inhibiting means in the circuit supply pipes 109. The use of the flow inhibiting means to be described subsequently as applied to the boiler tube, for the vapor generator is not meant to limit the use of the flow inhibitors to this particular type of unit, but rather it should be understood that the invention concept is applicable to any type of forced flow vapor generating unit, including those of the type operating at pressures above the critical pressure.

By way of illustration, and not of limitation, in the unit herein described, the estimated required flow through the circuit requiring the lowest flow is 34,000 lb./hr., while the corresponding flow through the circuit requiring the highest flow is 54,000 lb./hr. The lowest flow circuit and the highest flow circuit are shown in FIG. 4 respectively identified by brackets and marked L and H, and the supply pipes 109 leading to them are identified as 109L and 109H. In this illustration the estimated required flow through the remaining thirty-four circuits is somewhere between 34,000 and 54,000 lb./hr. To obtain the necessary restriction of flow in the circuit L, a flow restrictor tube 200L is provided in the supply conduit 109L.

The preferred type of flow restrictor tube 200 is shown in FIG. 5 as being welded in flowing communicating relationship in a supply pipe 109. The flow restrictor tube 200 is constructed in the form of a venturi having a conical converging inlet portion 201, an intermediate constricted throat portion 202 and a conical diverging outlet portion 203. The internal conical surface of the inlet portion 201 should preferably have an included angle of less than thirty degrees, and the corresponding surface in the outlet portion 203 should preferably have an included angle of less than fifteen degrees, as is the practice in the application of venturi flow meters. The internal diameter and the length of the throat portion 202 are to be sized to provide the pressure drop required to obtain the desired flow inhibition; however, the internal diameter of the throat portion 202 should preferably not be smaller than 7/8 of the internal diameter of the supply pipe 109 in which it is interposed.

It is shown that the solubility of solids in fluids such as boiler water is a function of pressure, and that a change in pressure tends to facilitate precipitation of the solids from solution. There is no known quantitative data representing these phenomena. However, experience has indicated that the amount of deposition is proportional to the rate of change of pressure drop. Thus a flow restrictor tube of the type described will tend to reduce the deposition of solids precipitated from the heating fluid because of the gradual changes and reduction in pressure and because of the elimination of eddy currents. Furthermore, any precipitation which may still occur will be deposited over a larger area and will therefore not have a material effect on fluid flow. It should be observed that a portion of the pressure drop is obtained is caused by the known phenomena of incomplete pressure regain commensurate with the temporary constriction of a stream of flowing fluid, while an additional portion of the pressure drop is obtained by virtue of the increased friction between the high velocity fluid stream and the inner wall of the throat portion 202.

Although the description has been in terms of a particular type of flow restrictor, it should be recognized that the present invention is not so limited, and that it is intended that the invention cover the restriction of flow by any venturi shaped flow restrictor, that term being herein used as applying to any of the class of tubes having flaring ends connected by a constructed middle section forming a throat. Again by way of illustration and not limitation, in the unit disclosed the length of the throat portion 202 in the flow restrictor tube 200L having the longest throat is 76.5 inches while the internal throat diameter is 0.637 inch. Thus, it can be seen in this instance that the major portion of the pressure drop is due to friction of the fluid flowing through the throat. The total pressure drop imposed by the restrictor as applied to the above description is for purposes, the flow restrictor tube affording the least amount of flow restriction has a throat diameter of 0.810 inch and a throat length of 4 inches, resulting in a pressure drop of 10 p.s.i. Thus it can be seen that great latitude is available in the design of flow restrictor tubes of the type disclosed. For purposes of simplicity it is preferred to limit the number of different throat diameters used on a single installation to two or three, and to obtain the desired pressure drop by varying the throat length. It should be also noted in FIG. 4 that some of the supply pipes 109 are not provided with flow restrictor tubes. These supply pipes serve the higher heat absorption flow circuits where none or only a small flow restriction is required.

In conjunction with the use of variable orifice type valves as discussed above, it has been found that no deposition problems are experienced so long as the pressure drop across the valve is maintained below about 30 p.s.i. Therefore, in order to compensate for any errors in estimating the flow through the various circuits, variable orifice valves 205 have been installed in those flow circuits requiring the largest flow restriction. For example, a valve 205L is disposed upstream of the flow restrictor tube 200L in the supply pipe 109L. In the event that the estimated flow proves to be incorrect, or in the event, for example, that slag accumulations in some portion of the furnace cause a radical change in the fluid flow requirements of any circuit, the flow may be adjusted accordingly by means of the valves 205.

Thus in practice the venturi-shaped flow restrictor tubes 200, sized according to calculated expected flow requirements in the various circuits, are provided with some of the supply pipes 109 and provide the major flow inhibiting means for compensating for the inequality of heat absorption in the fluid heating circuits. In addition, the variable orifice valves 205 are installed in some of the supply pipes 109, in series with the flow restrictor tubes 200 to afford fine adjustment so that errors in estimated flow may be compensated for as well as to allow adjustments required as a result of a change in conditions within the furnace.

While in accordance with the provisions of the statutes there are illustrated and described herein a specific embodiment of the invention, those skilled in the art will understand that changes may be made in the form of the invention covered by the claims, and that certain
features of the invention may sometimes be used to advantage without a corresponding use of the other features.

The claims are:

1. In a forced circulation fluid heating unit, walls defining a furnace chamber, burner means supplying high temperature heating gases to said chamber for flow therethrough, at least one of said walls including a row of upwardly extending contiguous upflow tubes arranged for parallel flow of fluid therethrough and grouped into a plurality of parallel flow circuits some of which are arranged to absorb less heat than others, a fluid distributor vessel, means for supplying a vaporizable fluid to said vessel, a plurality of conduits connecting said vessel for parallel flow of fluid to the inlet ends of said upflow tubes and through said circuits, each of said conduits supplying said fluid to a plurality of said upflow tubes, and means for inhibiting the flow of fluid in those of said circuits having a relatively low heat absorption rate to compensate for the inequality of heat absorption in said circuits including flow restrictors disposed in at least some of said conduits, each of said flow restrictors having means forming gradually diminishing and increasing transverse flow areas to avoid abrupt changes of pressure therein.

2. In a forced circulation fluid heating unit, walls defining a furnace chamber, burner means supplying high temperature heating gases to said chamber for flow therethrough, a row of upwardly extending contiguous upflow tubes disposed in said walls and arranged for parallel flow of fluid therethrough and grouped into a plurality of parallel flow circuits some of which are arranged to absorb less heat than others, a fluid distributor vessel, means for supplying a vaporizable fluid to said vessel, a plurality of conduits connecting said vessel for parallel flow of said fluid to the inlet ends of said upflow tubes and through said circuits, each of said conduits supplying said fluid to a plurality of said upflow tubes, and means for inhibiting the flow of fluid in those of said circuits having a relatively low heat absorption rate to compensate for the inequality of heat absorption in said circuits including flow restrictors disposed in at least some of said conduits, each of said flow restrictors having means forming gradually diminishing and increasing transverse flow areas and an elongated throat portion to avoid abrupt changes of pressure in said flow restrictor, said throat portion being sized to effect a substantial portion of the total pressure drop across said flow restrictor by friction between the inner wall of said throat portion and the fluid flowing therethrough.

3. In a forced circulation fluid heating unit, walls defining a furnace chamber, burner means supplying high temperature heating gases to said chamber for flow therethrough, a row of upwardly extending contiguous upflow tubes disposed in said walls and arranged for parallel flow of fluid therethrough and grouped into a plurality of parallel flow circuits some of which are arranged to absorb less heat than others, a fluid distributor vessel, means for supplying to said vessel a vaporizable fluid containing contaminants which tend to be separated from the fluid upon the occurrence of abrupt pressure changes, a plurality of conduits connecting said vessel for parallel flow of said fluid to the inlet ends of said upflow tubes and through said circuits, each of said conduits supplying said fluid to a plurality of said upflow tubes, and means for inhibiting

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