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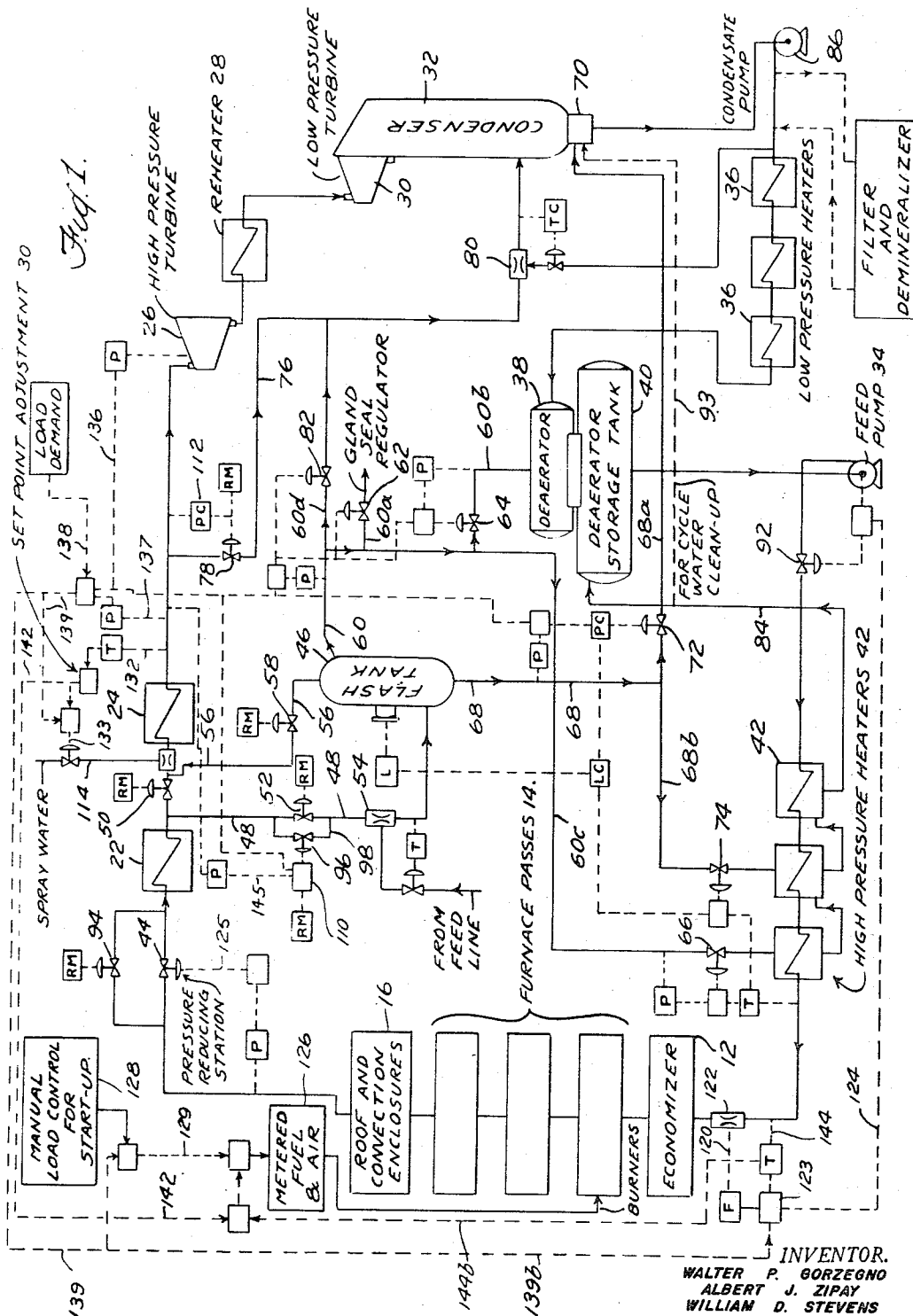
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3,338,055

ONCE-THROUGH VAPOR GENERATOR START-UP SYSTEM

Original Filed May 20, 1963

4 Sheets-Sheet 1



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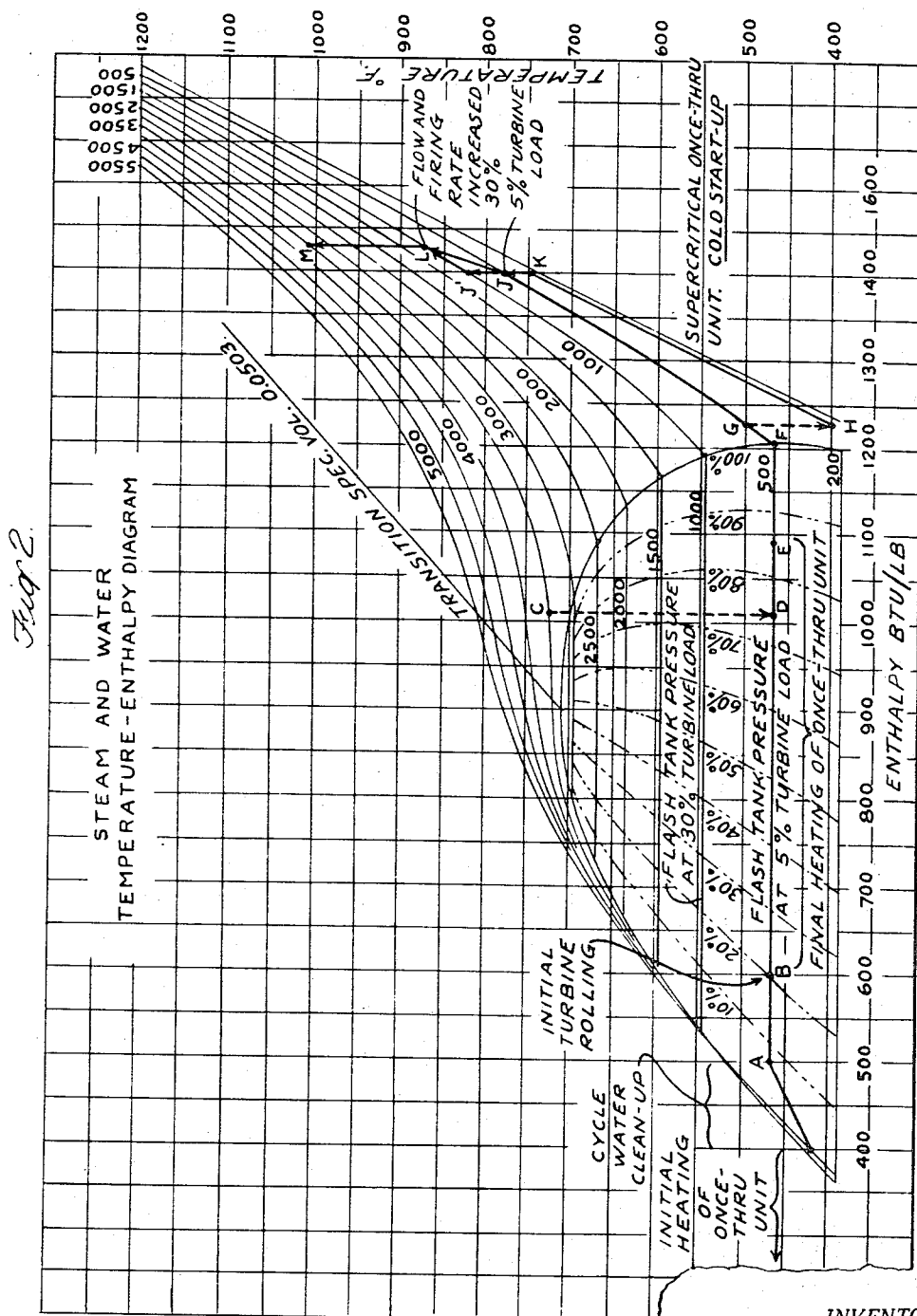
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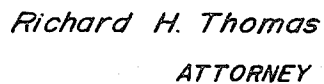


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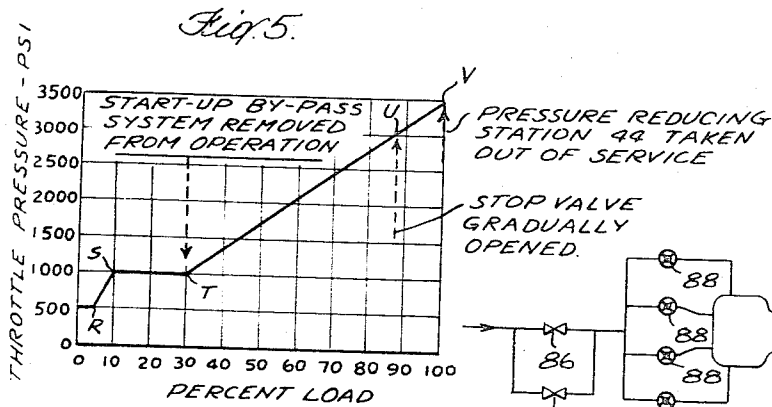
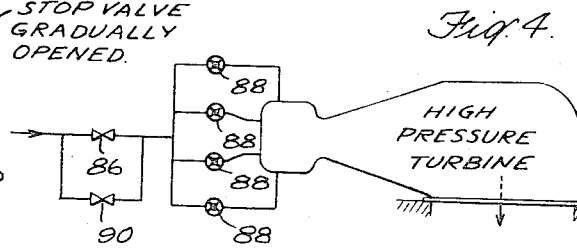
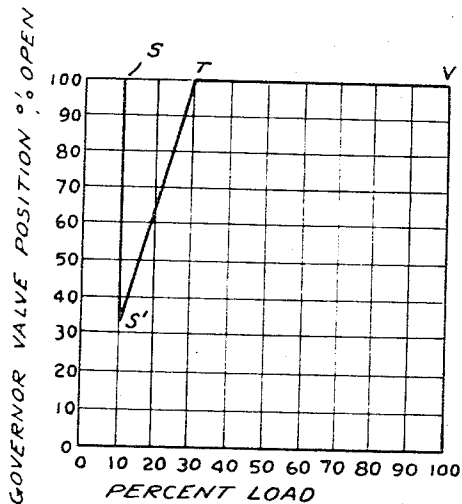
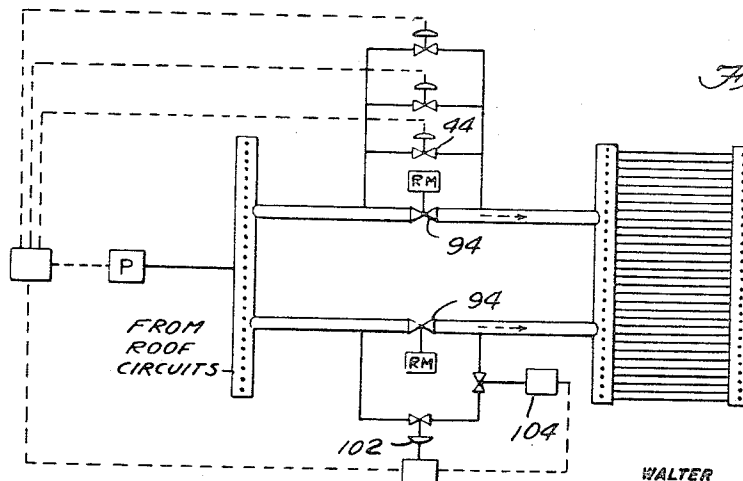
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3,338,055 ONCE-THROUGH VAPOR GENERATOR START-UP SYSTEM

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Original application May 20, 1963, Ser. No. 281,452.
Divided and this application June 8, 1966, Ser. No. 581,120

11 Claims. (Cl. 60—107)

ABSTRACT OF THE DISCLOSURE

A once-through vapor generator-turbine control system in which a load signal controls fuel and air input to the unit, including means for modifying the signal relative to turbine throttle steam temperature and temperature of the feed flow to the unit, the unit further including means to divert at low loads and start-up a variable amount of flow which preheats the feed flow.

This application is a division of application Ser. No. 281,452, filed May 20, 1963.

This invention relates to a start-up and low load system for vapor generators, and in particular to apparatus, methods and controls for starting-up, re-starting and low-load operation of a sub-critical or super-critical vapor generator of the forced flow once-through type.

An important item in the operation of a once-through vapor generator is the by-pass system around the turbine. This system is necessary to protect the furnace walls by providing sufficient circulation of fluid at start-up and low loads. However, the quantity of water or steam flowing through the tubes, to prevent the tubes from overheating and burning out, may exceed the turbine capabilities, and the by-pass system then passes the excess flow around the turbines.

Protection of the high pressure turbine parts also is an important matter in the design of a once-through generator. These parts, which must be capable of handling flows at high temperatures and at very high pressures, for instance, 3500 p.s.i., constitute a large portion of the expense of a turbine unit. Avoiding excessive thermal stresses in the parts is, therefore, a critical consideration.

It is known in by-pass systems to use a flash drum in the turbine by-pass line for the purpose of handling the excess flow and separating the vapor content from the flow for warming and rolling of the turbine. It is also known in start-up systems to use a throttling valve upstream of the turbine for the purpose of furnishing reduced pressure vapor to the turbine. This low pressure throttle vapor allows start-up operation of the turbine with minimum temperature differentials (and therefore minimum thermal stress) in the turbine inlet parts.

The objectives in such prior systems include the avoidance of excessive thermal losses, the reduction of the time required for start-up, and the exercise of care in the protection of turbine parts and vapor generating and superheating sections against thermal shock and high temperature damage. It is found that the application of principles in accordance with the present invention results in substantial improvements in these respects beyond those heretofore achieved.

Accordingly, it is an object of the invention to provide a system in which, during start-up, or low-load operation, the heating surfaces are more fully protected, and in which steam is made available earlier in the start-up period for warming-up and rolling the turbine, and for other uses, to reduce to a minimum the period required

for start-up and, incidental to this, to reduce the heat losses which result from start-up and low load operation. It is also an object of the invention to supply the turbine with steam at a pressure and temperature which is gradually increased during the start-up period and which reaches full throttle pressure at a load more compatible with the turbine design for maximum protection of turbine parts.

These and other objects are accomplished in accordance with the invention by providing in a once-through unit, which includes a vapor generating section and superheating sections, means upstream of the superheating sections for reducing the fluid pressure so that heat is imparted to the fluid in the superheating sections at a reduced pressure. For a given flue gas flow over a superheating bank, and for a given fluid flow within the bank, for the same thermodynamic entering conditions of flue gas and fluid, the reduction of the fluid pressure greatly increases the amount of heat absorbed by the fluid flowing through the bank.

This scheme provides the necessary vapor for rolling and warming the turbine earlier in the start-up cycle than if heat was imparted to the fluid in the superheating sections or banks at full pressure. In addition, a shorter period for start-up and rolling contributes, in conjunction with suitable heat recovery circuits, towards obtaining a minimum loss of start-up heat input.

Also provided in accordance with the invention is a by-pass system which includes a flash tank disposed between or intermediate superheating surfaces or sections arranged to receive the reduced pressure flow which has been reheated following the reduction in pressure in the upstream superheating section. Also included in the by-pass system are piping and valves for the distribution of the by-pass flow to various areas of a condensate-feedwater system, and to the superheating surface or section downstream of the flash tank. In this latter respect, means are provided for controlling the heat content of the fluid at the outlet of this section. Since the fluid flow is reheated in the upstream surface at a reduced pressure, a greater amount of vapor is flashed in the flash tank earlier in the start-up cycle. Since at least a portion of the vapor is then passed to the superheating section downstream of the flash tank, this portion is supplied to the turbine in a superheated state. In this way the turbine gets a superheated vapor earlier in the start-up period, and, in a manner to be described, a vapor flow at a heat level required for maximum protection of turbine parts.

In operation of the start-up and low load system, in accordance with the invention, the method of operation includes the steps of first establishing the required flow and pressure for cooling the circuitry and then placing the burners in service at a reduced firing rate. The fluid pressure is reduced at an intermediate point in the heating surfaces of the generator, and the circulating fluid is reheated at the reduced pressure. The reheated fluid is flashed to provide a vapor flow, which is further heated, the amount of flow being adjusted to achieve, in the step of further heating, a desired level of heat content.

In this respect, for a cold start, the flow may be adjusted to provide to the turbine throttle, for warming and rolling the turbine, vapor in a slightly superheated state. In a hot re-start, the vapor is superheated to a level, at which, after throttling through a conventional turbine throttle by-pass valve, the temperature of the vapor matches the temperature of the turbine inlet parts.

When the turbine reaches a steady state condition, it is synchronized and loaded, and the load is increased by increasing the flow to the turbine up to full start-up flow established in the high pressure circuitry, and then by increase of the temperature, pressure, and flow through

standard combustion control of the unit up to full load flow.

It will be shown that this sequence of steps results in substantial improvement in avoiding stress during warming and loading of a turbine for either a hot re-start or cold start, and in other improvements.

These and other objects and advantages of the invention will become apparent upon consideration of the specification and accompanying drawings, in which:

FIGURE 1 illustrates schematically an embodiment of a start-up system in accordance with the invention;

FIGURES 2, 3, and 3A are temperature-enthalpy diagrams illustrating operation of typical high temperature convection surfaces for a cold start and a hot re-start, respectively, in accordance with the invention;

FIGURE 4 illustrates a throttle valve and governor valve control arrangement for a high pressure turbine;

FIGURE 5 is a diagram showing operation of a turbine-generator unit in accordance with the invention during a cold start with respect to variation in pressure with percent load at the turbine throttle;

FIGURE 6 is a diagram for the unit of FIG. 5 showing the variation of the turbine governor valve position with load; and

FIGURE 7 is a diagram illustrating a main flow line reducing station in detail in accordance with an embodiment of the invention.

Referring to the embodiment of FIG. 1, there is illustrated schematically a vapor generating and turbine installation which includes in series an economizer 12, furnace passes 14, and roof and convection enclosure passes 16. Also constituting sections of the installation are a primary and/or platen super-heating section 22 and a finishing superheating section 24. During normal operation of the unit, the flow is through the superheating sections and from the outlet of the finishing superheating section 24 to a high pressure turbine 26, the exhaust steam from the turbine being reheated at 28 and passed to a low pressure turbine 30, and from there to a condenser 32. Being of the once-through type, the system is pressurized by a feed pump 34, the feed flowing from the condenser through low pressure heaters 36, deaerator 38, storage tank 40 and high pressure heaters 42 to the economizer 12.

In accordance with the invention, the start-up system includes a pressure reducing station 44 (which may constitute a plurality of valves, although, for the purpose of illustration, only a single valve is shown) in the main flow-path upstream of the primary superheater 22 receiving flow from the roof tubes 16. Also constituting part of the start-up system is a flash tank 46 disposed in a by-pass line 48 leading from the outlet of the primary superheater 22 and by-passing a main flow line valve (or valves) 50 between the superheating sections 22 and 24. A stop-valve 52 in the by-pass line 48 separates the flash tank from the outlet of the primary superheating section 22. Also disposed in the by-pass line 48 is a spray attemperator 54, and a supplemental by-pass line 98 extending around the stop valve 52 between the primary superheater outlet and the flash tank.

The piping and valves for the distribution of the by-pass flow from the flash tank consist of a line 56 extending to the inlet end of the finishing superheater 24, this line containing a valve 58; and a vapor line 60 leading from the vapor space of the flash tank including branches 60a to the turbine gland seal regulator, 60b to the deaerator, and 60c to the high pressure heaters, these lines being opened and closed by valves 62, 64 and 66 respectively. Drain flow from the flash tank is handled by a line 68 having branches 68a and 68b leading to the condenser hot-well 70 and to the high pressure heaters 42, respectively, these lines being opened and closed by valves 72 and 74, respectively.

Also constituting part of the by-pass system is a line 76 leading from the outlet of the finishing superheater 24 to the condenser, by-passing the high pressure and low pres-

sure turbines, a pressure reducing valve 78 in the line, and a spray attemperator 80 immediately upstream of the entrance port for the condenser. The line 76 is joined by a branch line 60d, from the flash tank vapor space through valve 82.

The invention will be better understood by way of example, with reference to FIGS. 2-6, FIGS. 2, 3 and 3a showing steps of operation of a once-through unit in accordance with the invention, and FIGS. 4-6 showing the manner of tie-in of the unit with a turbine. Although in the following example, specific numbers are given with respect to pressures, flow rates, temperatures and the like, it is to be understood that the concepts of the invention are not limited to these specifics.

Initial heating of once-through unit

Initially, in a cold start-up period (FIG. 2), the feed pump is driven to pressurize the unit upstream of the reducing station 44, and the fluid flow necessary for cooling the high pressure circuitry is established through the unit to the primary or platen superheater outlet 22. Since the turbine is incapable of handling this flow, the output from the superheater is fed through stop valve 52 to the flash tank 46, and through the flash tank drain line 68 to the condenser hot well and the second stage of the high pressure heaters. In the example, a flow rate of 30% of full load flow through the unit is required for satisfactory cooling of the high pressure circuitry. The pressure upstream of the reducing station is initially held at 600 p.s.i., and the fluid is throttled through the reducing valves to a pressure of about 100 p.s.i. If the upstream pressure were held at a higher value, the high velocity flow would seriously erode the valves. The lower upstream pressure permits the valves of the reducing station to be opened wider to reduce erosion. When the fluid becomes hotter and more compressible, erosion is less severe and the setting for the valves can then be moved to a higher upstream pressure.

The high pressure heater stages drain, as shown, to the deaerator storage tank 40 through line 84. The condensate from the condenser is pumped by the condensate pump 86 through the low pressure heaters 36 to the deaerator storage tank 40. At this stage, the main flow line valve 50, between the outlet end of the primary or platen superheater 22 and inlet end of the finishing superheater 24, is closed so that no flow passes through this valve. Also closed are the flash tank vapor discharge line valve 58 leading to the inlet end of the finishing superheater and the turbine by-pass line valve 78, between the outlet end of the finishing superheater and the condenser 32. Also at this stage of start-up, the vapor lines from the flash tank are closed by valves 62, 64, 66 and 82.

The burners are put into operation, for instance, at about 15% full load firing rate, controlled so that the furnace exit temperature entering the area of the finishing superheater does not exceed 1200° F., the maximum safe temperature for the tube metal.

When the fluid temperature upstream of the reducing station 44 reaches a predetermined temperature, for instance about 300° F., the reducing valves of the station 44 are adjusted to increase the upstream pressure from 600 p.s.i. to 3650 p.s.i. (This would be a suitable full load pressure for the high pressure circuitry of a supercritical unit, 2550 being that for a subcritical unit.) At this point, the flow remaining at 30%, no significant steam quantity is flashed in the flash tank, but as heating is continued, at constant pressure and flow, steam flashing will take place in the flash tank in increasing quantities, and a level will be established in the flash tank. This will occur prior to point A on FIG. 2.

Also prior to point A, the cycle water is cleaned up. This is effected through line 93 (shown as a dashed line in FIG. 1) leading from the outlet of the high pressure heaters to the condenser hot well. In conventional once-through units, the flash tank drain flow is used for this

purpose. By this aspect of the invention, the temperature drop in the feedwater that would occur, if flash tank drain flow were used, is avoided.

At point A, the flash tank pressure reaches its set point value of about 500 p.s.i., at which it is controlled until, in the start-up cycle, a predetermined turbine load, in this example, a 5% load, is achieved. The valve 58, in the vapor line 56 leading from the flash tank vapor space to the inlet of the finishing superheater 24, is cracked open on remote manual control to furnish warming steam to the main steam line. Also, the valve 78 in the turbine by-pass line 76 is cracked open to allow the warming steam to pass to the condenser 32. As an alternative, any suitable main steam line drain may be provided for this purpose.

All of the flash tank drain flow is still routed through the valves 72 and 74 to the condenser hot well and high pressure heater. Subsequently, some steam is made available for the gland seal regulator and the deaerator. Valves 72 and 74 respond to level control (LC) to hold the desired flash tank liquid level.

Initial turbine rolling

At point B on FIG. 2, about 5% of full load steam flow is available from the flash tank, and a portion is diverted through valve 58 and the finishing superheater for rolling and warming the high pressure turbine, the remainder being disposed of through the by-pass line 60 leading from the vapor space of the flash tank to the high pressure heater 42, and elsewhere.

A typical high pressure steam turbine unit (FIG. 4) will include a main throttle stop valve 86, and a plurality of governor control valves 88 in series with the stop valve. Associated with the stop valve is a relatively small by-pass control valve 90 for control of the flow at low loads.

The warming and rolling flow from the outlet of the flash tank, to the high pressure turbine through the finishing superheating section is superheated slightly in the finishing superheater to point G on FIG. 2. At this stage, the turbine throttle stop valve 86 (FIG. 4) is closed and the by-pass valve 90 controls the flow to the turbine, reducing the pressure of the flow to about 50 p.s.i. (point H, FIG. 2).

Usually, only about 2 to 3% flow is needed for the purpose of warming and rolling the turbine, but if more than 2 to 3% flow is needed through the finishing superheater for control of the enthalpy of the fluid at the turbine throttle, the remainder may be dumped through the turbine by-pass line 76, from the outlet of the finishing superheater, to the condenser. The spray attenuator 80, on temperature control, insures that the temperature of the fluid entering the condenser does not exceed design limits of the condenser.

During the final stages of heating of the generator with flash tank pressure held at 500 p.s.i. (points B to E, FIG. 2), and prior to loading, the initial firing rate is maintained. Furnace exit temperature still is monitored and should not exceed 1200° F. and the 2 to 5% rolling flow for warming the turbine is maintained from the flash tank to the finishing superheater and the high pressure turbine. A steady-state condition for the generator is achieved when the enthalpy is approximately 1050 to 1100 B.t.u./lb. entering the flash tank (point E), at which point the generator circuit components are up to temperature at the flow rate, firing rate, and pressure set in the unit. At this point, firing rate may be decreased to accommodate only turbine warming and cycle losses.

The dotted line between points C and D in FIG. 2 represents the pressure reduction in the main flow line from 3500 p.s.i. to 500 p.s.i. at the reducing station 44. The compressible supercritical fluid at about 730° F. is expanded to a temperature of about 470° F. and a vapor content of about 75%. However, an advantage is obtained in that the transfer of heat to the reduced pressure fluid in the superheating sections is substantially greater.

For instance, for the same flue gas flow over the superheater bank and fluid flow in the bank, and for the same thermodynamic entering conditions of flue gas and fluid, the pressure reduction from 3500 p.s.i. to 500 p.s.i. increases the bank heat transfer efficiency by 54% approximately.

When the enthalpy of the steam at the outlet of the primary section (22, FIG. 1) is slightly superheated (between points F and G), or is at some other predetermined temperature, the main flow line valve 50 between the superheating sections can be opened on remote manual so that the flow is from the primary superheater outlet directly to the finishing superheating section and to the high pressure turbine. The set point for the flash tank still controls or regulates the pressure in the heating surfaces downstream of or following the reducing station 44.

Turbine synchronization and loading

As the heating is continued in the unit at constant pressure and flow, the steam temperature entering the turbine first stage through the turbine stop valve by-pass increases along the constant pressure line from point H to point K, concurrent with an increase in temperature of the throttle steam at 500 p.s.i. (temperature of the stem at the outlet of the finishing superheater) from point G to point J. At point J, the turbine parts reach a state of equilibrium, and the turbine is then synchronized and a load is applied and gradually increased to 5%, depending upon turbine design. This is accomplished by opening the turbine throttle stop valve by-pass 90 to full open position increasing the temperature and pressure of the flow entering the first stage to some intermediate point between points K and J. This is also shown as point R in FIG. 5, which illustrates the variation of turbine load with throttle pressure in a turbine unit in accordance with the invention. The turbine governor valves are wide open. An immediate further increase in turbine load, up to 10% load (finishing superheater outlet conditions at point J'), is achieved by increasing the flash tank set point from 500 p.s.i. to 1000 p.s.i. This is point S in FIGS. 5 and 6, the latter figure showing the relationship between load and governor valve position. At point J', the turbine is placed on governor valve control, by closing down on the governor valves until they assume control, and the throttle stop valve (86, FIG. 4) is opened wide. This is line S-S' on FIG. 6. The load is increased to 30% by opening the governor valves to full open position along the 1000 p.s.i. throttle pressure line, J'-L on FIG. 2, and S (or S')-T on FIGS. 5 and 6. The temperature and pressure of the fluid entering the turbine first stage follows approximately path J-L from 15% to 30% load. It will be recalled that a 30% full load start-up flow had been established through the unit for cooling the furnace circuitry. At 30% turbine load, the start-up system is removed from service (no longer being required for pressure control) and loading the turbine from 30 to 100% load is accomplished by combustion control regulation of throttle pressure and temperature in response to load demand. This is line L-M on FIG. 2 and lines T-V on FIGS. 5 and 6. The combustion control regulation includes means to control the feed pump output and to meter the fuel and air supply, so that the increase in throttle pressure and load with time is linear.

When the throttle steam pressure is approximately 3150 p.s.i. (point U, FIG. 5), the stop valve 94 around the pressure reducing station 44, in FIG. 1, is gradually opened on remote manual control for further increasing the throttle pressure, and the pressure reducing station, which is then at capacity, is taken out of service. At full throttle pressure, the valve 94 is fully open.

The advantages of the invention with respect to a cold start should now be apparent. Particularly evident should be the advantage of improved heat transfer in the super-

heating sections to reduce the start-up period, and correspondingly, heat losses from start-up. At the same time, full pressure is maintained in the furnace circuitry for protection of the heating surfaces.

With respect to turbine operation, the advantages also should be apparent. The normal mode of starting a natural circulation boiler is with reduced pressure steam through a turbine throttle stop valve by-pass to the turbine with governor valves wide open. To design the once-through boiler furnace circuitry to operate at a reduced pressure condition requires extra expense in the design of the furnace circuitry. The introduction of throttling valves in the once-through boiler circuitry (for instance, just downstream of the convection enclosure passes) enables the unit to operate with reduced pressures, at the turbine throttle and still permits the furnace circuitry to be designed to operate only at full pressure.

A further advantage is that the temperature differential across the throttle stop valve is reduced. With 500 lbs. throttle pressure, throttle to the 50 p.s.i. required at the turbine first stage inlet, the temperature differential across the turbine stop valve is only 100°. If the throttle pressure were 3500 p.s.i., this differential would be in the order of magnitude of 400° F.

Another advantage is that by having a stop valve 50 upstream of the turbine throttle stop valve, cold water is prevented from being in physical contact with the upstream side of the throttle stop valve. This is a condition which has long plagued operators of once-through units.

Full arc admission in the turbine unit is where all the turbine governor valves are in the wide-open position. Partial arc admission exists where one or more of the governor valves are open and some are closed. The use of a low pressure throttle steam and throttling through the turbine throttle stop valve by-pass permits the turbine governor valves to remain wide-open for full arc admission during the acceleration, synchronizing and initial loading period. By virtue of this, all the valve bowls are heated uniformly along with the valve chest and first stage shell region.

The turbine is operated on governor valve control (this corresponds to partial arc admission) only during the intermediate loading period from 10% to 30% load. Operation at partial arc admission is thus limited to a very narrow load range which would, in normal practice, be passed through rapidly. From 30% to 100% load, the turbine is again loaded using full arc admission, by combustion control regulation of throttle steam in response to load demand to give required turbine load.

FIG. 7 illustrates, as an embodiment of the invention, a main flow line reducing station wherein a plurality of reducing valves are used. It is found that a fewer number of valves is required by using at least one double-port high capacity valve, indicated by the numeral 102. This valve is not a tight shut-off valve and requires an isolation shut-off valve 104 to be operated in conjunction with it. However, the saving is substantial. Whereas three reducing valves may handle the initial 30% minimum flow (items 44), five additional valves could be required to handle the flow increase up to full flow and transfer to the main flow line valves 94. Instead, one double port valve can handle this additional flow. In the arrangement illustrated, the reducing valves operate in sequence, with, in the case of increased flow, the double port valve 102 opening last. However, before the valve 102 can open, the shut-off valve 104 must open. Conversely, the shut-off valve 104 closes after the double port valve closes.

Hot re-start

The invention is also useful for hot re-starts. Referring to FIGS. 1, 3, and 3A, a hot re-start is initiated by establishing a 30% fluid flow through the once-through unit and platen and/or primary superheater outlet. This flow also is through by-pass valve 52 to the flash tank. The flash tank drain flow is routed through valve 72 to the

condenser hot well, and the flash tank steam flow is routed to the gland seal regulator through valve 62, the deaerator through valve 64, the high pressure heater through valve 66, and to the condenser through line 60d and valve 82.

5 The pressure reducing station 44 maintains 3500 p.s.i. fluid pressure in the furnace circuitry, and the flash tank set point is 500 p.s.i. The stop valves 94 and 50 in the main flow line upstream of and intermediate of the superheating sections are closed. Valve 58 in line 56 also is closed.

10 The burners are set at start-up firing rate (for instance 15% full load firing rate), and the main line valve 50 between the superheating sections is opened on manual control. At this moment, point C on FIG. 3 represents the state of the fluid entering the reducing station, and the location of this point or the enthalpy of the fluid entering the reducing station may vary depending on the period of shut-down of the generator. The line C-D represents the pressure reduction through the reducing station so that point D will correspondingly vary depending on the period of shut-down. Line D-E represents the enthalpy pick-up in the primary and/or platen superheater, and E-F-G the enthalpy pick-up in the finishing superheater. The point G can be arranged to be higher or lower along the 15 500 p.s.i. constant pressure line (i.e., the fluid will have a higher or lower heat or enthalpy content) depending on the rate of flow through the finishing superheater and the amount of flow diverted through the flash tank. Valve 58 between the flash tank vapor space and the finishing superheater remains closed. Also during this initial firing period, the turbine by-pass valve 78 between the outlet of the finishing superheater and the condenser is opened to pass the flow through the finishing superheater to the condenser.

20 When the steam conditions at the outlet of the finishing superheater, point G, match the temperature of the turbine inlet parts, point H (the steam conditions entering the turbine first stage), the turbine by-pass steam flow is switched to the high pressure turbine by manually closing the by-pass valve 78 for initial rolling. The pressure for point H is about 50 p.s.i. by virtue of the reduction in pressure through the throttle stop valve by-pass 90.

25 When the turbine is synchronized, a load is applied, first by increasing the flash tank set point to 1000 p.s.i. (point K). Line G-K represents the desired change in steam conditions at the finishing superheater outlet when approximately 15% load is applied. Line H-J represents the change in steam conditions entering the turbine first stage when the load is applied. Following this, the load is increased to about 30% of full load by operation of the governor valves as with a cold start, closing the governor valves until control is assumed and then opening them to full open position at the throttle pressure of 1000 p.s.i. This increase in first stage entering pressure and corresponding increase in turbine load is represented by line J-K on FIGS. 3 and 3A. At point K, the stop valve 52 in the by-pass line leading to the flash tank is closed by remote manual means removing the start-up by-pass system from service. From this point on, aspects of the cold start are repeated to full load, point M.

30 The simplicity and suitability of the system for a hot re-start should be evident.

35 In providing means for reheating the throttle steam prior to admission to the turbine, an operator is enabled to better match first stage inlet steam temperature with the temperature of turbine inlet parts (point H). If full throttling is undertaken through the turbine throttle valve to point H, the temperature of the incoming fluid would fall below that of the turbine inlet parts, or at least it would make it difficult to match the temperature with turbine inlet parts. By obtaining the exact enthalpy required at the outlet of the finishing superheater in a fluid already throttled to a low pressure, the temperature of the fluid can be reduced to that of the turbine parts by the turbine throttle valve by-pass.

In addition, it is evident from FIG. 3 that the start-up can easily and efficiently be effected from any heat level or enthalpy level of the circulating fluid. For a quick start, a minimum degree of manual control is required, and the load is smoothly applied to the turbine for maximum protection of turbine parts. Other advantages of the cold start are also realized.

For instance, the invention has the same advantage as that in the initial start-up period in that all of the valve bowls are heated evenly, all being wide open during most of the start-up period. This enables a closer approach to ideal starting and loading of the turbine than heretofore achieved, and from this, a reduced turbine maintenance cost.

Low load

The invention is also useful for low load control, which may be affected either by variable pressure and temperature operation at the turbine throttle, or by turbine governor valve control at constant pressure and temperature.

In connection with the latter, a minimum flow, for instance a 30% flow, is required in the high pressure furnace circuitry for cooling the circuitry. In reducing the turbine load, through turbine governor valve control of the flow no problem is experienced as the flow is decreased to this minimum flow by combustion control regulation of the feed pump and firing rate. The flow is in the main flow path valve 94, FIG. 1, by-passing the pressure reducing station 44, so that full throttle pressure is maintained at the turbine inlet.

Below 30% flow to the turbine, the start-up system is advantageously used to dispose of the excess flow at the primary superheater outlet. However, for this purpose, the pressure reducing valve 96 in line 98, by-passing the stop valve 52 between the primary superheater outlet and the flash tank, handles the flow, holding full pressure at the turbine throttle. The pressure set point for the reducing valve 96 will not respond at 30% load and below, as indicated by the turbine first stage outlet pressure load signal, represented by dotted lines leading from the turbine to control box 110 for valve 96. If the capacity of the reducing valve 96 is exceeded, i.e., more flow must be diverted than the valve is capable of handling, the additional flow is passed through valve 78 to the condenser. This valve also is under set point regulation in response to the pressure (pressure controller 112) at the finishing superheater outlet. Subsequent manual adjustment of by-pass valve 52 may also be employed to increase the flow to the start-up by-pass system.

At very low loads, or on sudden changes in load, the fluid temperature at the primary superheater outlet may be too high, and can be reduced by means of spray atomizers 114 in response to an anticipatory turbine load signal (first stage turbine outlet pressure) and to the main steam temperature set point as compared to the measured temperature.

The role of variable pressure and temperature operation in the system is somewhat different. If prolonged operation at loads below the minimum required, for instance 30%, is desired with the start-up by-pass system in service, the pressure reducing station 44 is adjusted to hold upstream set pressure required for the supercritical or subcritical unit (for instance 3650 p.s.i. or 2550 p.s.i. respectively), and the by-pass valve 94 is slowly closed on remote manual control. When the pressure downstream of the reducing station approaches flash tank pressure, the stop valve 52 is fully opened on remote manual control and the turbine governor valve positioner will open to hold constant load at the lower throttle pressure.

The controls

Controls for the generator-turbine unit have been mentioned in the specification in greater or lesser detail.

In the cold start-up sequence, the feed pump (34, FIG. 1) is on flow control to hold the minimum flow

(for instance, 30% of full load flow). This is in response to a signal 120 from flow orifice 122 acting through controller 123 and signal 124. The pressure reducing valve 44 is on pressure control (signal 125) to hold the set point pressure upstream of the valve. The flash tank sub-loop valves and other start-up system valves are set as described in the foregoing description on the cold start-up sequence, holding the desired flash tank pressure. With the pressure reducing station in service, the superheater outlet pressure will follow the flash tank pressure set point. The metered fuel and air control (item 126) for the burners (the fuel and air input) is initially on manual adjustment (item 128 and signal 129).

When turbine rolling and warming steam is available from the flash tank (point B, FIG. 2), the desired steam temperature for the high pressure turbine (point G, FIG. 2) is achieved by manual adjustment of the firing rate and control of the flow through the finishing superheater (excess flow passing through the throttle stop valve by-pass valve 78). It will be recalled that the throttle stop valve (90, FIG. 4) admits a 2-3% flow to the turbine first stage. Manual control of the turbine by-pass valve (78, FIG. 1) passes a greater or lesser flow through the finishing superheater for control of the temperature at the finishing superheater outlet.

Up to loading the turbine (point J, FIG. 2), the temperature of the fluid entering the turbine is gradually increased at constant flow to the turbine and constant pressure (500 p.s.i.) from points G to J, the fuel and air input to the burners being subject to minor manual adjustment from the initially set firing rate. As previously indicated, after a steady-state condition for the generator is achieved (point E), the additional heat supplied is used for turbine warming and cycle losses.

At point J, the turbine parts reach a state of equilibrium, and the turbine is synchronized and loaded up to, for example, 5% of full load. This is accomplished by closing the turbine by-pass valve 78, and the full flow through the finishing superheater, for instance a 5% flow, is fed to the turbine. Since the by-pass control (valve 78) is no longer used for temperature control, the temperature at the finishing superheater outlet is controlled and held at a set point (point J, FIG. 2) by means of a temperature controller (130, FIG. 1). To the controller is fed a temperature signal 132, the signal 142 from the temperature controller 130, compared to anticipatory load signal 139, being transmitted by signal 133 to the emergency spray atomizers 114 located immediately upstream of the finishing superheater. It will be recalled that of the 30% start-up flow, about 5% only, at this stage, is passed through the finishing superheater to the turbine, the remainder being disposed of in the flash tank by-pass system.

During this period of loading to 5%, it is envisioned that although the fuel firing rate and air input responds primarily to manual control 128, it will adjust for feed and main steam temperature to a limited extent. For instance, a main steam temperature 132 below the set point will increase the firing rate (signal 142), whereas above the set point, with full atomization, it will decrease the firing rate. The feed water temperature (signal 144b) also is compared to the main steam temperature (signal 142). This combined signal then modifies the manual control signal (128-129) for control of the fuel and air supply. The purpose of the feed water temperature signal is, that when the by-pass system is used, with flow to the high pressure heaters (for conserving heat), the feed water takes an initial jump in temperature. This must be compensated for by reducing the fuel and air

input.

At a 5% load and above, control of the fuel and air input can be changed from manual to automatic by providing, for automatic control, a load demand signal 138, for instance a 5% demand signal which represents the load applied to the turbine. This signal is compared with

turbine first stage outlet pressure 136 and throttle pressure 137 to provide a load signal 139. The load signal 139 is then modified in a true control sense with the main steam temperature signal 142 and feed temperature signal 144b as described above, the resultant signal controlling the fuel and air input.

To achieve a load increase above 5%, the flash tank pressure set point may be immediately adjusted to 1000 p.s.i. from 500 p.s.i., increasing the load to 10%. Control is then shifted from the throttle valve to the governor valve, and the load is increased at constant pressure to 30% (point L) by moving the governor valve position to full open. This increases the flow to the turbine from 10% to 30%, and at 30% flow the flash tank start-up system can be removed from service, being no longer required for handling an excess flow. In this range 10% to 30% load, the temperature compensated load signal 139 controls the firing rate.

Following removal of the start-up system from service, the feed pump 34 responds also to the load demand signal 139b, and the heat input, flow rate, and pressure increase, preferably along the constant enthalpy line L-M of FIG. 2, programmed in a suitable manner with the load signal 139 and temperature signal 142. As indicated above, the pressure reducing valve (or valves) 44 under pressure control 125 opens wider to accommodate the increased flow while maintaining the desired upstream pressure.

For low load operation of the turbine, at constant throttle pressure, the flow is initially reduced (down to the minimum 30% flow) by combustion control regulation of pumping and firing rate (load demand signal 138 to the feed pump 34 and to metered fuel and air system 126). At and below 30% flow, the turbine first stage load measuring signal 136 fed to control 110 for valve 96, will permit this valve to respond to set point pressure control 145. Other aspects of low load control should be apparent from the earlier description on low load operation, and the above description covering control of the fuel and air input or firing rate.

It should be noted that the reason during start-up for increasing the flash tank pressure from 500 to 1000 p.s.i. stems from a desire to obtain maximum load within the capacity of the throttle stop valve by-pass 90, FIG. 4, this capacity being reached at 5% load, for 500 p.s.i. throttle pressure, or 10% load for 1000 p.s.i. throttle pressure. Governor valve control could be instituted at 5% load, but the valves would then have to assume control at a 15% open position instead of 30% open position.

Although the invention has been described with respect to specific embodiments, many other variations within the spirit and scope in the invention as defined in the following claims will be apparent to those skilled in the art.

What is claimed is:

1. A method for starting-up of low load operation of a once-through vapor generator-turbine unit which includes the steps of establishing a minimum flow at operating pressure in the high temperature circuitry of the generator for cooling the circuitry; reducing the pressure of the flow upstream of convection heating surfaces of the generator whereby the flow is reheated at a reduced pressure; passing a portion of said reduced pressure flow to the turbine to satisfy load requirements on the turbine and passing the remainder of the flow at least in part in heat exchange with the feed to the generator; increasing or decreasing the flow to the turbine in response to load requirements; providing said generator with a signal which in a first instance represents a manual set point for load and in a second instance represents a required load demand as compared to turbine first stage pressure and turbine throttle pressure; modifying said signal to a limited extent in the first instance and in a true control sense in the second instance with a second signal which is a func-

tion of the feed temperature and the temperature of the flow at the turbine throttle; and causing said modified signal to control the heat input to the generator.

2. A method for starting-up a once-through vapor generator-turbine unit comprising the steps of establishing in the high temperature circuitry in the generator a flow rate at operating pressure for the generator equal to the minimum flow required for cooling the circuitry; reducing the pressure of the flow upstream of convection heating surfaces of the generator whereby the flow is reheated at a reduced pressure; in one period of start-up passing a portion of said reduced pressure flow to the turbine to satisfy initial load requirements on the turbine and passing the remainder of the flow at least in part in heat exchange with the feed to the generator; providing said generator with a signal which represents a manual set point for load, and modifying said signal to a limited extent with a second signal which is a function of the feed temperature and the temperature of the flow at the turbine throttle; in a next period of start-up increasing the flow to the turbine at a constant pressure in response to an increase in load until said minimum flow in full is fed to the turbine, providing said generator with an automatic signal which represents load demand as compared to turbine first stage pressure and turbine throttle pressure, and modifying said automatic signal in a true control sense with said second signal; and causing said modified signals to control the heat input to the generator, said second signal compensating for the reduction in heat transfer to the feed resulting from the increase in flow to the turbine.

3. A method for starting up a once-through vapor generator-turbine unit comprising the steps of establishing in the high temperature circuitry in the generator a flow rate at operating pressure equal to the minimum flow required for cooling the circuitry; reducing the pressure of the flow upstream of convection heating surfaces of the generator whereby the flow is reheated at a reduced pressure; in one period of start-up passing a portion of said reduced pressure flow to the turbine for initial loading of the turbine and passing the remainder of the flow at least in part in heat exchange with the feed to the generator; in a next period of start-up increasing the flow to the turbine in response to an increase in load until said minimum flow in full is fed to the turbine, providing said generator with a load demand signal to control the heat input to the generator, measuring the feed water temperature and the temperature of the heated fluid at the turbine throttle, comparing said temperatures, and using said comparison to modify said load demand signal; said method further including the step of controlling the temperature of the fluid at the turbine throttle in said one period by controlling the flow in final heating surfaces of the generator and by-passing excess flow around the turbine and in said next period by attemperating the fluid with the introduction of a cooler fluid therein.

4. In a method for controlling a once-through vapor generator-turbine unit wherein during start-up and at low loads a portion of the heated fluid flow may be used to preheat the feed flow, the steps comprising comparing a load demand signal with the turbine first stage outlet pressure and the turbine throttle pressure to produce a load signal, using said load signal to control the fuel and air input to the burners, and modifying said load signal relative to the turbine throttle steam temperature and the temperature of said feed flow to adjust the fuel and air input compensating for a change in the temperatures of the feed flow.

5. In a method in accordance with claim 4 further including the steps of adjusting the pumping rate for said feed flow by means of a flow rate signal, correcting said flow rate signal relative to the temperature of the feed flow, and comparing said temperature corrected flow rate signal with said load signal.

6. In a method in accordance with claim 5 wherein

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said flow rate signal is provided with a minimum flow setting to prevent the feed flow from decreasing below a minimum value.

7. A control system for a once-through vapor generator-turbine unit which includes means for diverting a portion of the heated fluid flow to preheat the feed flow at low loads and start-up, comprising means for comparing a load demand signal with the turbine first stage outlet pressure and the turbine throttle pressure to produce a load signal, means for using said load signal to control the fuel and air input to the unit, and means for modifying said load signal relative to the turbine throttle steam temperature and the temperature of the feed flow to adjust the fuel and air input compensating for a change in temperature of the feed flow.

8. A control system in accordance with claim 7 further including means immediately upstream of the vapor generator-turbine unit for detecting the rate of flow of feed thereto and providing a flow signal proportional to the rate of flow, means responsive to said flow signal for adjusting the pumping rate for said feed flow, means for correcting said flow signal relative to the temperature of

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the feed flow, and means for comparing said temperature corrected flow signal with said load signal.

9. A control system in accordance with claim 8 wherein said means for adjusting the pumping rate for the feed flow is provided with a minimum flow setting to prevent the feed flow from decreasing below a minimum value.

10. A control system in accordance with claim 7 further including spray attemperator means arranged to introduce a cooling fluid into the heated fluid flow, and means to control said spray attemperator means to maintain a set-point fluid temperature at the turbine throttle.

11. A control system in accordance with claim 7 wherein said means for diverting a portion of the heated fluid flow includes means for reducing the pressure of the diverted fluid, further including control means by which said pressure reducing means maintains a set-point fluid pressure at the turbine throttle.

No references cited.

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