March 1, 1960 W. L. PAULISON, JR 2,926,636

STEAM TEMPERATURE CONTROL

Filed Nov. 18, 1953 4 Sheets-Sheet 1

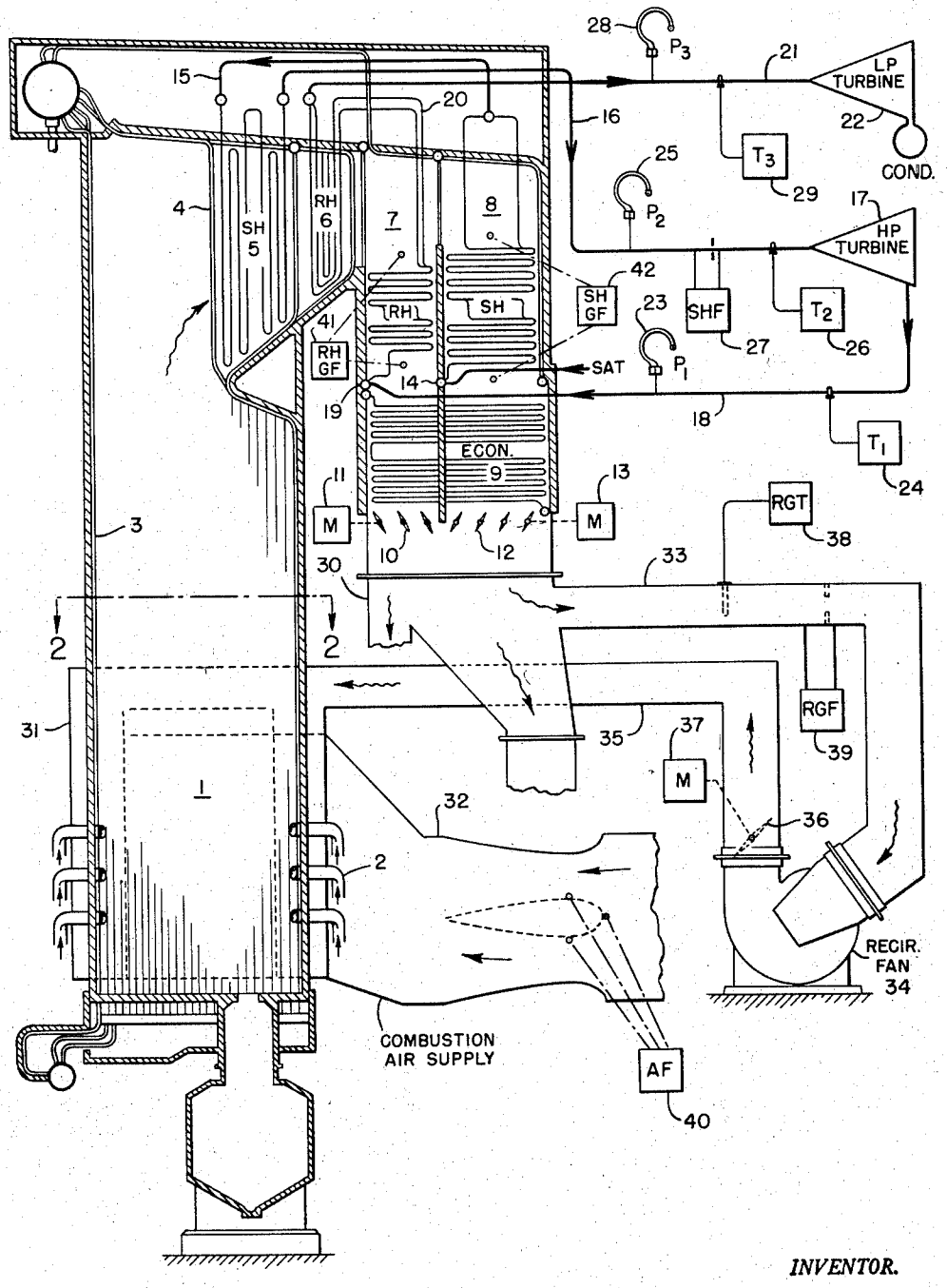


FIG. 1

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STEAM TEMPERATURE CONTROL

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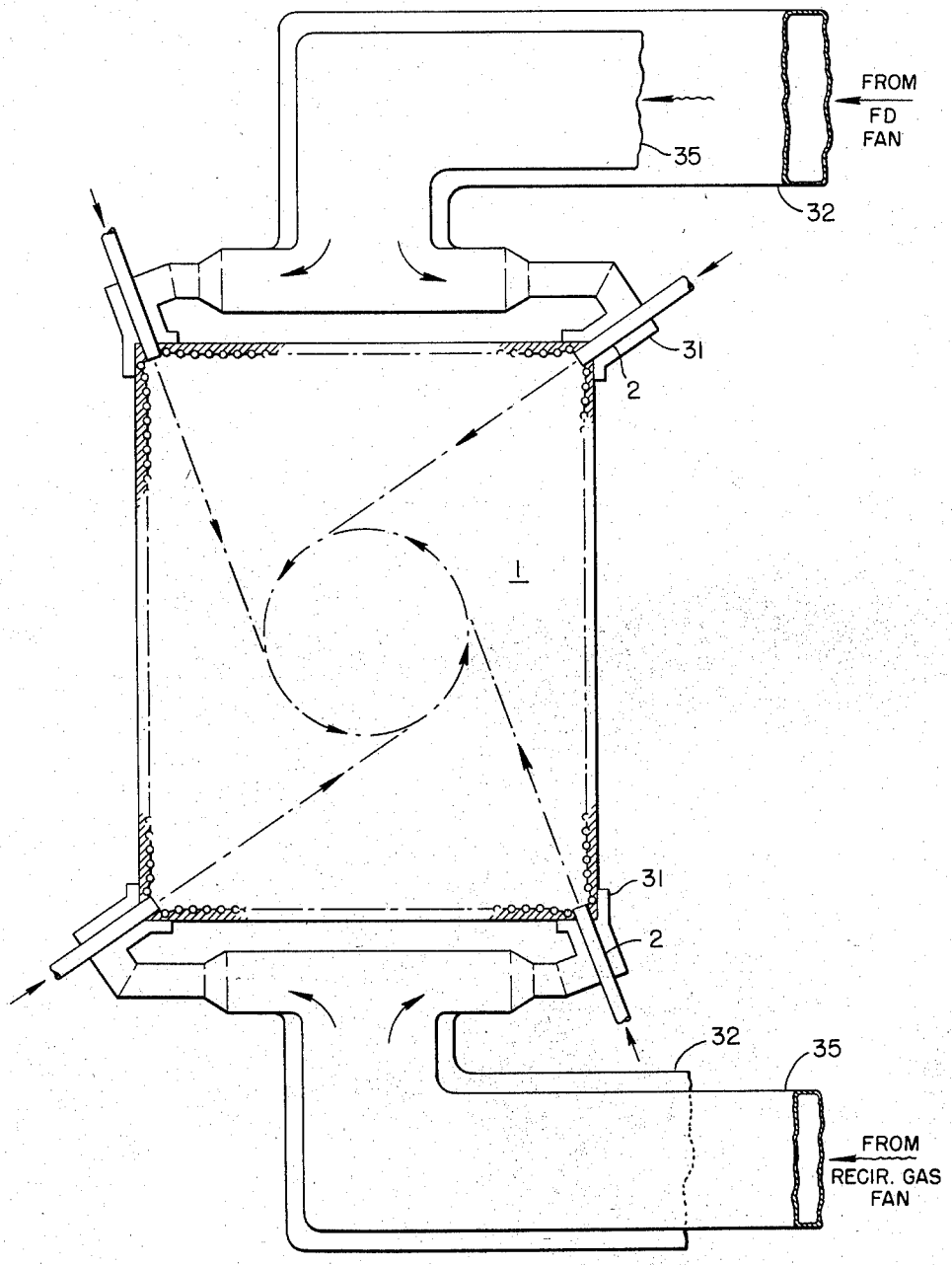


FIG. 2

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STEAM TEMPERATURE CONTROL

Filed Nov. 18, 1953 4 Sheets-Sheet 3

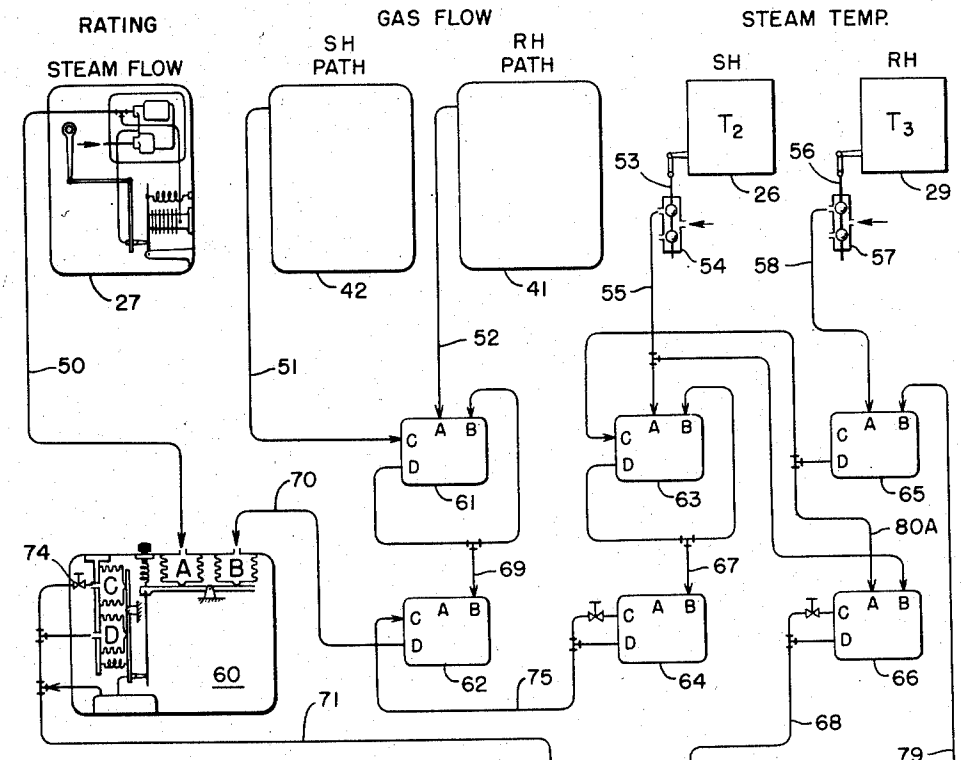


FIG. 4

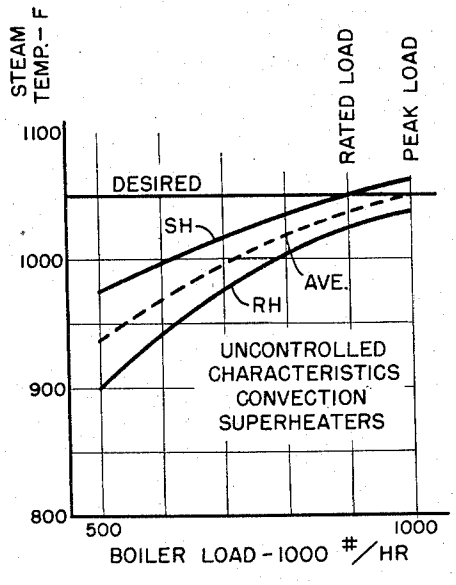


FIG. 3

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STEAM TEMPERATURE CONTROL

Filed Nov. 18, 1953 4 Sheets-Sheet 4

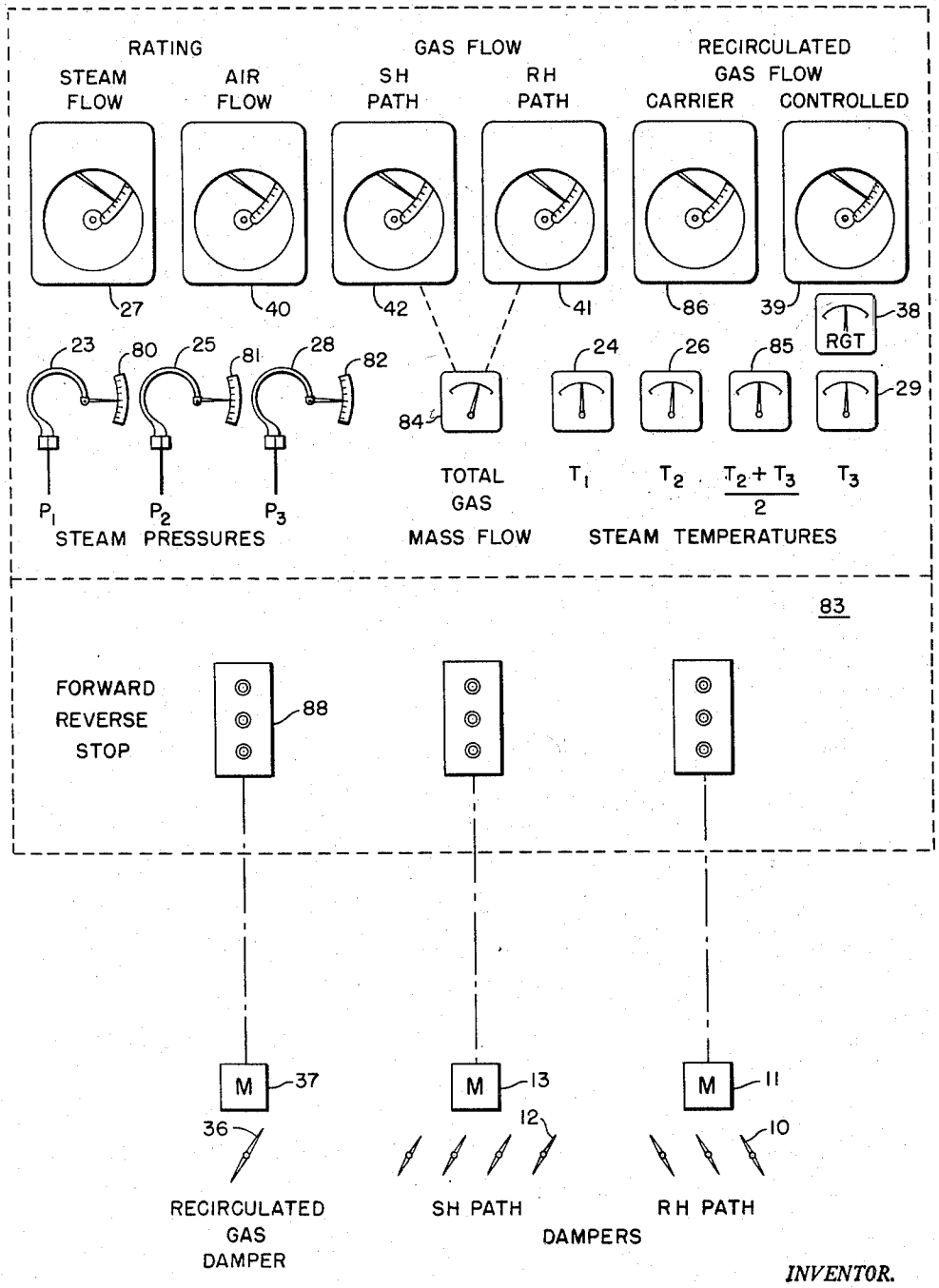


FIG. 5

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United States Patent Office 2,926,636
Patented Mar. 1, 1960

2,926,636

STEAM TEMPERATURE CONTROL

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Application November 18, 1953, Serial No. 392,968

7 Claims. (Cl. 122—479)

My invention lies in the field of steam power generation and particularly in the control of steam temperature in connection with present day vapor generators. I am particularly concerned with the problems encountered in units rated at 1,000,000 to 1,500,000 pounds per hour, operating at pressures of from 1500 p.s.i.g to 2000 p.s.i.g., and with final total steam temperatures of the order 1000 to 1050 F. Modern units of this type may have reheat provisions, taking steam exhausted from a high pressure turbine at say 600–800 p.s.i.g. and reheating this steam to a range of 1000–1050 F. The problems involved in the generation and close control of the properties of steam in such a unit are quite different now than was the case at the time of the inventions in this field which are shown in the prior art.

Superheat temperature control is particularly desirable in the generation of steam for the production of electrical energy in large central station power plants. In such plants, the upper limit of superheat temperature is governed by the materials and construction of the turbines served by the steam. In the interest of turbine efficiency the temperature of the steam delivered to the turbines should be maintained within close optimum limits throughout a wide range of operation.

I contemplate a unit wherein the furnace chamber is surrounded by walls of closely spaced bare tubes constituting a vapor generating section primarily radiantly heated. The superheating and reheating portions of the vapor flow path are preferably located beyond the gas outlet of the furnace in what is termed convection heating locations and, in the particular embodiment which I will describe, the primary superheating and reheating portions of the flow path are located in separate and parallel gas flow paths, with dampers for regulating the distribution of heating gases between the two vapor paths. The secondary superheater and secondary reheater may be located in series in the gas flow path between the furnace outlet and the division mentioned.

For any given furnace, as load increases, the rate of heat absorption does not increase as rapidly as the rate of heat liberation; therefore, the furnace leaving temperature will rise. With both the quantity rate and the temperature of the gases leaving the furnace increasing with load, it is apparent that a fixed surface convection superheater will receive a greater heat rate at higher loads than at lower loads and the heat transfer area is usually designed for receiving the volume and temperature of leaving gases at an expected rated load. Any further increase in the heat release rate, as by forcing operation beyond rated load to a possible peak load, supplies to the fixed superheater surfaces more heat by gas mass and by gas temperature than designed for and a corresponding excessive final steam temperature is experienced. On the other hand, at operations below rated load, the fixed superheater surface receives less volume and lower temperature gases leaving the furnace with a corresponding lowering of final steam temperature.

Inasmuch as the pressure and heat content per lb. of the low pressure steam returned to the reheater from the high pressure turbine exhaust decreases with reduction in load, while the pressure and heat content per lb. of the high pressure steam introduced to the superheater remains substantially constant with a corresponding variation in load, the customary installation of convection superheater and reheater will give a steam temperature-load graph which will slope downward from maximum load to low load, with the result that delivery temperatures from both the superheater and reheater will droop and the outlet temperature-load graph for the reheater will have a greater slope than the corresponding graph of the superheater. This is clearly shown in Fig. 3.

Due to the uncontrolled characteristics of convection superheaters, units of the type here under consideration are understood to be controllable as to final steam temperatures through an upper range of some 50% of load. That is, 50% of peak load, which may be some 10% beyond the normal rated load for the unit.

Such a unit is usually designed so that the final total temperature of the superheated steam and of the reheated steam will be at or above the optimum value throughout some portion of this upper 50% controllable rating. In other words the curves of Fig. 3 would be expected to cross the desired final temperature at some considerably lesser load value than shown in Fig. 3, for example, at 700,000 lb. per hr. load. The surfaces and passages would be so designed that the uncontrolled characteristics of each of the surfaces would cross the optimum final temperature line on the graph at 700,000 lb. per hr. load and lie above the optimum temperature value at all ratings from there to a peak of 1,000,000 lb. per hr. Operation below the crossing point would produce superheated stream and reheated steam at a final total temperature lower than the optimum and continued low-load operation is not expected.

In the present example, I contemplate a unit so designed as to normally have the optimum final steam temperature reached at or near peak load and thus the uncontrolled characteristic curves would lie below the optimum temperature value through the controllable operating range. Such design produces a saving in the expensive alloy superheater tubing but at the expense of normally expecting a final steam temperature below the optimum value. The present invention provides control for, in effect, raising the uncontrolled temperature curve until it coincides with the optimum or desired temperature curve throughout a major portion of the controllable load range. Preferably I accomplish this through recirculation of partially cooled gaseous products of combustion.

I preferably withdraw a controlled flow rate of the products of combustion from near the entrance to the air heater after the gases have left the superheating and economizer surfaces. The recirculated gas is introduced to the furnace near, or with, the elements of combustion and passes to the furnace outlet with the fresh products of combustion.

Recirculation of partially cooled products of combustion is not a new device. With a water cooled furnace it is known that the heat availability of the gases at the entrance to convection superheating surfaces is increased when the percentage recirculated is increased as rating decreases. As rating decreases from "rated load" the rate of gas recirculation is increased, thus relatively decreasing the absorption of heat by the radiant generating surface, by relatively increasing both the volume flow rate and the temperature of the gases leaving the combustion zone and entering the superheating surfaces. The present invention provides a control of the recirculation of products of combustion to lower the radiant heat absorption with decrease in rating and thus

increase the volume and temperature of gases leaving the furnaces and entering the superheating surfaces.

Due to the uncontrolled characteristic, operation in the controllable range will find the final total temperature of the superheated steam and of the reheated steam greater than that which is desired. Due to the difference in slope of the characteristic curve of the superheater and of the reheater it is seen that the two final temperatures may depart from equality or from desired relationship of values and this I correct by proportioning the total heating gases between the superheating and reheating convection surfaces.

In general then, I have provided a method and apparatus for maintaining optimum final temperature of the superheated steam and of the reheated steam for some upper controllable range of operation, as for example, 50%–100% peak load rating, through the agency of recirculation of partially cooled products of combustion and of gas proportioning over the parallel superheating and reheating surfaces and, where necessary, the two regulations simultaneously.

When, hereafter, I use the term "throttled" with reference to the position, or positioning, of a damper I intend to mean that the damper is in some position between "closed" and "open." If a damper is "closed" it is theoretically shutting off all flow of gases therethrough. If it is "open" then it theoretically allows flow of gases therethrough unimpeded by the damper. At any intermediate damper position the gas flow is "throttled" or impeded as to its free flow and, while it may be more strictly correct to speak of the gas flow as being "throttled in different degree" at different damper positions, it is not incorrect to say that the damper is "throttled" or in a throttling position. When the damper is moved in an opening direction or in a closing direction it is still in a throttling position so long as it is not "closed" or "open."

To reach the desired high superheated steam temperature, but not to exceed it, requires careful proportioning of the heat absorbing surfaces both for generating steam and for superheating it. But even if the desired superheated steam temperature be just obtained initially by very careful designing at some "rated load," the superheated steam temperature will vary during operation by reason of changes in cleanliness of the heat absorbing surfaces. Slag will form and adhere to the heat absorbing surfaces of the furnace thereby reducing the effectiveness of such surfaces and raising the furnace outlet temperature of the products of combustion. The furnace outlet temperature will also change with percent of excess air supplied for combustion, with the characteristics of the fuel burned, and with the rate of combustion and the corresponding rate of steam generation. All of these things will therefore affect the temperature of the gases, whether the superheating elements are located in series or in parallel paths. In other words, the theoretical characteristic curves are for ideal design conditions of operation and, while the general trend of the curve is followed with rating, the actual final temperature of the superheated steam or of the reheated steam may be above or below the theoretical curve or the optimum value at any time, for different reasons, and at different rates of operation. By the method and apparatus of my invention I tend at all times to return final superheated steam temperature and final reheated steam temperature toward optimum value, upon departure therefrom, regardless of the cause of such departure, through a selected controllable operating range. Below such a range, which may be for example the upper 50% of peak load capacity, the droop of the characteristic curves is such that optimum final temperatures are not expected to be achieved and normal operation is not expected within such lower ratings.

The principal object of the present invention is to so control a vapor generating and superheating unit of the

type having convection superheating surfaces as to tend to return final steam temperature toward optimum value upon any departure therefrom.

Another object is, in a unit of the type having convection superheating surfaces so designed that the uncontrolled characteristic temperature-load curve tends to cross the optimum temperature value at rated load, with a majority of expected load operation below rated load, and normally producing final steam temperatures lower than optimum, to properly control steam temperature toward optimum.

Still another object is to raise the final steam temperature values throughout the expected controllable range through recirculation of partially cooled products of combustion from beyond the convection superheating surfaces.

Still another object is to so control the final steam temperatures of a unit having convection superheating surfaces and convection reheating surfaces located in parallel gas paths that the two final temperatures will be of desired relative value, and accomplish this by distribution of heating gases over the two parallel paths.

Further objects will become evident from the following description in connection with the drawings.

In the drawings:

Fig. 1 is a somewhat diagrammatic sectional elevation of a steam generating and superheating unit of the type contemplated, having convection superheating and reheating surfaces located in parallel gas flow paths, and with gas recirculation provisions.

Fig. 2 is a somewhat diagrammatic sectional plan view through the furnace of Fig. 1, along the line 2—2 in the direction of the arrows, and showing schematically the supply of fuel, carrier gas, combustion air and recirculation gas to the furnace.

Fig. 3 is a graph of uncontrolled characteristics of convection superheating and reheating surfaces of a unit like that of Fig. 1.

Fig. 4 diagrammatically represents a pneumatic control system for the unit of Fig. 1.

Fig. 5 illustrates a manual control station for regulating the operation of the unit of Fig. 1.

Referring now to Fig. 1 I show therein in quite diagrammatic form, and not to scale, a vapor generating and superheating unit in connection with which I will explain my invention. The unit of the present example may operate at 1500 p.s.i.g. with optimum final superheated steam temperature of 1050 F. and optimum final reheat steam temperature also of 1050 F. The design is for rated load of 900,000 lb. per hr. and a peak capacity of 1,000,000 lb. per hr. The expected controllable operating range would be from 1,000,000 down to 500,000 lb. per hr. load.

The furnace 1 of the unit is supplied with fuel and air for combustion through a plurality of burners 2 (not detailed). Gaseous products of combustion leave the furnace after contacting the fluid heating surfaces thereof. The generator is of the radiant type, wherein the furnace 1 has walls 3 of vertical, closely spaced plain tubes constituting the vapor generating portion of the unit. Products of combustion pass upwardly through the furnace 1 in the direction of the arrow, through a tube screen 4, over a secondary superheating surface 5 and a secondary reheating surface 6, to the entrance of two parallel gas passages 7 and 8. In the gas passage 7 is located the primary reheat surface designated RH, while in the gas passage 8 is located the primary superheating surface designated SH. Spaced across the two gas passages 7, 8, beyond the convection surfaces RH, SH, is an economizer 9. Gas flow through the RH path is regulated by the positioning of dampers 10 through the agency of a motive means 11, while that through the SH path is regulated by the positioning of dampers 12 through the agency of a motive means 13.

Considering now the vapor flow path, it will be seen

that saturated steam from the separation drum enters a header 14, passes upwardly through superheating surface SH and through a conduit 15 to the secondary superheater 5. The superheated steam then leaves the unit through a conduit 16 to a high pressure turbine 17, exhausting through a conduit 18 to a header 19 entrance to the reheating surface RH. From the reheater RH the steam passes through a conduit 20 to the secondary reheater 6 from which it passes by way of a conduit 21 to a low pressure turbine 22 exhausting to a condenser. This, in simplified diagrammatic fashion, is the vapor cycle of the system. Pressure P1 of the steam leaving the high pressure turbine 17 is determined by a Bourdon tube device 23, while temperature T1 of the same steam is measured by a device 24. Pressure P2 and temperature T2 of the final superheated steam going through the conduit 16 to the turbine 17 are measured by the devices 25, 26 respectively. The weight rate of flow of superheated steam passing through the conduit 16 is continuously measured by a meter 27 designated SHF. In the reheated steam conduit 21 the pressure P3 is determined by a Bourdon tube device 28 while the temperature T3 is determined by a measuring device 29.

In the present example the fuel fired is pulverized lignite or lignite char from a bin system (not shown). Carrier fluid for the fuel to be burned in suspension is supplied by recirculated products of combustion which may be drawn through a duct outlet 30 following the dampers 10, 12. In the present example there are two constant speed carrier gas fans drawing from the duct 30 to supply a total of twelve burners 2. Approximately 100,000 lb. per hr. of carrier gas is required at rated load of 900,000 lb. steam per hr. and substantially the same 100,000 lb. per hr. of carrier gas at 50% peak load or 500,000 lb. steam per hr. It is not contemplated that any particular control of the rate of flow of carrier gas be had from the constant speed supply fans because, as load falls off a greater amount of recirculated gaseous products of combustion is desired in the furnace and a small portion of the total may continue to be supplied through the fuel feeder pipes. Control of the supply of fuel is had by pulverized fuel feeders to the individual carrier gas conduits. It may, in certain instances, be desirable to vary the rate of supply of carrier gas in accordance with rating or load.

The furnace is fired by a plurality of vertically spaced burners 2, corner located for tangential firing, and entering through burner boxes 31 which are supplied with fresh combustion air through ducts 32 from the forced draft fans. A recirculation duct 33 branches from the gas outlet beyond the dampers 10, 12 to a fan 34 which supplies a duct or ducts 35. Recirculation of gases is controlled by one or more dampers 36 positionable by motive means 37, and the duct 35 leads to the furnace 1 at or near the burner boxes 31.

Temperature RGT of the recirculated gas is measured by a device 38 while rate of flow RGF is measured by a device 39. An air flow rate meter 40 is arranged to continuously measure the rate of supply of fresh combustion air to the furnace 1.

It will be seen that three separate supplies of fluid are fed to the furnace, namely, the pulverized fuel in its carrier gas, fresh combustion air, and controllable recirculated products of combustion through the duct 35. It is contemplated that substantially no gas will be recirculated through the duct 35 to the furnace 1 at rated load of 900,000 lb. per hr., with a maximum of around 250,000 lb. per hr. at a rating of 500,000 lb. per hr.

Control of the fuel feeders and of the fresh combustion air supply through the duct or ducts 32 is preferably in accordance with steam pressure, in normal manner, and forms no part of the present invention. The present invention is particularly concerned with control of the damper 36 for supply of recirculated products of combustion and control of dampers 10, 12 for distribution of the heating gases over the SH and RH heating paths.

My present invention contemplates both method and apparatus for operating and controlling the operation of a vapor generating and superheating unit of the type illustrated and described, through the positioning of dampers 10, 12 and 36, in accordance with, or responsive to, measurement of variables such as pressure, temperature, or flow, of the superheated steam, or temperature or pressure of the reheated steam; as well as an indication of load upon the unit and a measurement of the recirculated gas temperature and flow rate. Furthermore, I take into account a measurement of the gas mass flow passing through the parallel convection heating paths 7 and 8.

In Fig. 1 I indicate at 41 a device for continuously determining the gas mass flow rate RHGF across the RH heating surface in gas path 7. Similarly the device 42 continuously determines the gas mass flow rate SHGF through the path 8 across the SH heating surfaces.

There is a clear distinction in the measurement of various fluid flows. A device 40 continuously determines the rate of supply of fresh combustion air to the furnace. The RGF device 39 continuously determines the rate of flow of recirculated gas fed to the furnace through the duct 35. Devices 41 plus 42 give the total of the heating gases passing over the surfaces RH and SH and therefore a continuous measurement of the total of products of combustion and excess air from the air supplied through duct 32 and the pulverized fuel; as well as gases recirculated through the duct 35 and the carrier gas which enters the furnace through the burners 2. While measurement of fresh combustion air by device 40 is one indication of load upon the unit, it is not necessarily a valid indication of the heating gases passing over the surfaces SH and RH for controlling the amount of recirculated gases entering through the duct 35 is part of the present invention.

Referring now to Fig. 3 it will be noted that I have therein plotted uncontrolled characteristic curves for convection heating surfaces, showing representative expected final steam temperatures for different boiler loads. I indicate a "peak Load" of 1,000,000 lb. per hr. primary steam flow and a "rated Load" of 900,000 lb. per hr., with the controllable range extending downward from peak Load to some rating such as 500,000 lb. per hr. Below this rating the drop-off of the uncontrolled characteristic curves is so rapid that, for the present designed unit, operation would not be expected beolw the controllable range.

For a desired final superheated steam temperature of 1050 F. and a desired final reheated steam temperature of 1050 F. the unit might be designed to have convection surface characteristics as shown. The SH curve might be designed to cross the temperature value of 1050 F. at rated load of 900,000 lb. per hr. and would reach some value of 1060 F. at peak load, if uncontrolled. At the same time the reheat characteristic curve RH might reach 900 F. at 500,000 lb. per hr. and 1040 F. at peak load with a temperature of 1025 F. at rated load. It will be noted that the RH curve drops off more rapidly with decrease in rating than does the SH curve. An average of the two curves would reach the desired value of 1050 F. at peak load. The particular problem to which my invention is directed is to tend to attain optimum or desired temperature of 1050 F. for both SH and RH throughout the controllable load range 500,000 lb. per hr. to 1,000,000 lb. per hr. The two characteristic curves are preferably raised through the agency of recirculation of gases and are individually turned toward the average value through gas distribution over the passages 7, 8. The method and apparatus for attaining this result will be explained.

Under certain conditions it may be that a unit of this type would have a desired final superheated steam tem-

perature of 1050 F. but the desired final reheated steam temperature might be 1000 F. In other words, a definite bias of 50 F. between the final temperatures would be desired. This I may accomplish through my invention.

It will be observed that, with the unit of the present contemplation, a considerable saving in expensive superheating and reheating surface is accomplished. Operation throughout the controllable range of boiler load would result in uncontrolled characteristic curves almost always below the desired final temperature and desirably the recirculation and gas distribution effects tend to raise the final steam temperature to the desired value. This design and operation compares with a design where a greater amount of expensive superheating surface is employed so that the uncontrolled characteristic curve crosses the desired value at some lower point of the controllable range, with consequent overheating possibilities throughout the normal operating portion of the range, and the necessity of attemperation or some similar device to take care of what would otherwise be excess heat in the final steam.

Referring now to Fig. 4 I illustrate therein, in quite diagrammatic fashion, a pneumatic control system for continuously and automatically regulating the final total temperature of the superheated steam and of the reheated steam, in connection with units such as that of Figs. 1, 2 and 3.

At 27 I show the steam flow rate measuring device as a controller of the type disclosed and claimed in the patent to Gorrie et al. 2,737,963 issued March 13, 1956. The device 27 establishes in the pipe 50 an air loading pressure continuously representative of rate of flow of superheated steam through the conduit 16 to the high pressure turbine 17. In similar fashion the controller 42 establishes a pneumatic loading pressure in a pipe 51 continuously representative of rate of flow of heating gases SHGF across the SH surfaces in heating path 8. Similarly the controller 41 establishes a pneumatic loading pressure in a pipe 52 continuously representative of the rate of flow of heating gases RHGF across the RH surfaces in the gas path 7.

A measuring device 26 of T2 is arranged to position the movable element 53 of a pneumatic pilot valve 54, establishing in a pipe 55 a fluid loading pressure continuously representative of final temperature of the superheated steam passing through the conduit 16 to the turbine 17. In similar fashion the temperature sensitive device 29 positions the movable element 56 of a fluid pressure pilot valve 57, establishing in a pipe 58 a fluid loading pressure continuously representative of the final temperature of the reheated steam passing through the conduit 21 to the low pressure turbine 22. The pilot valves 54 and 57 are of a known type as disclosed in the Johnson Patent 2,054,464.

Fluid pressure relays 60, 61, 62, 63, 64, 65 and 66 are included in the control circuit of Fig. 4, with only relay 60 shown in sufficient detail to illustrate the expansible-contractible chambers A, B, C and D of each. These relays are of the type disclosed and claimed in the patent to H. H. Gorrie No. 2,776,669 dated January 8, 1957. It may be said in general that relay 63 is arranged to produce in the pipe 67 a fluid loading pressure which is representative of the average of T2 and T3. Relay 66, on the other hand, is arranged to produce in the pipe 63, a fluid loading pressure which is representative of the difference between T2 and T3. The relay 61 is a totalizing relay providing in the output pipe 69 a fluid loading pressure representative of the total gas flow through the heating paths 7 and 8.

Relay 60 is a differential standardizing relay, the A chamber of which receives the loading pressure from the pipe 50 representative of boiler load or rating. For comparison the B chamber is loaded with a pneumatic fluid pressure from a pipe 70 output of the relay 62. Unbalance of the A—B relationshisp results in a change in control pressure in a pipe 71 which, leading through a manual-automatic selector station 72, and effective through a pipe 73, results in a positioning of the recirculation damper 36. The output of the relay 60 is available not only in the output pipe 71 but also in the D bellows chamber of the relay and, by way of a bleed valve 74, is slowly effective within the C chamber of the relay. Such a relay provides a proportional control with reset characteristics from a comparison of, or differential between, the value of the control pressure in pipe 50 and that in pipe 70. It provides, for the differential or discrepancy between these two loading pressures, a floating control of high sensitivity superimposed upon a positioning control of relatively low sensitivity. A function of the adjustable bleed connection 74, between the D and C chambers, is to supplement the primary control of the pressure developed in pipe 71, as representative of the differential between the pressures in pipes 50, 70, with a secondary control of the same or of a different magnitude, as a follow-up or suplemental action to prevent overtravel or hunting of the recirculation damper 36.

The selector station 72 provides the possibility of remote manual positioning of the damper 36 or of allowing the automatic control relay 60 to act directly through the selector station in automatically positioning the said damper.

The fluid loading pressure within pipe 69, representative of total gas flow over the surfaces 7, 8, is modified in a relay 62 by pressure within a pipe 75 the output of the relay 64, which receives by way of the pipe 67 a fluid pressure representative of the average of steam temperatures T2 and T3. Thus the relay 60 provides a balance between an index of rating and an index of total gas mass flow, modified by the average final temperature of the two superheat steam values.

At the same time the difference between T2 and T3, represented by the pressure within pipe 68, acts through a manual automatic selector station 76 and pipe 77 to simultaneously position the reheat control dampers 10 and superheat control dampers 12. The arrangement is such that the RH and SH dampers move in opposite directions, so that while one is closing the other is opening, at the same time.

The selector station 76 is of a type disclosed and claimed in the patent to P. S. Dickey 2,747,595 issued May 29, 1956, and has a bias or remote set point device 78 establishing in a pipe 79 a biasing loading pressure applicable upon the B chamber of relay 65. This provides the possibility of so manually biasing the RH temperature relative to the SH temperature, as previously explained, that one may desirably be held to a different value than the other. For example, the SH desired temperature might be 1050 F. while the desired RH final temperature might be 1000 F. This is provided by the manually adjusted amount of bias loading effective through pipe 79 upon the B chamber of the relay 65 which is receptive of a loading in pipe 58 representative of T3. The non-biased or biased output of relay 65, effective in pipe 80A, enters the A chamber of relay 66 and is compared to the pressure of pipe 55 which enters the B chamber of the relay. The output, in pipe 68, positions the SH and RH dampers oppositely for gas distribution over the flow paths 7, 8.

In general, the control is so arranged that steam flow rate, as an indication of boiler load, is compared to gas mass flow over the SH and RH heating surfaces and this ratio, relation, or comparison is readjusted in accordance with the final average of the steam temperatures or biased temperatures. Thus, for any given rate of boiler load, there is a corresponding normally expected total rate of heating gas mass flow over the two paths 7 and 8. If this balance of gas mass flow versus load, does not satisfactorily produce the desired final SH temperature and RH temperature then these temperatures, or the average thereof, biased or unbiased, is used to readjust the com-

parison, and the final loading pressure in pipe 73 positions the recirculation damper 36 to increase or decrease the gas mass flow over the two heating paths 7 and 8 until the average of the final temperatures is as desired. Thus, the rate of supply of recirculated heating gases through the duct 35 is regulated to produce the proper total gas mass flow over the surfaces 7, 8 and result in the desired average final temperature of T2 and T3; the normal operation being to increase the rate of flow of recirculated gas with a decrease in load to a full opening of damper 36 and maximum capacity of the recirculating fan 34. Operation at loads below this point, presumably below the preselected controllable range of rating, would probably be at an expected final steam temperature lower than that desired.

While the recirculation of gases is controlled to raise the average of the SH and RH temperatures toward optimum value, at the same time the proportioning dampers 10, 12 are operated, if necessary, by a signal, in pipe 68, calling for a change in the proportion of the gas flow over the superheating and reheating surfaces, acting upon the dampers 10 and 12, but in reverse direction. This portion of the control is not concerned with load or the average of the final RH and SH temperatures, but is concerned totally with the difference between the final temperatures SH and RH. The two temperatures T2 and T3 are continuously compared and the difference, if any, establishes a loading pressure in the pipe 68 to readjust gas distribution across the paths 7 and 8. If the final desired temperature T2 and T3 is the same, or a single temperature, then no biasing effect is added to the bellows B of relay 65 and a straight difference comparison of T2 and T3 acts to move the dampers 10, 12 for gas distribution. In that event, if the SH final temperature is higher than the RH final temperature and the two should desirably be together, regardless the absolute value, then the dampers are so automatically positioned that dampers 12 are throttled in a closing direction while dampers 10 are throttled in an opening direction, thus passing a greater proportion of the gases through path 7 than through path 8 and tending to raise the temperature T3.

On the other hand, if the final desired temperature T2 is different than that desired for T3, then the difference or bias is put into the system by way of manual loading device 78 and pipe 79, impressing a fluid control force upon the B chamber of relay 65.

It will be appreciated that steam flow as a load index may be used, or combustion air flow by way of meter 40 may also be used as a load index. Furthermore, that when I speak of the loading pressure in pipe 67 being representative of the average temperatures T2 and T3, this might equally as well be the total of the two for, after all, an average is only a total divided by some numerical value such as two.

In Fig. 5 I have schematically illustrated certain indicating or recording instrumentalities useful as a guide for manual remote control of the variable operating factors to allow operation of the unit in accordance with my new methods. Bourdon tubes 23, 25 and 28 are respectively sensitive to the steam pressures P1, P2 and P3 and visually indicate manifestations of such pressures upon scales 80, 81 and 82 respectively. The steam flow recorder 27 and the combustion air flow recorder 40 provide not only a visual manifestation but a permanent continuous record of these rates of flow. The points of measurement of these operational variables may be widely scattered but I preferably group the meters at a central control location having a manual control station 83. Preferably the panel board may be of the bench-board type wherein indicating and recording instruments are mounted on the vertical upper portion while the manual control instrumentalities are below.

Recorders 42 and 41 give a visual manifestation and continuous record of the heating gas flow over the SH path and the RH path respectively while indicator 84 shows the total gas mass flow over the heating surfaces of the paths 7 and 8. Indicators 24, 26 and 29 visually provide manifestations of temperatures T1, T2 and T3, while indicator 85 is arranged to show the average T2 plus T3 divided by 2. Recorder 86 continuously indicates and records the rate of recirculated gas flow used as a carrier fluid for the pulverized fuel, while recorder 39 indicates and records the rate of flow of the controlled recirculated gas through conduit 33.

Upon the bench-board portion of the panel I indicate three Forward-Reverse-Stop push button stations 88 respectively controlling motive means 37, 13 and 11 for remote manually positioning of the damper sets 36, 12 and 10.

It will now be clear that my improved methods of operation of the unit may be manually performed by an operator located at the station 83 observing the measuring instrumentalities, and selectively remotely activating the motive means 37, 13 and 11 for positioning the dampers 36, 12 and 10. Selective and sequential operation may be obtained, as well as proper proportioning of the gases over the superheater and reheater paths 7 and 8, with or without recirculation being accomplished.

It is understood in this art that either vapor out-flow rate or air flow rate may be used as an index of output or boiler rating.

While I have chosen to illustrate and describe certain preferred embodiments of my invention, it will be appreciated that the invention may be embodied in other forms, and thus I do not desire to be limited to the specific showings disclosed.

What I claim as new, and desire to secure by Letters Patent of the United States, is:

1. In an apparatus for controlling the operation of a vapor generating and superheating unit of the type having a convection superheater and a convection reheater disposed respectively in divided and separate parallel structurally defined gas flow paths from a common combustion zone, the combination including, load index means establishing a fluid loading pressure continuously representative of rate of flow of generated vapor, flow measuring means establishing a fluid loading pressure continuously representative of heating gas flow rate passing over the convection superheater, other measuring means establishing a separate fluid loading pressure continuously representative of the heating gas flow rate over the convection reheater surface, relay means totalizing the effects of said second and third mentioned fluid loading pressures and providing a resultant fluid loading pressure continuously representative of the total heating gas flow over the two divided heating paths, comparison relay means receiving the first mentioned fluid loading pressure and the said resultant fluid loading pressure and providing a control loading pressure representative of the said comparison, means for recirculating a controllable portion of the products of combustion from beyond the said heating paths back to the combustion zone, and control means responsive to said control pressure and arranged to regulate the recirculating means, said control means acting in general to increase the rate of recirculated gases as unit load decreases to thereby proportionately increase the heat availability of the heating gases entering the convection heating surfaces.

2. The combination of claim 1 including first temperature measuring means establishing a fluid loading pressure continuously representative of final superheat temperature, second temperature measuring means establishing a second fluid loading pressure continuously representative of final reheat temperature, and averaging relay means receiving and responsive to the first and second temperature representative fluid pressures and establishing an output fluid loading pressure representative continuously of the average of said temperatures and arranged to modify the positioning of the recirculation control means.

3. The combination of claim 2 including difference relay means receiving and responsive to said first temperature measuring fluid pressure and said second temperature measuring fluid pressure and providing a difference representative outlet fluid loading pressure, damper means for proportioning the heating gases over the two convection heating paths, and control means for said damper means responsive to the said difference representative outlet fluid loading pressure.

4. The method of operating a vapor generating unit having a convection superheater and a convection reheater disposed respectively in divided and separate parallel gas flow paths from the same combustion zone, the superheater and reheater having uncontrolled load-temperature characteristic curves with different degrees of slope; the method including recirculating partially cooled products of combustion from beyond the convection surfaces back to the furnace through a selected controllable range of loads, measuring the rate of flow of vapor generated, measuring the rate of flow of heating gases over the two parallel paths, regulating the rate of gas recirculation in accordance with the difference of such measurements, the regulating acting generally in direction to decrease gas recirculation rate as heating gas flow rate predominates over generated vapor flow rate and vice versa, and modifying the regulation in accordance with departure of the average of the two final vapor temperatures from an optimum value, the modifying acting generally in direction to decrease gas recirculation rate as the average of the two final vapor temperatures tend to depart above optimum value.

5. Apparatus for controlling the operation of a vapor generating and superheating unit of the type having a convection superheater and a convection reheater disposed respectively in divided and separate parallel structurally defined gas flow paths from a common combustion zone, including in combination, fan and duct means for recirculating a controllable portion of the products of combustion from beyond the said heating paths back to the combustion zone, load index determining means, means measuring total heating gas flow over the two paths, comparison means for the index means and measuring means arranged to regulate the recirculation means in direction acting in general to increase the rate of recirculated gases as unit load decreases to thereby proportionately increase the heat availability of the heating gases entering the convection heating surfaces, means continuously determining the average of the final superheated vapor temperature and the final reheated vapor temperature, and means responsive to said average determining means arranged to modify the action of the comparison means in regulating the recirculating means in direction tending to increase recirculation rate as the said average of temperatures tends to decrease further below optimum value and vice versa.

6. Apparatus for controlling the operation of a vapor generating and superheating unit of the type having a convection superheater and a convection reheater disposed respectively in divided and separate parallel structurally defined gas flow paths from a common combustion zone, including in combination, fan and duct means for recirculating a controllable portion of the products of combustion from beyond the said heating paths back to the combustion zone, load index determining means, means measuring total heating gas flow over the two paths, comparison means for the index means and measuring means arranged to regulate the recirculation means in direction acting in general to increase the rate of recirculated gases as unit load decreases to thereby proportionately increase the heat availability of the heating gases entering the convection heating surfaces, means continuously determining the average of the final superheated vapor temperature and the final reheated vapor temperature, means responsive to said average determining means arranged to modify the action of the comparison means in regulating the recirculating means in direction tending to increase recirculation rate as the said average of temperatures tends to decrease further below optimum value and vice versa, separate damper means for controlling the heating gas flow through the two parallel convection heating paths, control means for the separate damper means arranged to throttle the same in opposite direction so that one will be opening while the other is closing, means continuously determining the differenec in final superheated vapor temperature and final reheated vapor temperature positioning the damper control means in direction tending to restore the said difference of temperatures toward predetermined value, and manually actuated means for biasing the damper control means to maintain one of said final temperatures at a different temperature value than the other.

7. Apparatus for controlling the operation of a vapor generating and superheating unit of the type having a convection superheater and a convection reheater disposed respectively in divided and separate parallel structurally defined gas flow paths from a common combustion zone, including in combination, fan and duct means for recirculating a controllable portion of the products of combustion from beyond the said heating paths back to the combustion zone, load index determining means, means measuring total heating gas flow over the two paths, means responsive to said load index determining means arranged to regulate the recirculating means in direction acting in general to increase the rate of recirculated gases as unit load decreases to thereby proportionately increase the heat availability of the heating gases entering the convection heating surfaces, and including a comparison means for the load index means and measuring means arranged to modify the regulation of the rate of recirculated gases by the load index determining means to maintain a predetermind relationship between the unit load and rate of recirculated gases.

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