United States Patent

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[54] **FORCED CIRCULATING STEAM GENERATOR AND METHOD OF GENERATING STEAM**

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[30] **Foreign Application Priority Data**

Dec. 24, 1969 Switzerland 19211/69

[52] **U.S. Cl.** ..... **122/406 S**

[51] **Int. Cl.** **F22b 29/06**

[58] **Field of Search** ..122/406 S, 406 ST, 451, 451 S, 122/448, 448 S

[56] **References Cited**

UNITED STATES PATENTS

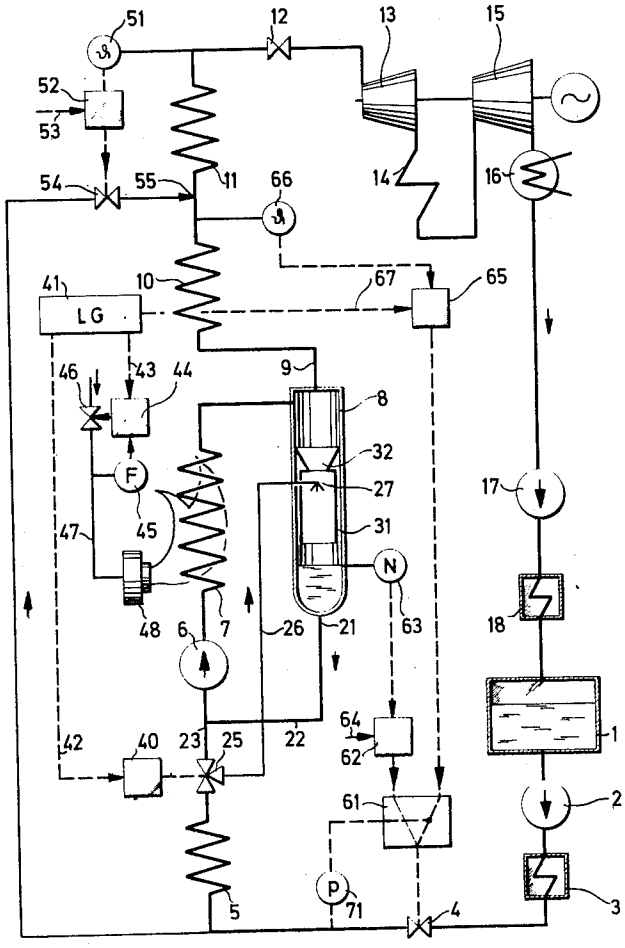
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[57] **ABSTRACT**

The steam generator is operable at supercritical pressure at the upper end of the load range and is operable under sliding pressure control. At a pressure near critical, the working medium entering the evaporator is heated by heat obtained from the steam generated in the evaporator. This is accomplished by spraying a portion of the working medium into the separator to mix with all the working medium passing from the evaporator before passing to the evaporator or by spraying all the fresh working medium into a portion of the working medium passing from the evaporator prior to passage to the evaporator.

5 Claims, 12 Drawing Figures



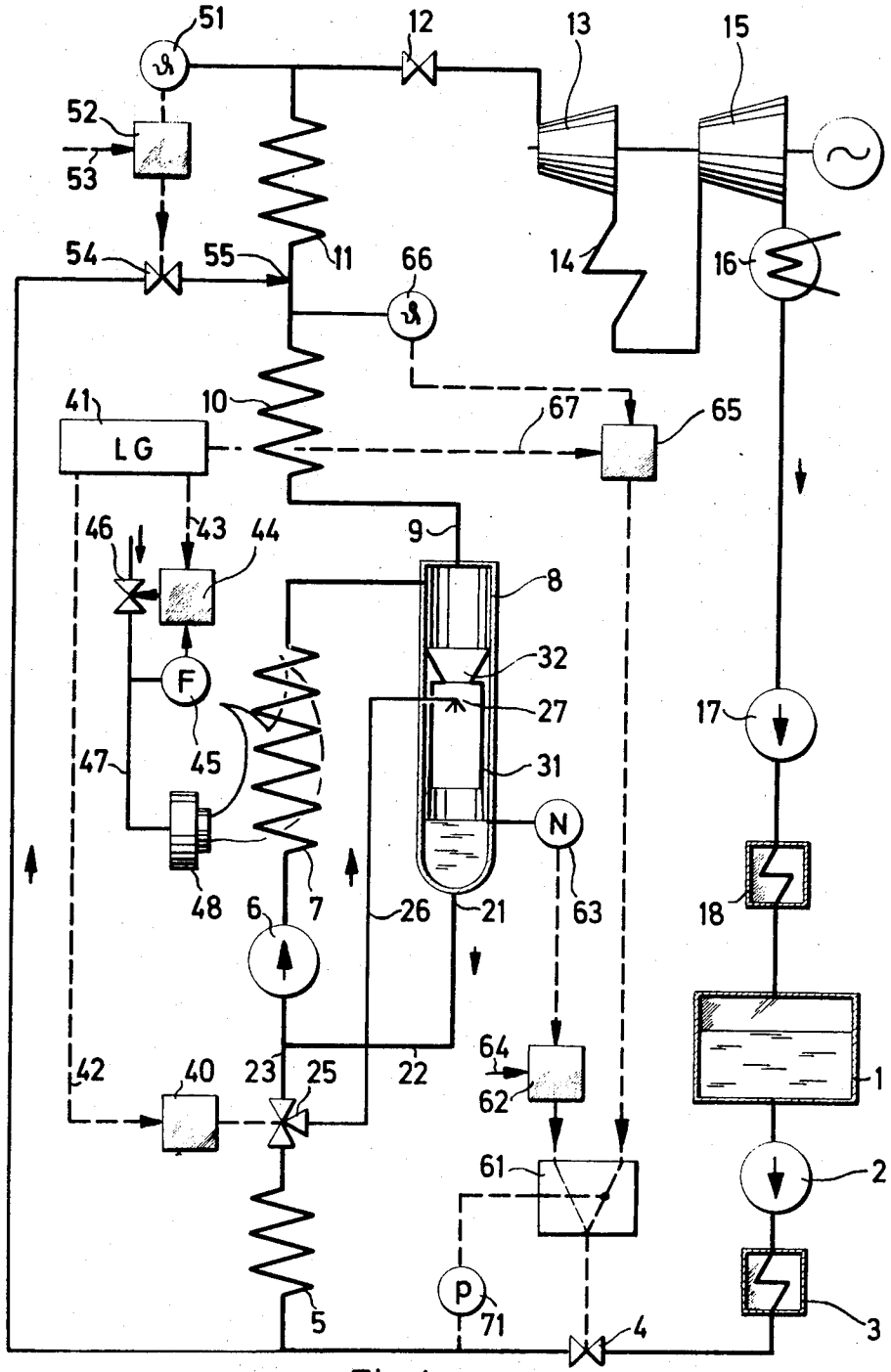


Fig.1

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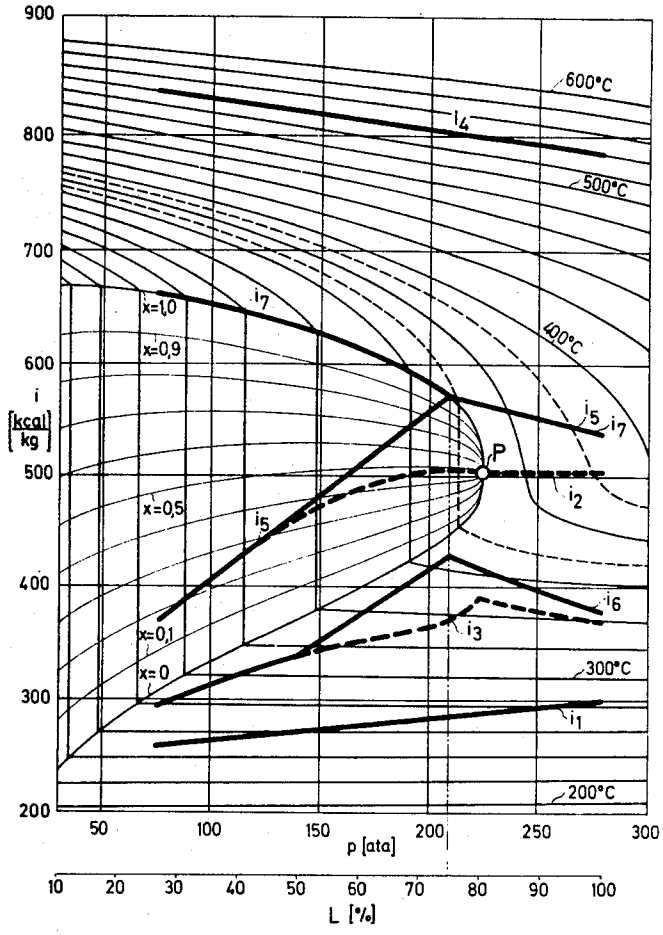


Fig. 2

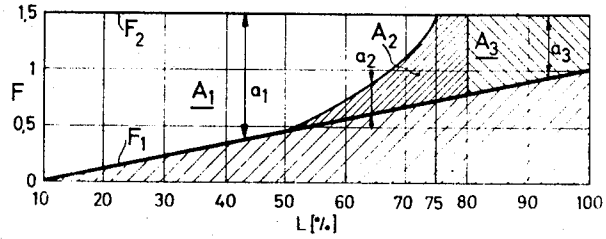


Fig. 3

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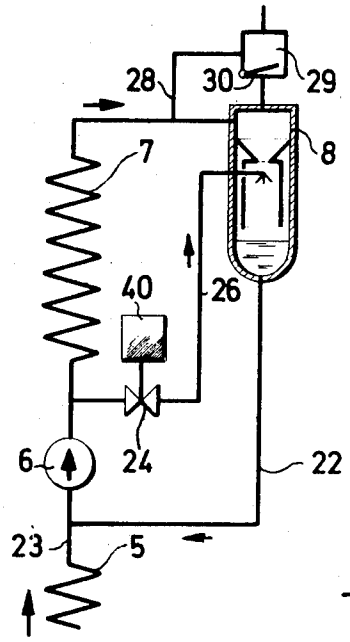


Fig. 4

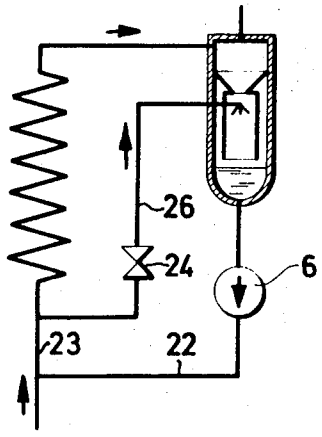


Fig. 5

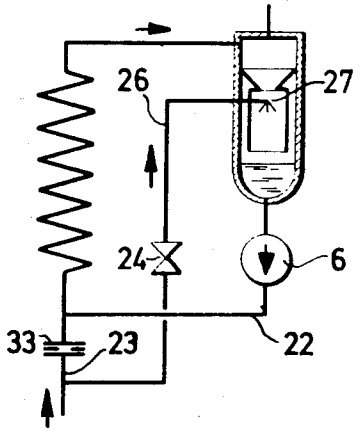


Fig. 6

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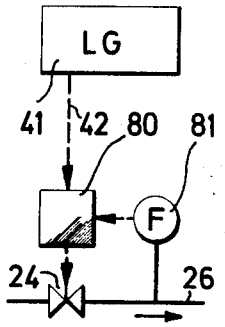


Fig. 7

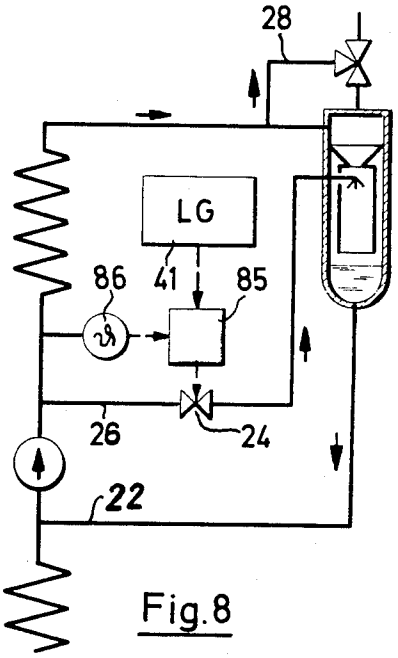


Fig. 8

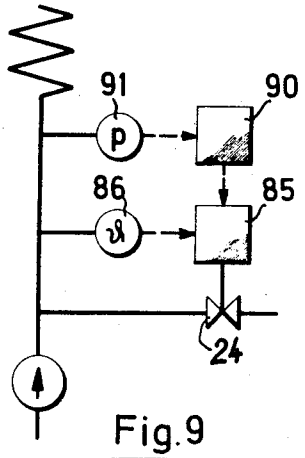


Fig. 9

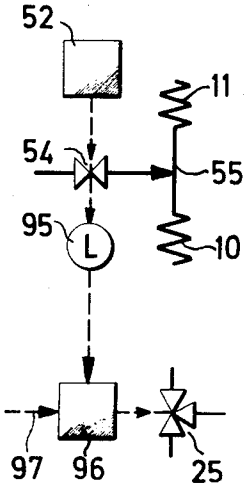


Fig. 10

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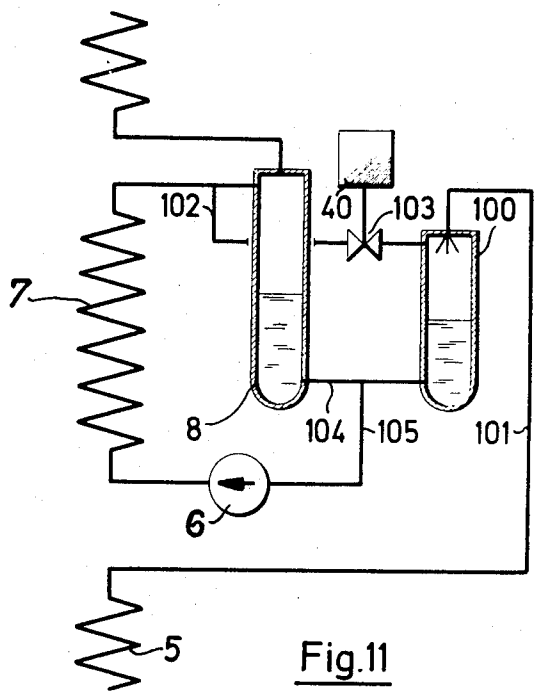


Fig.11

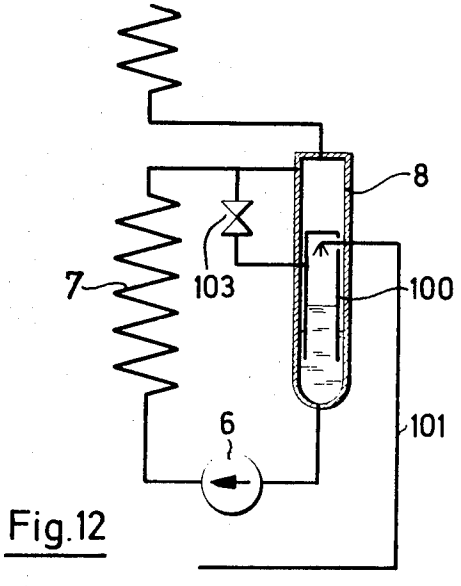


Fig.12

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FORCED CIRCULATING STEAM GENERATOR AND METHOD OF GENERATING STEAM

This invention relates to a forced circulation steam generator and a method of generating steam.

Forced circulation steam generators have been known to be constructed with an evaporator at least a part of which lines a combustion chamber, a superheater downstream of the evaporator with respect to the flow of working medium, and a return line for returning part of the working medium leaving the evaporator through a circulating pump to the inlet to the evaporator. Further, it has generally been preferable to return only the liquid part of the working medium to the evaporator and to conduct only the vaporized part to the superheater in order to avoid cavitation in the circulating pump and in order to supply the superheater with a working medium which is as homogeneous as possible. In order to accomplish such a purpose, a separator has been used. However, this has resulted in the disadvantage that the amount of heat which can be absorbed in the evaporator for each kilogram (kg) of the working medium diminishes rapidly near the critical pressure due to the rapidly diminishing saturated steam enthalpy which occurs near the critical pressure as the pressure increases to the critical point, a feature which conversely requires a steep increase of heat consumption in the superheater if the final temperature is to be maintained at a constant value. This phenomenon which results from the properties of water can be compensated by the injection of a large quantity of water into the upstream or downstream connected heating surface under conditions of small load and correspondingly small pressure. Thus, by reducing the amounts of water injected, it is possible for the amount of heat available for superheating to be increased in the region of the critical pressure. Such a procedure, however, requires a large superheater and a small evaporator. This is a disadvantage because the evaporator represents a less expensive heating surface than the superheater due to the lower working medium temperature prevailing in the evaporator. Further, this procedure becomes particularly detrimental if the combustion chamber and, in appropriate cases, the other boiler walls are lined with evaporator tubes, and more particularly, if these tubes are welded to each other in a gas-tight manner. As a result, since the evaporator surface and the amount of heat incident thereon are very large, particularly in combustion chambers for firing coal, the combustion chamber can only be just lined with evaporator heating surfaces, even in normal sub-critical operation where the amount of evaporation heat is large.

Accordingly, it is an object of the invention to increase the amount of heat absorbed in the evaporator in the region of the critical pressure in a simple efficient manner.

It is another object of the invention to permit the lining of a combustion chamber of a steam generator operable at super-critical and near critical pressures with evaporator tubes which are secured together in gas-tight manner.

Briefly, the invention provides a method of generating steam in a forced circulation steam generator having an evaporator, at least part of which lines a combustion chamber, and at least one superheater downstream of the evaporator with respect to the flow of working medium. In accordance with the method, the steam generator is operated with a sliding pressure control and operates at supercritical pressure at the upper end of the load range. In use, part of the working medium leaving the evaporator is returned to a point upstream of the evaporator and the remainder is passed to the superheater. However, when the generator operates at least in the load range at which the working medium pressure is in the vicinity of the critical pressure, the working medium entering the evaporator is heated by heat from the steam generated by the evaporator.

The forced circulation steam generator of the invention is also provided with a return line for returning part of the working medium leaving the evaporator through a circulating pump to the inlet to the evaporator, and means for heating the working medium entering the evaporator with heat from the steam generated by the evaporator.

In order to heat the working medium flowing into the evaporator, various embodiments can be used. In one embodiment, a three-way valve is interposed in a line feeding the working medium to the evaporator and connects the line to the interior of a separator in which the working medium passing from the evaporator is separated into vapor and liquid. When heating of the working medium is required, the three-way valve delivers a portion of the fresh working medium into the steam in the separator. This allows the injected working medium to be heated while the steam temperature is reduced. The return flow of working medium from the separator to the evaporator is in a heated state. In other embodiments, various valves are incorporated in an injection line between the feed line to the evaporator and the separator. In still other embodiments, the entire flow of fresh working medium to the evaporator is first sprayed into a chamber or mixer and mixed with a portion of the working medium flowing from the evaporator in order to heat the fresh working medium prior to entry into the evaporator.

These and other objects and advantages of the invention will become more apparent from the following detailed description and appended claims taken in conjunction with the accompanying drawings in which:

FIG. 1 diagrammatically illustrates a forced circulation steam generator according to the invention;

FIG. 2 illustrates an enthalpy-pressure/load diagram;

FIG. 3 diagrammatically illustrates the characteristics of the various mass flows plotted against the load in the steam generator shown in FIG. 1 and in relation to FIG. 2;

FIGS. 4, 5 and 6 each illustrates a means for regulating the flow of working medium through the injection line to the separator of a steam generator as in FIG. 1;

FIGS. 7, 8, 9 and 10 each illustrate a means for regulating a valve in the injection line;

FIG. 11 illustrates a modified steam generator according to the invention in which the fresh working medium is sprayed into a mixer for mixing with a portion of the working medium leaving an evaporator; and

FIG. 12 illustrates a further embodiment similar to FIG. 11.

Referring to FIG. 1 the forced circulation steam generator has a circulation system in which a working medium is fed from a feed water vessel 1 by a feed

pump 2 via a high pressure preheater 3 and a feed valve 4 to an economizer 5. The economizer 5, in turn, delivers the flow of working medium through a circulating pump 6 which pumps the flow to an evaporator 7 and a separator 8. The separator 8 is connected at the upper end to a duct 9 to pass a generated vapor through the duct 9 via a first superheater 10, a final superheater 11 and a live steam valve 12 to a high pressure turbine 13. After expansion in the turbine 13, the steam flows via a reheater 14 to a low-pressure turbine 15 and the steam is then condensed in a condenser 16. The condensate thus yielded passes into a condensate pump 17 and then via a low-pressure preheater 18 to the feed vessel 1.

The evaporator 7 comprises a plurality of vertical tubes (not shown) which are connected in parallel and are welded to one another in gas-tight manner and which form the wall of a combustion chamber as is known. A forced circulation of the working medium is provided in order to ensure a flow of sufficient working medium in the tubes. To this end, the working medium is withdrawn from the separator 8 from a lower outlet 21 and is returned by means of a line 22 into a line 23 which connects the economizer 5 to the circulating pump 6.

A three-way valve 25 is interposed in the line 23 and connects with an injection line 26 which extends into the separator 8 and terminates in a condenser in the form of a spray distributor 27 and an insert 31 which protects the wall of the separator 8 from thermal shock and at the same time provides a restrictor 32. This restrictor 32 acts in the manner of a non-return valve permitting downward flow but preventing upward flow so that different steam temperatures in the spaces above and below the restrictor 32 do not, in practice, compensate each other.

The three-way valve 25 is operated by a controller 40 as is known which is provided with a set-value signal from a load signal generator 41 via a signal line 42. A fuel set-value is also transmitted by the load signal generator 41 through a line 43 to a fuel flowrate regulator 44 which, in turn, receives a measured value signal from a flow metering apparatus 45 in a fuel line 47 which extends to a burner 48 for the combustion chamber to drive a valve 46 in the fuel line 47.

The live steam temperature is regulated by being measured by a temperature sensing element 51 at the exit of the final superheater 11, a measured value signal being formed in the temperature sensing element 51 in response to the temperature measured. The measured value signal is transmitted to act upon a temperature regulator 52 in which a control signal is formed by reference to a set-value supplied through a controller input 53 to influence a water injection valve 54 in a branch line extending in parallel to the economizer 5. The water is injected into the flow of working medium at a position 55 in the line connecting the first superheater 10 and the final superheater 11.

In the interests of completeness FIG. 1 also illustrates one of the possible methods of influencing the supply of feed water. That is, the feed valve 4 is driven by a changeover apparatus 61 having a first input which is connected via a signal line to a level controller 62. The input variables of the controller 62 includes a measured value signal originating from a level measuring apparatus 63 for detecting the level of liquid in the separator 8, and a set-value signal supplied via a line 64. A second input of the changeover apparatus 61 is fed by the output signal of a temperature regulator 65, which has as inputs a measured value comprising the signal of a temperature sensing element 66 situated at the output of the first superheater 10, and a set-value signal comprising a signal derived through a line 67 from the load signal generator 41. The changeover element 61 is actuated by a pressure sensing element 71 in the feed line by the economizer 5 which functions to connect the temperature regulator 65 to the valve 4 if a supercritical pressure prevails and to connect the level controller 62 to the valve 4 if a subcritical pressure prevails.

The system illustrated operates so that in the event a load rises to a value near the critical pressure, the three-way valve 25 is set so that part of the working medium from the economizer 5, initially constantly increasing and thereafter diminishing with a rising load, is injected into the separator 8. This injection condenses a part of the steam generated in the evaporator 7 and accordingly, the entry as well as the exit enthalpy of the working medium flowing through the evaporator 7 is increased to a greater or lesser extent. The point at which the evaporator initially delivers 100 percent steam thus occurs at a smaller load, a feature which will be explained below.

Alternatively, a flow regulating pump (not shown), controlled by the controller 40, may be provided in the line 26 in place of the three-way valve 25. Such a pump would also offer the advantage of a smaller pressure drop in the line 23.

Referring to FIG. 2, curves of identical temperature and identical wetness are plotted for the generator of FIG. 1 in the enthalpy (i)-pressure (p) diagram to form a parameter network in which:

- i_1 = the characteristic of the exit enthalpy of the economizer 5;
- i_2 = the characteristic of the exit enthalpy of the evaporator 7 on the assumption that the working medium entering into the evaporator is not heated in accordance with the invention;
- i_3= the characteristics of the temperature of the mixture of liquid and vapor at the entry to the evaporator 7, also on the assumption that the working medium is not heated in accordance with the invention;
- i_4= the characteristic of the live steam enthalpy corresponding to 540°C;
- i_5= the characteristic of the exit enthalpy of the evaporator 7 when the steam generator is operated in accordance with the invention;
- i_6= the characteristic of the entry enthalpy of the evaporator 7, also when the steam generator is operated in accordance with the invention;
- i_7= the characteristic of the entry enthalpy of the first superheater 10 when the steam generator is operated by the method according to the invention.

The scale of load (L), expressed as a percentage of the full load is plotted at the base of the diagram in parallel to the pressure scale and extends proportionally thereto. As shown by this relationship, the diagram refers to a vapor or steam generator operated in

accordance with a previously proposed sliding pressure system; the full load of this team generator in the supercritical pressure range being at $p_{100\%} = 280$ atmospheres absolute. It is assumed that the circulating pump **6**, connected into the feed system delivers a constant flow of 1.5 times the full load boiler flow.

When the steam generator is operated in accordance with previously proposed methods, that is, without the working medium being heated as above, the curve of the entry enthalpy of the first superheater **10** would extend along the curve $x = 1.0$ to the point P and from there, with rising load, along the i_2 curve, that is, with a distinct instability at the point P (80 percent of the full load). This means that $i_4 - i_2 = 798 - 505 = 293$ kcal/kg of heat must be supplied to the heating surfaces downstream of the entry to the first superheater at 80 percent of full load while only $i_4 - i_7 = 805 - 570 = 235$ kcal/kg, that is, approximately 20 percent less, must be supplied at 75 percent of full load (210 atm abs). To enable the superheater to satisfy the conditions of the 80 percent full-load point, the superheater **10** must therefore be constructed of a size approximately 20 percent larger than that corresponding to the requirements of the 75 percent of full load point.

However, when the steam generator is operated in accordance with the invention, the enthalpy of the working medium at the exit from the evaporator **7** is raised to the curve i_5 so that the entry enthalpy of the first superheater **10** follows the curve i_7. This is achieved by increasing the entry enthalpy of the evaporator **7** in accordance with curve i_6. The bend extending between the curves i_4 and i_7 is of a width which increases slightly and practically constantly with increasing load and provides a practically feasible characteristic. This means that injections provided in the superheater zone can be operated with amounts of water representing a small percentage over the entire load range.

Referring to FIG. **3**, the characteristic of different working medium flow rates (F) are plotted against load; the straight heavy line $F_1 \approx L$ referring to the characteristic of the amount of steam produced or of the amount of feed water supplied to the steam generator. No reference is made to the amount of injection water supplied to the superheaters in the same way as in FIG. **2**. The straight line $F_2 = 1.5$ represents the working medium flow delivered by the circulating pump **6**. The ordinate section a_1 of the area A_1 which is not hatched corresponds to the amount of water separated by the separator **8** and thereafter returned to the circulating pump **6**. The ordinate section a_2 of the closely hatched area A_2 represents the amount of working medium condensed in the separator **8** and subsequently circulated, while the ordinate section a_3 of the hatched area A_3 represents the steam discharged in the supercritical condition from the separator **8** through the lower outlet **21**.

Referring to FIG. **4** wherein like reference characters indicate like parts as above the injection line **26** branches off from the line **23** downstream of the circulating pump **6** and the three-way valve **25** is replaced by a simple regulating valve **24** which is controlled by the controller **40**. An advantage of this system is that the pressure drop in the connecting line **23** is reduced. However, it has the disadvantage that the evaporator **7** is supplied with a correspondingly small amount of working medium in the load range in which working medium flows through the injection line **26**.

In addition, a line **28** is connected in the line between the evaporator **7** and separator **8** to bypass the separator **8** and extends into a chamber **29** with a flap **30**. The flap **30** is operated in accordance with the pressure in a known manner to be positioned horizontally in the supercritical pressure range to thus close the supply line extending from the separator **8**, or vertically in the subcritical pressure range so that the bypass line **28** is closed at the point at which the line **28** extends into the chamber **29**. During operation, the pressure loss across the separator **8** can be eliminated for the supercritical pressure range by employing the bypass line **28**, chamber **29** and flap **30**.

Referring to FIG. **5** wherein like reference characters indicate like parts as above, the circulating pump **6** is situated in the return flow line **22**. This transfers some of the pump load to the feed pump **2** (FIG. **1**). In this case, the injection line **26** is also attached to the connecting line **23** downstream of the connection of the return line **22**, a feature resulting in the same advantages and disadvantages as the modification illustrated in FIG. **4**.

Referring to FIG. **6** wherein like reference characters indicate like parts as above, the circulating pump **6** is situated in the return line **22** and the return line **22** joins the connecting line **23** downstream of the injection line **26**. In order to provide sufficient injection pressure at the spray distributor **27** and in order to avoid the provision of a three-way valve, a restrictor **33** is provided in the connecting line **23** downstream of the injection line inlet.

Referring to FIG. **7** the control of the regulating valve **24** (FIGS. **4** to **6**) or of the three-way valve **25** (FIG. **1**) can be carried out by a load signal generator **41** which transmits an injection flowrate set-value signal instead of a position set-value signal through a signal line **42** to a controller **80** to which a signal, formed by an injection flow-metering apparatus **81** connected in the injection line **26**, is supplied as the measured value. By comparison with the control system illustrated in FIG. **1**, this system offers the advantage of avoiding injection flowrate faults resulting from blockages or washing out.

Referring to FIG. **8**, an alternative control circuit can have the load signal generator **41** transmit a load-dependent temperature set-value signal to a temperature controller **85** for comparison in the controller **85** with an actual value signal provided by a temperature-sensing element **86** connected downstream of the return line **22** to the connecting line **23**. The output of the controller **85** operates the valve **24** or the three-way valve **25**.

Referring to FIG. **9**, a control system similar to that shown in FIG. **8**, can obtain the set-value signal for the temperature controller **85** from a function generator **90** which produces a set-value signal depending on the boiler pressure p as measured by a pressure measuring apparatus **91** connected to the connecting line **23** between the pump **6** and evaporator **7** and to the input of the function generator **90**.

Referring to FIG. **10**, a cascade control can also be used for controlling the three-way valve **25** or the regu-

lating valve 24. Such a control is based on a cascade control of the water injection valve 54 and utilizes a displacement transducer 95 which converts the position of the valve 54 into a signal which is supplied as a measured value signal to a position controller 96. At the same time, a corresponding position set-value signal is supplied via a line 97. The cascade control prevents the water injection valve 54 from closing completely for periods longer than short control oscillations, namely, by increasing the amount of water supplied to the separator through the line 26 as soon as the measured value of the valve displacement becomes smaller than the set-value. This feature allows the temperature to increase at the inlet, and therefore, also at the outlet, of the heating surfaces connected downstream of the evaporator.

Referring to FIG. 11, the steam to be condensed or cooled can alternatively be metered into a chamber in which the working medium to be heated is sprayed, instead of having a part of the working medium to be heated being metered to the steam to be cooled as described above. To this end, a mixer 100 is provided in addition to the separator 8 and the entire working medium flow supplied through a line 101 from the economizer 5 is sprayed by suitable means into the mixer 100. In order to supply a matching amount of the working medium from the evaporator 7 to the mixer 100, a branch line 102 with a valve 103 is branched off the line extending from the evaporator 7 to the separator 8. The valve 103 is controlled by the controller 40 in an analogous manner as the regulating valve 24 referred to above. The lower ends of the separator 8 and the mixing vessel 100 are connected by means of a line 104 from which a branch line 105 extends to the circulating pump 6. Owing to the different pressure drops in the feeds extending to the separator 8 and to the mixing vessel 100, the water levels in the two vessels will not be identical. However, this may be compensated easily by appropriate dimensioning of the vessels.

The advantage of the system is that it is possible to dispense with the restrictor 32 as described above and therefore less headroom is required for the vessels 8 and 100.

Referring to FIg. 12, the steam generator is similar to that of FIG. 11 as indicated; however, the mixing vessel 100 is situated within the separator 8.

What is claimed is:

1. A forced circulation steam generator comprising
an evaporator having at least a portion for lining a combustion chamber and an inlet,
a superheater downstream of said evaporator with respect to a flow of working medium passing therethrough,
a return line for returning a part of the working medium leaving said evaporator to said inlet of said evaporator,
a water-steam separator connected to said evaporator to receive the working medium leaving therefrom, said separator having a water outlet connected to said return line and a steam outlet connected to said superheater;
a circulating pump for pumping said part of the working medium in said return line to said evaporator, and
means for heating the working medium entering said evaporator under a heat exchange with steam generated in and leaving said evaporator, said means including means for condensing steam from said evaporator and for supplying the condensed steam to said inlet of said evaporator.

2. A forced circulation steam generator as set forth in claim 1 wherein said means for condensing includes a condenser in said separator having a spray distributor for injecting working medium into said separator prior to delivery through said return line to said evaporator.

3. A forced circulation steam generator as set forth in claim 2 which further comprises an economizer in the flow of working medium upstream of said evaporator, a line connecting said economizer to said evaporator, and a regulating valve in said line communicating said line with said spray distributor.

4. A forced circulation steam generator as set forth in claim 3 which further comprises a servomotor connected to said regulating valve for operating said valve and means for controlling said servomotor in response to the load on the steam generator.

5. A forced circulation steam generator as set forth in claim 1 wherein said means for condensing includes a mixer spaced from said separator and connected in the flow of working medium upstream of said evaporator, and means connecting said mixer to said evaporator to receive a part of the flow of working medium leaving said evaporator.

* * * * *