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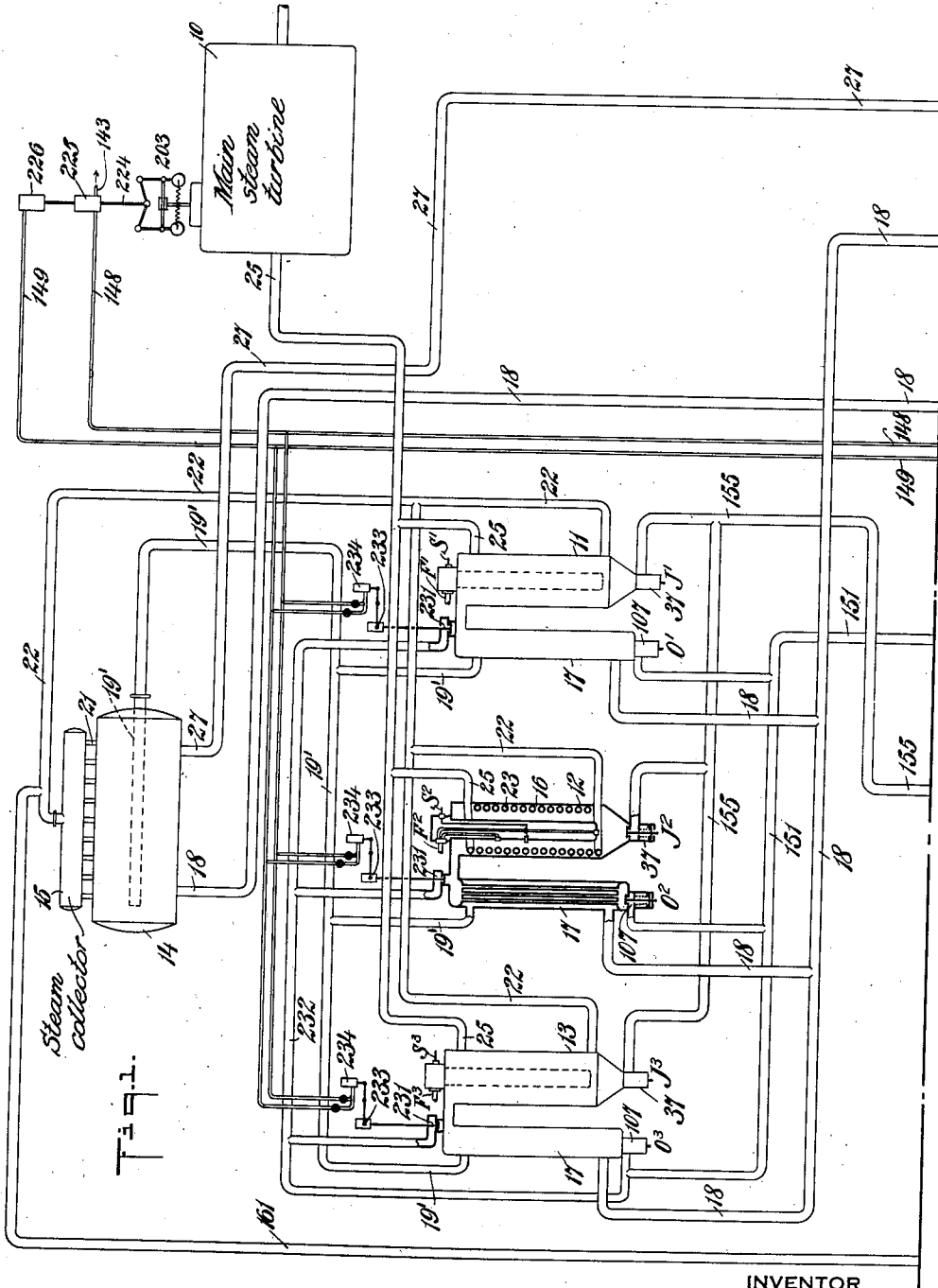
W. G. NOACK

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CONSTRUCTION AND OPERATION OF STEAM GENERATORS

Filed July 1, 1929

5 Sheets-Sheet 1



INVENTOR
W. G. NOACK
BY
Cromwell, Greist & Warden,
ATTORNEYS

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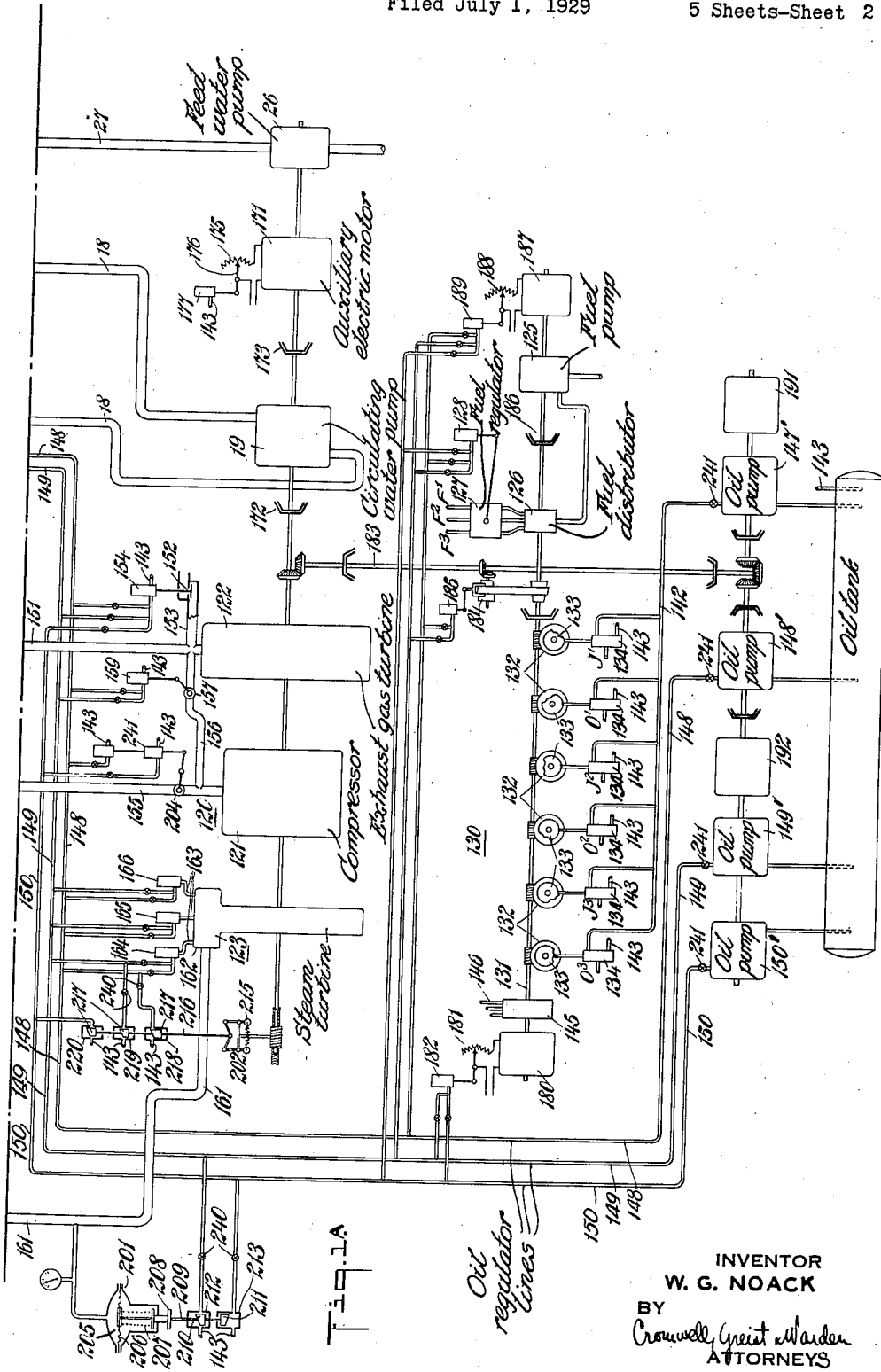
W. G. NOACK

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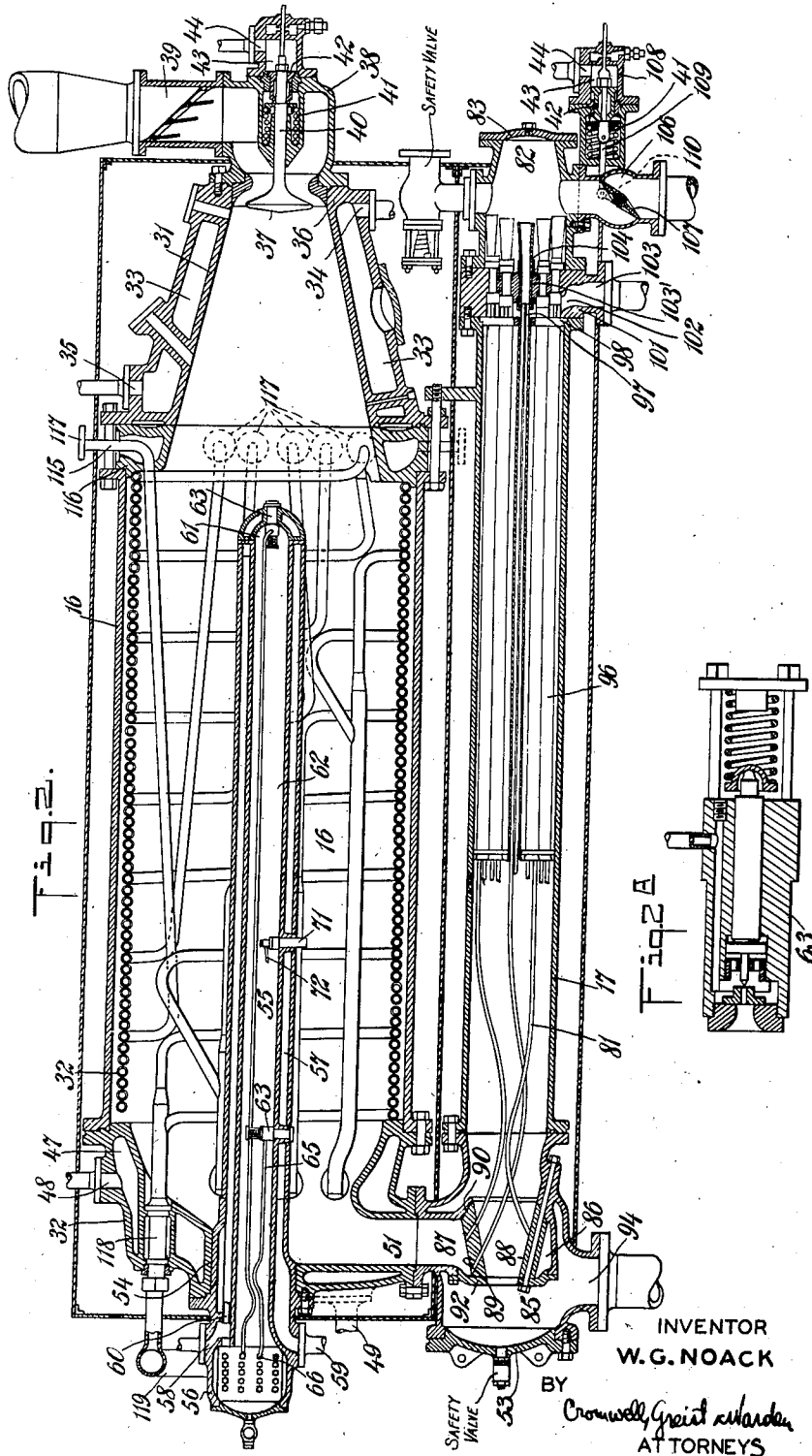
W. G. NOACK

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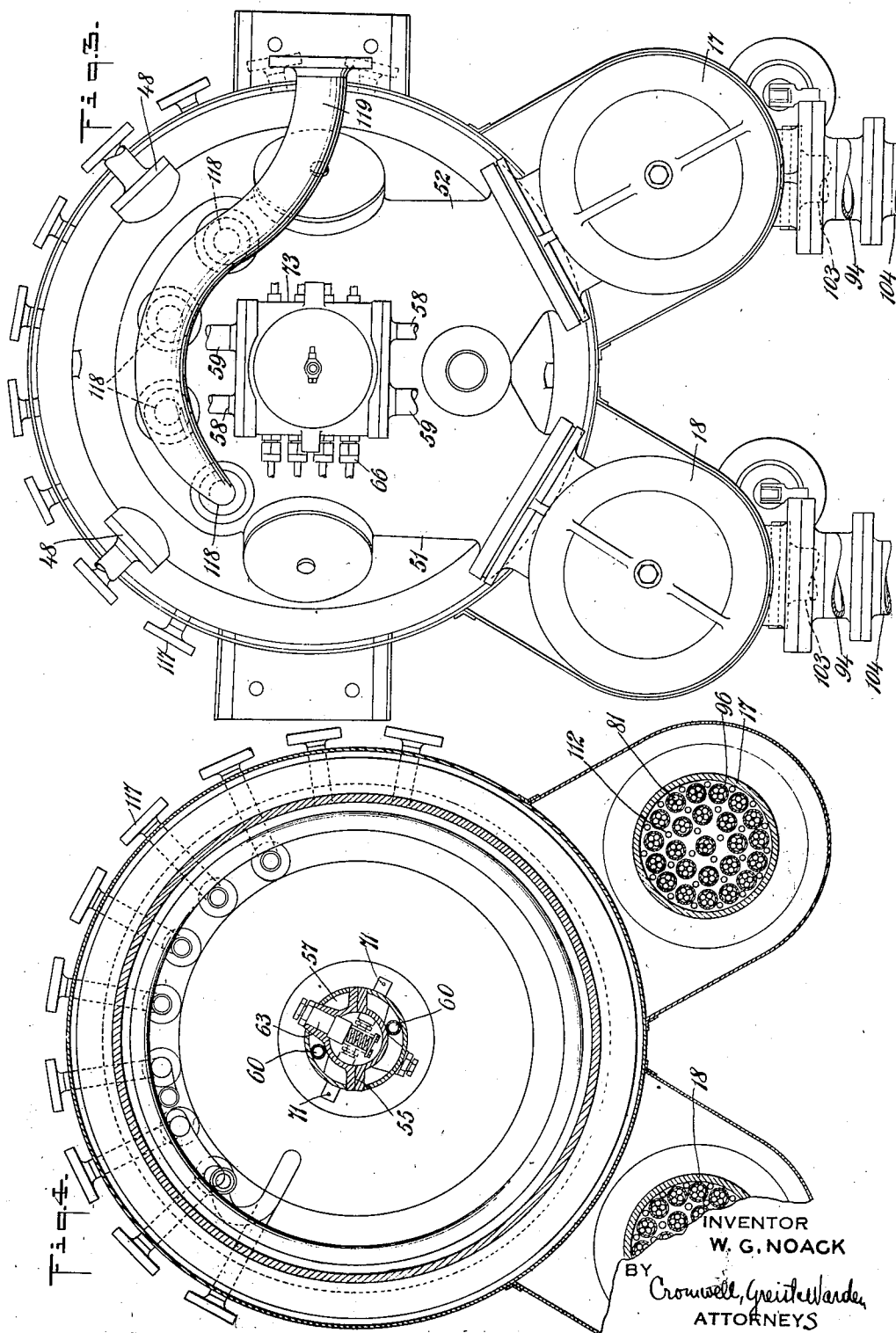
W. G. NOACK

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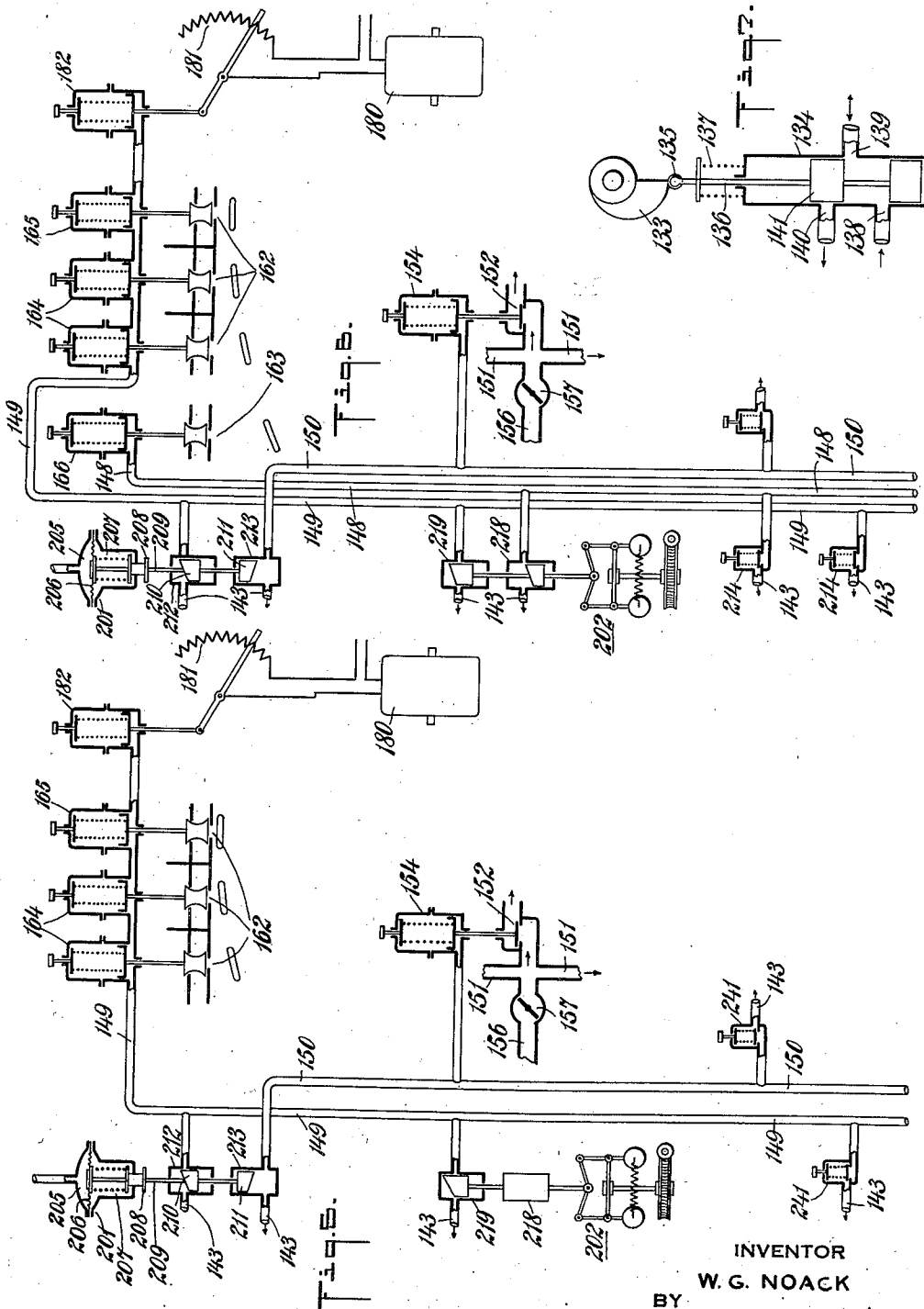
W. G. NOACK

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INVENTOR
W. G. NOACK
BY
Cromwell, Geist & Warden
ATTORNEYS

UNITED STATES PATENT OFFICE

1,993,746

CONSTRUCTION AND OPERATION OF
STEAM GENERATORS

Walter Gustav Noack, Baden, Switzerland, assignor to Aktiengesellschaft Brown Boveri & Cie., Baden, Switzerland, a joint-stock company of Switzerland

Application July 1, 1929, Serial 375,138
In Germany July 4, 1928

20 Claims. (Cl. 122—24)

These improvements are continuations and further developments of the inventions constituting the subject matter of my copending applications Serial No. 333,453, filed January 18, 1929, 343,745 and 343,746, both filed on March 1, 1929, the latter of which have matured into Patents No. 1,948,535 and 1,948,536, respectively, both issued on February 27, 1934.

This invention relates to the operation and construction of steam generators and it has among its objects the provision of a novel steam generating system in which a combustible charge is burned in a pressure-proof combustion chamber under pressure, and the pressure in the chamber is utilized to discharge the gases of combustion through a set of heat exchange ducts at a high velocity near sound velocity or above to transfer the major part of the heat in the gases to a steam generating fluid surrounding the ducts, and improved control and operating methods therefor.

The invention is particularly useful in steam power plants and makes it possible to greatly reduce the space and cost requirements of such plants, as well as to secure better control of their operation. The broad features of the invention have, however, many other applications.

The features and objects of the invention will be best understood from the following description thereof, reference being had to the accompanying drawings wherein,

Figs. 1 and 1A are diagrammatic views of a steam power plant embodying the invention;

Fig. 2 is a vertical, sectional view through the novel combustion and heat exchanger unit of my invention;

Fig. 3 is a top horizontal view of the combustion and heat exchanger unit of Fig. 2;

Fig. 4 is a horizontal sectional view through the combustion and heat exchanger unit of Fig. 2, along line 4—4;

Figs. 5 and 6 are detail diagrammatic views of some of the elements of the control system shown in Fig. 1, arranged to carry out regulating processes of my invention; and,

Fig. 7 is a detail diagrammatic view of the cam shaft valve control unit of the system of Fig. 1.

In Figs. 1 and 1A there is shown diagrammatically, a steam power plant embodying the novel combustion and heat exchange apparatus of my invention and arranged to constitute a novel heat power system adapted to be operated in accordance with any one of several regulating methods and systems devised by me.

In the exemplification shown in the drawings,

a main steam turbine 10 which may serve any of its usual purposes, for instance, for driving an electric generator or for ship propulsion, is arranged to be supplied by steam from a steam plant comprising a battery of steam generating units 11, 12, 13, a steam separator drum 14, a steam collector 15, and associated auxiliary and control apparatus referred to hereinafter. Each of the generating units 11, 12, 13 comprises a combustion chamber 16 and two heat exchangers 17 and 17'.

Water from the collector drum 14 is supplied through supply pipes 18 by means of a circulating water pump 19 to individual heat exchangers 17 of the several boilers. In passing through the heat exchangers, part of the water is evaporized, and water with the steam commingled with it is then returned through the pipes 19' to the collector drum 14 where the steam separates and collects in the collecting vessel 15 mounted on the top of the collector drum 14, and connected thereto by a system of ducts 21. The wet steam from the collecting vessel 15 is then conveyed through the steam supply pipes 22 to superheaters 23 mounted in the interior of the individual combustion chambers 16 of the boilers. The superheated steam from the superheaters 23 is then conveyed through turbine supply pipes 25 to the main turbine where it is utilized. Fresh feed water is supplied into the collector drum 14 by means of a feed water pump 26 through feed water supply pipes 27.

In accordance with the principles of the present invention, the steam boilers 11 to 13 are of a special construction. In distinction from the prior art constructions where the fuel was subjected to the combustion at atmospheric pressure, the boilers of my invention operate on the principle of quick explosion-like combustions. To this end, the combustible mixture is delivered into the combustion chamber at increased pressure and the combustion within the chamber is arranged to take place so as to develop a high pressure, as in an explosion engine. Combustible mixtures burn or explode much quicker under development of high temperatures in closed vessels under increase of pressure, than at atmospheric pressure in open spaces. Thus, for instance, relatively large combustion spaces are required for open fire chambers of prior art steam boilers in order to give the combustible mixture the required time for complete combustion and to prevent the escape of incompletely burned parts of the mixture. On the other hand, fuel such as gas, oil, coal powder and simi-

lar finely distributed materials, will burn or explode in closed combustion chambers in a fraction of a second. The size of the combustion chamber space in the latter case depends principally on the quantity and the pressure of the combustible mixture used for each combustion cycle.

In the steam boilers of my invention, this principle of quick combustion under increase of pressure is made use of and the fuel such as gas, oil or carbon dust is exploded under increased pressure in a closed chamber, the heat released during the combustion being utilized for the generation of steam in heat exchangers.

The heat exchangers of my invention are so designed as to cause the combustion gases from the combustion chamber to pass along the heat exchanger surfaces at a very high velocity of the order of sound velocity, thereby securing a high degree of heat transfer. This transfer of heat at the heat exchanger surfaces depends to a large extent on the density of the combustion gases and their velocity of flow past these exchanger surfaces. By securing a high flow velocity of the order of sound velocity, much smaller heat exchanger surfaces are sufficient for transferring the heat set free by the combustion to the medium that is to be heated, such as boiler water.

The effectiveness of the high combustion gas velocities on the heat transfer becomes pronounced at gas velocities of about 200 meters, and practical steam generators according to my invention are obtained by operating with such or higher average velocities.

A detailed illustration of one of the boiler units 11 to 13 is given in Figs. 2 to 4, which are made in accordance with the drawings of one of the first boilers actually built in accordance with my invention. The combustion chamber 16 of each boiler unit consists of a relatively long cylindrical vessel of heavy sheet metal having at the bottom side an inlet head 31 and at its top side an outlet head 32. The inlet head 31 is of double wall construction and is provided with a hollow space 33 through which water is circulated by passing it into the hollow space at the water inlet opening 34 and passing it out at the water outlet opening 35.

Over a central opening 36 of the inlet head is mounted an inlet valve 37 with an associated inlet valve casing 38 by means of which compressed air, or in general a compressed combustible gas, may be admitted into the combustion chamber through an inlet duct 39. The valve 37 has a stem 40 to which is secured a valve closing spring 41 by means of which the valve is normally brought to closed position. The valve may be opened by means of an actuating piston 42 mounted on the lower end of the stem, the piston 42 moving in a small cylinder 43 to which oil under pressure is arranged to be supplied through an oil inlet 44.

The construction of this valve actuating mechanism may be substantially the same as that used in the standard steam turbine designs of the Brown, Boveri Company, the assignee of the present application, as described in various articles appearing in past numbers of the Brown Boveri Review, published by the said company at Baden, Switzerland. The construction of these valve actuating mechanisms and the oil pressure control and regulating means therefor are also described in the book on steam and gas turbines by A. Stodola, Berlin, 1922.

The space of the cylinder 43 under the oil piston 42 is connected to a source of oil under pres-

sure, and depending on the pressure of the oil, the piston will move the valve to its open position against the closing action of the spring 41. By increasing the oil pressure the valve will be opened to a more or less extent. By reducing the oil pressure the spring 41 will cause the valve to close.

The outlet head 32 enclosing the top of the combustion chamber 16 is likewise of double wall construction and has a hollow space 47 through which water may be circulated by admitting it at the inlet opening 48 and discharging it at the outlet opening 49. The gases from the combustion chamber 16 are discharged through a pair of discharge ducts 51, 52, laterally extending from the outlet head and arranged to be connected to inlet heads 53 of the two heat exchangers 17 and 18 of each of the boiler units. The center part of the combustion chamber outlet head is provided with a large opening 54 in which is mounted a distributor cylinder 55 by means of which the fuel for the combustion mixture and the ignition sparks are supplied to the combustion chamber.

As shown in the drawings, the distributor cylinder 55 has an inlet head 56 mounted on the combustion chamber opening 54, and the distributor cylinder extends downwardly below the same, longitudinally within the combustion chamber. This distributor cylinder 55 is likewise double-walled and is provided with hollow spaces 57 through which water may be circulated, the water being admitted at the inlet opening 58 and discharged at the outlet opening 59. In order to secure better circulation of the water, there are provided special water conduits 60 on the inlet side of the hollow spaces 57 in the distributor cylinder so that the water admitted at the inlet openings 58 shall be conveyed by the water conduits to the lower end 61 of the cylinder.

As seen in Fig. 4, the central distributor cylinder 55 of the combustion chamber is made of two halves so that the spark plugs and the discharge nozzles may be conveniently mounted on each half, together with their fuel supply tubings and the ignition cables, and the two halves then assembled into one unit for mounting in its place within the combustion chamber.

Arranged along the surface of the distributor cylinder 55 are a number of fuel discharge nozzles 63. The boiler shown in Figs. 2 and 4 is constructed for use with oil as a fuel and the discharge nozzles 63 are constructed in a way similar to the nozzles employed in the familiar Diesel engines. In order to secure a uniform combustible mixture, the number of fuel discharge nozzles is made relatively large, and may be 10 or 20 or more, depending on the size of the combustion chamber. The fuel is delivered to the nozzles by means of fuel supply conduits 65 connecting each of the nozzles to one of a series of fuel inlet connections 66 mounted on one of the side walls of the distributor head 56. The combustible mixture is ignited by means of spark plugs 71, a number of which are arranged along the surfaces of the distributor cylinder so as to simultaneously ignite the combustible mixture within the combustion chamber at a number of different points, thereby securing complete and uniform combustion. The ignition current is supplied to the individual spark plugs by means of insulated conductor cables 72 which are led through the central hollow duct 62 of the distributor 55 to a set of ignition plug terminals 73 mounted on the side wall of the distributor head 56.

Each of the two heat exchangers 17 and 18 consists essentially of an elongated cylindrical vessel

of sheet metal, through which extend a large number of gas discharge tubes 81 through which the gases coming from the outlet ducts 51, 52, of the combustion chamber, are discharged and conveyed to the gas outlet space 82 within the outlet head 83 mounted at the bottom side of the heat exchanger.

109 acting upon an arm 110 mounted on the shaft of the rotary valve 107.

As shown in Fig. 2, the oil controlled actuating unit of the inlet valve 37 holds at the particular moment, the inlet valve 37 open, oil under pressure being admitted to the oil cylinder 43 while the actuating unit of the valve 107 is holding the latter closed, the oil pressure having been released and the spring 41 having brought the piston 42 to its lower position.

Through the provision of the nesting pipe sections 96 around the individual groups of gas discharge tubes 81, the water is passed along the heat exchange surfaces of the tubes 81 in thin layers, securing a high rate of steam generation and heat exchange action. In addition to the openings 97 in the bottom wall 98 of the heat exchanger, this wall may have additional openings as shown at 112 in Fig. 4, through which the water entering through the inlet 103 of the water inlet head 101 may pass into the heat exchanger cylinder 17 and therefrom flow to the water outlet opening 94 at the top of the exchanger.

The superheater 23 that is mounted within each of the combustion chambers 16 comprises a series of pipe coils, there being six such pipe coils in the exemplification shown in the drawings. These pipe coils 23 are arranged one above the other along the surface of the combustion chamber, the inlet ends of each coil being connected to inlet pipe sections 115 mounted in a separate section 116 at the top side of the inlet head 31 of the combustion chamber. The outer end of each pipe section is provided with a flange 117 for connection to the wet steam supply pipe 22 leading from the steam collector 15 of the boiler plant, there being in the exemplification six such pipe connections with flanges. The other ends of each coil are connected in pairs to steam outlet tubes 118 extending through and mounted in the outlet head 32 of the combustion chamber, the several tubes terminating into a common superheated steam outlet duct 119 from which the steam is led through the pipe 25 to the main turbine 10.

In order to supply a combustible mixture for the boilers there is provided a compressor aggregate 120 as shown in Fig. 1A. The compressor aggregate comprises a compressor 121 arranged to be driven by a gas turbine 122, and an auxiliary steam turbine 123, the three being shown directly coupled. The compressor 121 may be of any of the familiar turbo compressor constructions. Likewise, the gas turbine 122 and the auxiliary steam turbine may be of the constructions familiar in the art, all these three types of machines being well known in the art and descriptions thereof being given in the aforementioned book of A. Stodola.

The fuel oil for the several combustion chambers is supplied by means of a fuel pump 125, the oil being distributed to the different fuel supply inlets 66 on the distributor heads of the several combustion chambers by means of a fuel distributor 126. This fuel distributor may be arranged so as to permit adjustment of the pressure with which the oil is supplied to the discharge nozzle and to control the admission of the oil to the various discharge nozzles in the sequence of their operation. It may also be arranged so as to permit variation of the number of effective nozzles in each boiler unit. This fuel distributor 126 is preferably driven in unison with the cycle control aggregate referred to here-

The inlet head 53 of each of the heat exchangers forms a water chamber 85 in the interior of which is mounted an annular gas inlet chamber 86 provided on its outer side facing the boiler with a duct 87, communicating with the outlet duct 51 or 52 of the combustion chamber. The inner wall of the annular gas inlet chamber

86 is formed by a frusto-conical terminal member 88 provided with a large number of inlet openings 89 in which the inlet ends of the gas discharge tubes 81 are seated, so as to form a gas-and-water-tight connection. As shown in the drawings, the frusto-conical terminal member 88 has at its lower side a conical seating surface 90 pressed against the cooperating surface of the outer gas inlet chamber wall 86 by means of a cover member 92 tightly clamped to the top of the frusto-conical terminal member 88 and the upper end of the outer wall of the gas inlet chamber 86. The inlet chamber cover 92 is open at its center, and the entire chamber is placed at a distance from the outer wall 53 of the heat exchanger inlet chamber 85, so that the water may freely pass through the central opening of the annular gas inlet chamber and along the outer surfaces thereof to the water outlet opening 94.

In order to facilitate their mounting, and increase the efficiency of the heat exchange, the gas discharge tubes 81 are arranged in groups of four, each group being nested in a pipe section 96 mounted over openings 97 in the bottom wall 98 of the heat exchanger chamber. On the under-side of this bottom wall 98 is mounted the water inlet head 101 of the heat exchanger, forming a water inlet chamber 102 which water may enter through the water inlet opening 103 and pass thence through the openings 97 into the pipe sections 96 and therefrom through the upper portion of the heat exchanger chambers 17, 18 to the water outlet chamber 85 and the water outlet opening 94.

The lower wall of the water inlet chamber 102 is provided with a series of gas discharge ducts 103'. Each group of four gas discharge tubes 81 nested in the pipe section 96, and passing through the opening 97, is connected to the upper end of such gas discharge duct 103'. The lower ends of these discharge ducts 103' terminate in diffuser shaped discharge nozzles 104 so as to secure high flow velocity for the combustion gases, the diffuser nozzles opening into the gas outlet chamber 82 on the bottom side of the heat exchangers. From this gas outlet chamber 82 the gases are discharged through a discharge duct 106 provided with a rotary valve 107, which may be closed or opened by means of an oil controlled actuating unit 108. This actuating unit 108 is of the same construction as that used to operate the inlet valve 37 of the combustion chamber and, like the latter, consists of a piston 42, a piston chamber 43 with an oil inlet opening 44 through which oil may be admitted to the oil cylinder 42 to force the piston to its upper position, and a spring 41 by means of which the piston 42 is pressed to its lower position. The movement of the piston is transmitted to the rotary valve 107 by means of a link

inafter, by means of which the sequence of the combustion cycles in the individual combustion chambers is controlled. There is also shown in the drawings, an additional fuel flow controller 127 adapted to be operated by an oil controlled actuating unit 128, permitting variation of the quantity of fuel supplied to the individual fuel discharge nozzles 63, or the number of active fuel discharge nozzles as the case may be, depending on the system of control used.

With such arrangement, the distributor 126 controls the distribution of the fuel supply to the individual boiler units so that fuel is delivered to the unit that is at the particular moment being charged for combustion, and the fuel flow controller 127 is used to vary the amount of fuel delivered to the individual combustion chambers or the number of nozzles used in each combustion chamber. The amount of fuel delivered to the individual chamber may also be controlled by the speed or displacement of the fuel pump 125, thereby increasing the fuel pressure and the amount of fuel delivered through the discharge nozzles of the several combustion chambers.

In order to control the cyclical combustion process in the several boilers there is further provided a cycle control aggregate 130 comprising a driving shaft 131 from which are driven, as by worm drives 132, a plurality of cams 133. These cams 133 in turn, are arranged to cyclically actuate a series of control valves 134 by means of which the admission and release of the oil to the oil cylinders 43 of the inlet and outlet valves of the several boiler units is controlled.

Such arrangement of control valves is shown diagrammatically with more detail in Fig. 7. The cam 133 cooperates with a cam roller 135 mounted on the end of the valve rod 136 of the valve 134, a spring 137 pressing the valve rod to its upper position. The valve 134 comprises a valve cylinder with a compressed oil inlet opening 138, a compressed oil delivery opening 139, and an oil discharge opening 140. By means of slide members 141 on the valve rod 136, the compressed oil delivery opening 139 may be connected either to the compressed oil inlet opening 138 or to the discharge opening 140, depending on the position of the slide rod 136 as determined by the cam 133. Thus, when the slide rod 136 is pressed downwardly by the cam 133, compressed oil will pass from the inlet opening 138 to the delivery opening 139 and the oil pressure will be transmitted to the piston 42 of the actuating unit to which the respective control valve is connected, thereby moving the piston of the particular actuating unit against the action of its spring 41 to open the particular valve. When the cam 133 permits the cam roller 135 to return to its upper position, the compressed oil supply from the inlet opening 138 is cut off from the opening 139 and the compressed oil supply line is directly connected to the oil discharge opening, thereby releasing the pressure from the oil cylinder 43 of the respective valve actuating unit and permitting the valve to close.

With such arrangement, each inlet and outlet valve of each boiler unit will close and open in accordance with the movement of the respective control valves 134. In the drawings Fig. 1A, there are indicated six such control valves 134, there being two such control valves for each boiler unit, one control valve for the inlet valve into the combustion chamber, and one control valve for the two outlet valves at the end of the two associated heat exchangers respectively.

Each control valve 134 has its compressed oil inlet opening connected to a compressed oil supply pipe 142, and its compressed oil delivery opening 139 connected to the inlet opening 44 of the oil cylinder 43 of the actuating unit which is to be controlled thereby. The compressed oil discharge openings 140 of all of the control valves may be connected to one common oil return line through pipes 143.

On the same shaft 131, Fig. 1A, that drives the cams of the boiler control, valves 37 and 107 shown in Fig. 1A, there is also mounted an ignition device 145 by means of which the ignition current is delivered to the spark plugs of the several boiler units in accordance with the sequence of operation of the same. This ignition device 145 may be of the same construction as that generally employed in connection with gasoline engines, high voltage spark energy being applied to the spark plugs of the individual combustion chambers at the moment when the combustion is to take place therein, in the same way as the spark energy is supplied to the combustion chamber of the engines at the beginning of the power stroke. The ignition current is conveyed to the distributor heads 56 of the several combustion chambers and therethrough to the spark plugs 71 of the same, by means of suitably insulated cables 146. The compressed oil for the oil control valves 134 is delivered to the supply line 142 by means of an oil pump 147. There are provided also, additional oil pumps 148', 149', 150', which have each connected thereto, correspondingly numbered control oil supply lines by means of which the various regulating devices of the system referred to herein after are actuated and controlled.

The gas discharges from the outlet openings 107 of the several boiler units are led through pipes 151 into the gas turbine 122, there being provided a gas outlet 153 in front of the gas turbine to permit the gases to be discharged directly into an exhaust by passing the gas turbine so as to control the speed of the latter. The by-passing of the gases around the gas turbine through the outlet opening 153 may be controlled by an exhaust discharge valve 152, which is arranged to be actuated by an oil controlled actuating unit 154 of the same construction as the actuating unit 108 of the boiler.

Compressed air derived from the compressor is delivered to the inlet valves 37 of the several boiler units through a compressed air supply pipe 155, Fig. 1A. There is also shown a pipe connection 156 between the compressed air supply line 155 and the gas turbine supply line 151, the pipe connection being controlled by a valve 157 arranged to be actuated by an oil controlled actuating unit 159. The steam for operating the auxiliary steam turbine 123 is arranged to be admitted through a pipe 161 branching off the pipe 22 leading to the superheater. The admission of the steam from the pipe 161 to the auxiliary steam turbine 123 is controlled by a number of steam inlet valves 162, 163, each of these valves being provided with an oil controlled actuating unit 164 to 166 like the actuating unit of the boiler valves 37 and 107.

The various operating elements shown in Fig. 1A are so arranged as to permit their connection to form a plurality of different operating systems, and for operation in accordance with a plurality of different operating methods.

The circulating water pump 19 and the feed water pump 26 are adapted to be driven either

directly from the compressor aggregate 120 by direct coupling with the shaft of said aggregate, or by means of a separate electromotor 171. By means of clutches 172, 173, the several units may be either coupled together to be driven in unison; or the circulating pump may be coupled to the compressor aggregate 120 and the feed pump driven separately; or the circulating pump 19 and feed pump 26 driven together, separately from the compressor.

The electric motor 171 may be of any of the well known speed control types and is shown provided with a control rheostat 175 having a control lever 176 which may be operated by an oil controlled actuating unit 177 of the same construction as the actuating unit 108 of the boiler valves. By admitting oil under pressure to the actuating unit 177, the rheostat 175 is cut out, causing the electric motor 171 to run at high speed. By reducing the oil pressure, the rheostat lever 176 is permitted to move upwardly, cutting in resistance into the motor circuit, thereby decreasing its speed. By varying the pressure of the oil admitted to the oil cylinder of the actuating unit 177, the rheostat lever 176 may be maintained in any desired position depending on the type of operation that is desired to be secured.

The cycle control aggregate 130 referred to before may be arranged to be driven either in unison with the compressor aggregate 120 or separately therefrom by means of a separate motor 180. As shown in the drawings, the electromotor 180 is arranged to drive directly the shaft 131 to which are connected the cam driving members of the several boiler valve control units 134, the ignition distributor 145 and the fuel distributor 126, so that by varying the speed of the motor 180, the number of combustion cycles per unit time may be varied, and thereby the steam generation decreased or increased, depending on the control of the speed of the shaft 131. The speed of the shaft 131 may be readily varied by suitably controlling the motor 180 as by means of a rheostat 181 adapted to be controlled by means of an oil controlled actuating unit 182, this control arrangement being of the same type as the control arrangement for the feed pump motor 171.

In addition to the drive of the shaft 131 by a separate electromotor 180, provision is also made for driving the shaft directly from the compressor aggregate, as by means of a shaft 183 geared to the shaft of the compressor aggregate, the shaft 183 driving in turn a variable speed transmission mechanism 184 through which the shaft 131 is driven so as to permit variation of the speed of the shaft 131 against the speed of the compressor aggregate. The variable speed transmission mechanism may be of any of the well known types and is indicated diagrammatically in the form of a conical pulley belt drive, the transmission ratio being adjustable by moving the belt along the pulleys. This may be effected for instance, by an oil controlled actuating unit 185 of the same type as the oil controlled actuating unit used in connection with the control of the electromotors and of the valves referred to before.

The fuel pump 125 may be arranged for direct coupling to the cycle control aggregate 130 as by means of a clutch 186, or may be arranged to be independently driven by an electromotor 187, the speed of the motor being arranged to be varied by a rheostat 188 operated by an oil con-

trolled actuating unit 189 of the type referred to before. The oil pumps 147', 148', 149' 150' for supplying oil under pressure to the several control oil lines 142, 148, 149, 150, are arranged either for drive by independent motors 191, 192, or for direct gearing or drive by a common motor, or for coupling to the shaft 183, for drive through the compressor aggregate. Any one of these drives may be used, depending on the type of control system used.

The various oil controlled actuating units of the several devices arranged to be controlled thereby, are actuated by oil under pressure from oil control lines 142, 148, 149, 150, and the pressure of the oil in these lines is arranged to be regulated by the various regulating devices of the system, namely, the steam pressure regulator 201, the compressor speed regulator 202, the main turbine speed regulator 203 and the compressor regulator 204.

The steam pressure regulator 201 may be of any of the well known devices adapted to be actuated by the pressure of steam, and as shown diagrammatically, consists of a vessel constituting a steam chamber 205 with a flexible diaphragm 206 forming one wall of the chamber. This flexible diaphragm 206 is pressed downwardly by the steam pressure as shown in Fig. 1A, this movement being opposed by a compression spring 207, the pressure of which is adjustable, as by means of an adjusting member 208 threadedly mounted within the casing of the spring 207. The diaphragm 206 has connected thereto, a spindle 209 on which are mounted slides 210, 211, of slide valves 212, 213.

These slide valves 212, 213, serve to control the pressure in the several oil control lines and consist, each, essentially of a small cylindrical vessel having an opening communicating on one side with the oil control line, and on the other side with an oil outflow or return line. Depending on the position of the slide member 210 or 211, of the respective slide valves, a larger or smaller opening will be provided for the outflow of the oil from the control line connected to the respective valves. By moving the slide valve to a position where the opening is small, less oil will be permitted to flow out from the control line and a greater oil pressure will be established in the respective line. By moving the slide member to a position so as to make a large oil outflow opening for the respective oil control line, the pressure of the oil in the respective line will be reduced. In the drawings, I have shown the slide valve 212 of the steam pressure regulator 201 connected to the oil control line 149 and the slide valve 213 connected to the oil control line 150. Through this arrangement, the pressure of the oil in these two lines will depend on the position of the diaphragm 206 of the steam pressure regulator and the oil pressure in these two oil control lines will be regulated in accordance with the steam pressure.

The compressor speed regulator 202 may have the form of any of the well known speed regulators used in steam turbines and similar devices. It is indicated diagrammatically in the drawings in the form of a fly ball governor 215 rotated in accordance with the speed of the compressor aggregate 120. The movement of the balls 215 of the governor is arranged to actuate a spindle 216 on which are mounted slides 217 of a plurality of slide valves 218, 219, 220. The slide valves 218 to 220 are constructed according to the same principle as the slide valves 212, 213 of the steam

pressure governor and serve to regulate the pressure of the oil control lines connected to the respective valves. In the drawings, I have shown the slide valve 218 of the compressor speed regulator 202 connected to the oil control line 148, and the slide valve 219 connected to the oil control line 149.

The main turbine speed governor 203 may likewise be of the fly ball type and provided with a spindle 224 arranged to actuate slide valves 225, 226 for controlling the oil pressure of the oil control lines connected to said slide valves, slide valve 225 being shown connected to the oil control line 148, and slide valve 226 being shown connected to the oil control line 149.

In order to permit additional control of the speed of the compressor aggregate there are also provided additional means for admitting a part of the combustion gases from the several boilers directly to the exhaust line 151 leading to the gas turbine 122. To this end, there are provided on each of the boiler units, by-pass valves 231 by means of which a part of the gases of combustion is diverted into a pipe line 232 leading directly to the exhaust pipe line 151 through which the gas turbine is supplied. By opening the valves 231, a part of the combustion gases from the combustion chambers of the boiler units is led directly to the gas turbine, permitting increase of its speed and thereby increasing the pressure of the compressed air delivered to the combustion chamber. The by-pass valves 231 of the several boiler units are arranged to be actuated by oil controlled actuating units 233 of the same construction as the oil actuating units of the inlet and outlet valves of the boiler units.

The actuating units 233 of the several by-pass valves are operated so as to cause the by-pass valves 231 to open at the appropriate moment after the combustion in the respective boiler unit has taken place, causing a part of the combustion gases to be diverted directly into the exhaust line 151, the bulk of the gases going through the heat exchangers in the manner described before. It may be desirable to leave the by-pass valves 231 closed during one range of operation and to cause them to open only during another range of operation. To this end, there may be provided in connection with each of the actuating units 233, an additional control unit 234, the latter being actuated by the pressure in the oil control line in response to the various regulating actions of the system as explained before. By varying the pressure in the respective oil control lines, the actuating unit 234 may be moved to a position where the oil flow from the control line operating the actuating unit 233 of the respective by-pass valve 231 is cut in or cut off, thereby eliminating the by-passing of a part of the combustion gases, or bringing it into adjustably variable action.

I have also shown in the connections of the various control valves of the oil controlled actuating units and the regulating valves associated therewith, cut-off valves 240, whereby the individual valves may be disconnected from the oil control lines so as to enable the arrangement of the several elements into different control and regulating systems. Each of the oil control lines is also shown provided with a safety choke valve 241 which permits escape of oil if the oil line pressure exceeds a predetermined value, this being the usual construction employed in the familiar oil control regulating systems used in steam turbines and like apparatus.

By using the various elements of the system described above to a more or less extent, it is possible to operate the novel heat power system in a large number of different ways, thereby obtaining a wide range of regulating actions best adapted to the various desired operating conditions and best suited for securing the highest efficiency.

In distinction from the steam boiler systems of the prior art, only a relatively small quantity of water and only a relatively small stored quantity of steam need to be used in my improved heat power system. Because of the relatively small masses of the steam generators there is possible an immediate and direct heat exchange, each change of the quantity of fuel subjected to combustion resulting in a relatively immediate corresponding change in the delivery of the steam. As a rule, the fuel is combined with a quantity of air theoretically necessary for its complete combustion, together with a lesser or larger amount of excess of combustion air. The regulation of this system of boilers requires therefore, in addition to the regulation of the fuel supply, also the regulation of the air supply. With a given number of combustion chambers, there is as a rule a fixed volume of the combustible mixture that may be utilized in the individual units during each charge. The quantity of fuel utilized in the combustion may however be controlled either by delivering it with a variable pressure, or by changing the number of the fillings per unit time, or by cutting in or out one or more of the combustion chambers. I accordingly regulate the system either by varying the number of combustion cycles per unit time, or by varying the compression of the combustion charge delivered to the combustion chambers, or by varying the number of effective combustion chambers. The number of combustion cycles may be readily varied by the cycle aggregate 130 described above, an increase in the speed of the shaft 131 of the cycle aggregate increasing the number of combustion cycles taking place per unit time; and a decrease of the shaft speed reducing the number of combustion cycles per unit time. The decrease of compression of the combustion charge may be varied by changing the speed of the compressor aggregate 120, an increase of the compressor speed increasing the compression, and a decrease of the compressor speed decreasing the compression.

Simultaneously with the control of the charging operations or independently thereof, there may also be controlled the quantity of fuel supplied to the combustion chamber. This may be effected either by varying the speed of the fuel pump, or the displacement of the pump, or the pressure with which the fuel is delivered to the individual fuel discharge nozzles, or by varying the number of effective discharge nozzles utilized in each boiler unit. This control action may be secured by controlling the fuel pump 125 and its associated control devices 126, 127, 128 and 187.

My invention provides for the utilization of any of the foregoing regulating processes by itself, or for their combination to conjointly carry out the different systems of regulation adapted to the particular requirements of the system.

The steam pressure regulator used in my regulating system as described above may be used either in connection with the oil or in general, hydraulic control systems such as are used in large steam turbines, or it may be used in connection with the direct mechanical transmission of its regulating action, and I desire it to be distinctly understood that I do not intend to limit my in-

vention to the details of the several elements of the system described as shown above, but that I contemplate the use of equivalent elements as will suggest themselves to those skilled in the art.

5 In like manner, the control of the inlet and outlet valves of the several boiler units may be effected instead of with the hydraulic arrangement described hereinabove, by means of the mechanical arrangement in the form of a cam shaft or the like, such as shown in my applica-
10 tion Serial Number 333,453. Where an electric motor is used for driving such cam shaft or the cams actuating the control valves of the oil control system of the inlet and outlet valves of the
15 boiler units as described above, the motor may be either of the direct current type or of the alternating current type. Instead of using rheostats for controlling the speed of such motors, any of the other familiar systems for controlling
20 their speed may be utilized. In general, the speed of such motors is considerably above the speed of the cam shaft by means of which the valves are controlled, the number of the combustion cycles in each boiler unit varying as a
25 rule between 40 to 100 per minute, although more or less deviation from this number of cycles may be permitted, depending on the character of the fuel used and the details of construction and operation of the system.

30 Ordinarily, the compressors for delivering the compressed air to the combustion chambers are made of the turbo compressor type although other types of suitable compressors may be used instead. When using turbo or rotary compress-
35 sors, the pressure and the quantity of air delivered increases with the increase of the number of revolutions of the compressor. The highest pressure and the highest delivered quantity of compressed gases are determined by the speed
40 of the compressor. Because of these characteristics, the compressor will, when operated at a given number of revolutions, always deliver approximately the same weight of air into the combustion chambers irrespective of whether the
45 number of the charges or the cycles is made larger or smaller. If the quantity of air that can be taken up by the combustion chambers is so small that there is a danger that the compressor may start to pump, this may be avoided by providing
50 for the discharge of the excess of air to the gas turbine. Such discharge of excess compressed air directly into the gas turbine may be effected, for instance, by the control valve 157 through the actuating unit 159 controlling the
55 by-pass of compressed air through the by-pass duct 156 connecting the compressed air line 155 with the exhaust line 151, the back flow of air into the heat exchangers being suitably prevented, as by a check-valve in line 151.

60 In the preferred construction I contemplate that the exhaust gas turbine 122 be used as a principal source of driving energy for the compressor. This gas turbine is supplied by the exhaust gases which leave the heat exchanger
65 nozzles 104 with a relatively high velocity and contain still a substantial quantity of energy. Ordinarily, no special regulation of the gas turbine is required since the available energy of the exhaust gases has always a relatively definite relation to the energy input necessary for operating
70 the compressor. As a rule therefore, I arrange the system in such way that the amount of exhaust gases supplied to the gas turbine is just sufficient for driving the compressor so as
75 to enable the latter to supply to the various com-

bustion chambers the combustible mixture required for its operating condition.

In order to provide for quick unloading of the gas turbine and the stopping of the compressor action, I provide the outlet valve 152 so that by
5 opening the latter, the gases that would ordinarily be delivered to the gas turbine are permitted to pass by the latter, leaving it de-energized. This by-pass valve 152 is controlled in accordance with the desired regulating action of
10 the system by means of the actuating unit 154.

In the preferred arrangement, the compressor aggregate includes an auxiliary steam turbine 123 as explained before. The system may also be
15 operated without such auxiliary steam turbine, in which case I provide for additional regulation of the operation of the gas turbine. This additional regulation may be effected for instance, by means of the by-pass valves 231 on the boiler unit which admit to the turbine a part of the
20 combustion gases of the boilers before they have delivered their energy to the heat exchanging surfaces, or in some of the other ways described in my prior applications referred to above. By the control of the admission of such additional
25 working gases to the gas turbine the speed of the gas turbine may be controlled within relatively wide limits and thereby the compressor operation controlled without taking resort to other auxiliary driving sources such as an auxiliary
30 steam turbine 123.

I prefer however, as a rule, to utilize the auxiliary steam turbine 123 in connection with the compressor aggregate 120 because in this way
35 the operation of the compressor may be influenced more effectively and more quickly than by the control of the gas turbine. With ordinary fuel oil I design the compressors so that they deliver approximately 8 to 15 per cent of the power of the generated steam. In the preferred con-
40 structions, I design the auxiliary steam turbine 123 so that it shall deliver approximately one-fourth to one-half of the energy consumed by the compressor. The auxiliary steam turbine delivers about 4 to 5 per cent of the power of the
45 generated steam. It is understood of course, that the foregoing figures are illustrative only, and are not intended to limit the scope of my invention.

In the preferred arrangements, such as indicated above, the auxiliary steam turbine 123
50 takes care of the regulation of the operation of the compressor. The operation of the steam turbine itself is controlled by means of the steam inlet valves 162 and 163, and the turbine may be
55 caused to deliver more or less energy to the compressor by closing or opening one or more of the steam inlet valves 162, 163. The opening and closure of these valves is in turn controlled by the oil controlled valve actuating units 164, 165, 166, in accordance with the pressure of the oil control lines to which the respective valves are connected. This pressure in these control lines is in turn controlled by the compressor speed controller 202.

65 It is desirable to operate the auxiliary steam turbine so that it shall always run above a predetermined minimum number of revolutions. This may be effected by so connecting the control valves 218, 219 actuated by the governor 202 of
70 the compressor aggregate that upon decrease of the speed below the predetermined value, the oil in one or another of the oil control lines controlled by one of said valves 218, 219, will be caused to rise to a value at which the steam inlet
75

valve 162 will open, admitting steam to the auxiliary steam turbine 123 and cause it to pick up its speed and run above the predetermined minimum value. In like manner, the auxiliary steam turbine may be arranged to operate at constant speed, as is practiced in ordinary steam turbines, the governor 202 controlling through its slide valves 218, 219, the opening and closure of the steam inlet valves 162, 163 of the auxiliary steam turbine so as to maintain it at a predetermined constant speed.

The slide valves 218, 219, are as a rule arranged also as safety control valves so that upon exceeding a predetermined maximum speed, the steam inlet valves 162, 163 are caused to be closed and the exhaust outlet valve 152 is caused to be open, thereby preventing the compressor aggregate from exceeding its maximum speed.

In determining the relative magnitude of the auxiliary steam turbine 123 with respect to the compressor 121 and the exhaust gas turbine 122, I preferably arrange the several elements so that during steady operation substantially all of the energy of the compressor is delivered by the exhaust gas turbine. Under such conditions the auxiliary steam turbine is relatively small and there is always available sufficient steam for carrying out with it, the desired regulating action as explained before.

The oil fuel distributing system of my improved arrangement may be designed in accordance with the principles established in the construction of Diesel motors and need not be explained here in detail. Where oil fuel is used as in the arrangement shown in the drawings, the fuel is injected into the combustion chamber by increasing the oil fuel pressure. The regulation of the fuel supply will vary, depending on the character of the drive of the pump and on the type of regulation employed.

Where the steam generator is regulated purely by the variation of the number of cycles, the fuel pump may be directly connected to the cam driving shaft 131 and be driven in unison therewith without further regulation. Where the pump is driven by the cam driving shaft 131 and the steam generation is regulated by other elements of the system, additional control of the pump operation or fuel distribution is necessary, such as by utilizing the control device 127 for varying the pressure of the oil or for cutting out or in, more or less of the fuel discharge nozzles. Where the number of fuel discharge nozzles is varied, excessive pressure in the fuel supply lines leading to the active discharge nozzles may be avoided by providing in association with the fuel pumps, overflow valves which maintain the oil fuel pressure constant, irrespective of the oil consumption. As explained before, the fuel pump may also be driven by the compressor shaft or by an independent motor with either constant or variable speed. Where the fuel pump is driven at a speed independent of the cam shaft 131, it is desirable to associate with the fuel supply line a pressure balancing reservoir which prevents excessive variation of the injecting pressure of the fuel. The fuel pressure in the several lines and the quantity of the fuel may also be readily controlled by choke valves in the individual fuel supply lines.

While I have described an embodiment of my invention employing oil or liquid fuel injected into a compressed charge of air delivered to the combustion chambers, the arrangement may also be used with a gaseous fuel source, in which latter

case the gaseous fuel is admixed to the air prior to its entering into the compressor 121 or after it has passed the compressor, the gaseous fuel being supplied to the combustion chamber by a separate compressor line. Instead of gaseous or liquid fuels I also contemplate the use of powdered coal in any of the approved ways in which its use has been described in connection with combustion apparatus.

The various elements described above may be connected into a number of different regulating systems for carrying out a variety of novel regulating operations. I contemplate particularly, two different types of regulation and regulating systems, namely, regulation for constant steam pressure, or in general, the pressure condition of the steam generator; and regulation for constant speed of the main turbine, or in general, the operating condition of the main turbine. I shall now describe the various regulating systems in order.

A. REGULATION FOR CONSTANT STEAM PRESSURE IN THE STEAM COLLECTOR

In this system of regulation, the steam turbine or turbines that are supplied by the steam generating plant receive steam under constant pressure as in the usual prior art arrangements, the steam pressure being delivered at substantially constant pressure and with a substantially constant degree of superheating. The steam turbines or apparatus driven by the steam are in this case provided with the usual regulating devices associated with such turbines and the steam pressure is controlled so as to maintain it at a constant value. With this type of regulation I utilize either one of the following classes of regulation:

1. Pre mixture regulation—Control of fuel quantity

With this regulating system, the compressor is arranged to operate with a constant speed, and likewise, the cycle controller is arranged to operate at a constant number of cycles per minute. Only the amount of the fuel delivered to the combustion chamber is controlled in accordance with the pressure of the steam. When the steam pressure rises the fuel supply is reduced, and when the steam pressure falls the fuel supply is increased.

To maintain this regulating action, the auxiliary steam turbine is set so as to operate at constant speed. This may be effected by the fly ball governor 202. All the steam inlet valves 162, 163 of the auxiliary turbine 123 are then connected to one of the oil control lines, say to control line 148, and this oil control line is regulated by one of the sleeve valves, say valve 218, of the fly ball governor 215. This fly ball governor will regulate the pressure in the control line 148 so that an increase of the speed of the compressor shaft will cause more oil to be released from the oil control line, reducing the pressure applied to the actuating pistons of the actuating units 164, 165 and 166, and thereby causing the valves to close. As a rule, the springs of the several actuating units 164, 165, 166 of the several steam inlet valves of the auxiliary turbine are set so that the individual valves open at different pressures of the oil acting on the actuating pistons of the several actuating units. When the oil pressure is high, all the valves will be open; with the decrease of the oil pressure, one valve after another

other will close, thereby reducing the admission of the steam to the auxiliary steam turbine.

As explained before, the compressor aggregate will be principally driven by the gas turbine 122, and the auxiliary steam turbine will only be controlled to prevent the speed of the compressor aggregate from deviating from the desired fixed value.

With this kind of regulation, the cam shaft 131 may be coupled to the shaft of the compressor aggregate and its speed maintained in unison therewith; or the cam shaft 131 may be driven by its separate electric driving motor 180, arranged for constant speed operation.

The fuel supply control, which is the only control carried out with this system of regulation, is effected by either varying the speed of the fuel pump or its displacement, or by controlling the pressure of the oil in the fuel supply lines, or by varying the number of injecting nozzles, in accordance with the pressure of the steam. The steam regulator 201 with its associated control valves 212, 213 may be used for this purpose. Where the fuel supply is controlled by varying the speed of the fuel pump, the rheostat 188 of the pump motor 187 is arranged to be actuated by an actuating unit 189 and the pressure of the oil line connected thereto is in turn regulated by steam pressure regulator 201. To this end, the oil control line 149 may be used, the oil inlet of the actuating device 189 being connected to this oil control line 149 and the slide valve 212 of the steam pressure regulator being likewise connected to this control line. The slide valve 212 is adjusted so as to vary the pressure of the oil in the oil control line 149 and the oil from the latter line in turn, causes more or less of the resistance of the rheostat 188 to be cut into the circuit of the motor 187, thereby varying its speed.

The action of the slide valve 212 in this arrangement is so adjusted that upon increase of the steam pressure and consequent downward movement of the slide valve 212, the outflow of oil from the oil control line 149 is increased, thereby reducing the pressure in the oil control line 149, and as a consequence, causing the piston of the actuating unit 189 to be moved downwardly by the spring of said unit. This cuts in the rheostat 188 in the circuit of the motor 187 and reduces its speed, thereby slowing down the action of the fuel pump 125 and reducing the amount of fuel delivered to the combustion chambers. In a similar way, the decrease in the steam pressure causes the slide of the slide valve 212 to move in opposite direction, increasing the pressure of the oil control line 149 and causing the actuating unit 189 to cut out the resistance of the rheostat to increase the speed of the fuel pump and delivery of fuel to the combustion chambers.

Instead of controlling the fuel supply by varying the speed of the fuel driving motor 187 in accordance with the pressure of the steam, the fuel pump may be operated at constant speed and the fuel control device 127 operated in accordance with the steam pressure to choke off or release a part of the fuel oil flow through the fuel discharge nozzles of the combustion chambers, or the number of effective discharge nozzles may be varied. To this end, the actuating device 128 is connected to the fuel control line 149 instead of the actuating device 189, and the actuating device 128 is controlled by the steam pressure regulator 201 in the same way as the motor 187 is controlled in the last described arrangement. In other words, an increase of the steam pressure

causes the slide valve 212 to decrease the pressure in the oil control line 149, thereby actuating the actuating unit 128 to reduce the size of the fuel supply openings or the number of active discharge nozzles, or both; and conversely, a decrease of the steam pressure causes the slide valve 212 of the pressure regulator 201 to increase the oil pressure in the oil control line 149, thereby causing the spring actuating unit 128 to operate the control device 127 so as to increase the fuel supply to the combustion chambers or increase the number of effective discharge nozzles.

2. Pure quantity regulation—Cycle control

With this arrangement the compressor aggregate 120 is operated at constant speed in the same way as in the control system described under (1). However, the speed of the cycle aggregate 130 is varied in accordance with the pressure of the steam, increasing the number of cycles if the steam pressure decreases, or decreasing the number of cycles if the steam pressure increases above the predetermined desired value.

To carry out this system of regulation, the cycle aggregate may be operated in accordance with any one of a number of different arrangements. The cycle aggregate may be driven by a separate driving motor 180 and the speed of the latter may be controlled by a rheostat 181 through the oil controlled actuating unit 182, by the action of the steam pressure regulator 201, and its controlling effect on the pressure of the oil control line connected to the actuating unit 182, for instance, the oil control line 149.

Instead of the control through a separate electric motor 180, the driving shaft 131 of the cycle aggregate may be coupled and directly driven from the compressor aggregate through the variable speed transmission mechanism 184. With the latter arrangement, the speed of the cycle aggregate is varied in accordance with the steam pressure by controlling the operation of the actuating unit 185 of the speed change device 184 in accordance with the pressure of the oil line 149 that is regulated by the steam pressure regulator 201.

The fuel pump in the foregoing system of regulation is preferably directly connected to the cycle aggregate 130 and driven in unison therewith. The fuel supply will then vary together with the variation of the number of cycles so that upon increase of the number of cycles, the fuel supply will be increased, and vice versa.

3. Combined mixture and quantity regulation—control of air and fuel supply

In this system of regulation, the number of cycles is maintained constant and both the compressor action and the fuel supply are varied in accordance with the steam pressure.

To carry out this system of regulation, the cycle aggregate 130 is operated by its separate driving motor 180 at constant speed, a motor of constant speed characteristics being used for this purpose. The speed of the compressor aggregate and of the fuel supply however, are controlled in accordance with the action of the steam pressure regulator 201. This may be effected by subjecting the operation of the inlet valves 162, 163 of the auxiliary steam turbine and of the control valves 157, 152, 231, of the auxiliary gas turbine, to the action of the steam pressure regulator 201.

To this end, the actuating units of the several valves, namely, the actuating units 164, 165, 166, 154, 159 and 234 are all connected to the oil con-

5 trol line 149, the pressure of which is regulated by the steam regulator 201 in the way described above. Instead of having all the control valves 162, 163, 153, 157 and 231 controlled from one control line and by a single slide valve 212 of the steam control regulator 201, only some of these valves may be subjected to the control action of the regulator, and some of these valves may be controlled by one oil control line, and others by another of the oil control lines.

10 Where the several valves which control the operation of the auxiliary steam turbine 123 and the exhaust gas turbine 122 are actuated from a common oil control line, the actuating units of the several valves are so adjusted as to successively respond to different pressures so that one valve after another is called into action to carry out the desired regulating operation. Thus, for instance, the valves 162, 163 of the auxiliary steam turbine 123 are so adjusted that valve 162 opens first at a predetermined pressure of the oil control line 149, and the valves 163 open afterwards in succession as the oil pressure in the oil control line 149 is caused to rise with the continued upward movement of the slide 210 in the slide valve 212, in response to the drop of the steam pressure, the slide valve 212 being in this case so adjusted that upon decrease of the pressure, the oil outflow from the control line 149 is cut down and the pressure in the oil control line 149 thereby increased, and vice versa.

20 The admission of additional steam to the auxiliary turbine 123 in response to a drop in the steam pressure results in an increase of the speed of the compressor aggregate with the consequent increase in the amount of air supplied to the combustion chambers of the boiler units. With this is coupled an increase of the fuel supply to the combustion chambers so that as a result, upon decrease of the steam pressure below the predetermined value, the combustion chambers of the several boiler units are charged with a greater quantity of combustible mixture, thereby increasing the steam delivery of the boiler units. This increase of the combustion energy increases also the energy of the exhaust gases discharged from the heat exchange units by the outlet valves 107, thereby delivering to the gas turbine more energy. As a consequence, the gas turbine takes over from the auxiliary steam turbine the delivery of the additional power necessary to drive the compressor in its new condition, until the steam pressure has again been restored and the steam inlet valves 162, 163 of the auxiliary turbine have again returned to their normal position.

55 In a like way, an increase of the steam pressure causes the steam pressure regulator 201 to bring about the closure of the steam inlet valves 162, 163 of the auxiliary steam turbine, thereby reducing the speed of the compressor, and if necessary, a further speed reduction of the compressor is brought about by calling into action the gas by-pass valve 152, the latter being operated by its actuating unit 154 when the pressure of the oil control line 149 with which it may be connected increases above a predetermined value in relation to the operation of the actuating units of the other valves of the regulating system.

70 Where no auxiliary steam turbine is used, and the control of the speed of the compressor aggregate is effected entirely by the control of the gas turbine 122, as for instance through valves 231 which admit part of the combustion gases directly into the gas turbine, the actuating units 234 of said by-pass valves are connected to the oil con-

trol line 149, the pressure of which is regulated by the steam pressure regulator 201 so as to cause the gas turbine to increase the speed in response to a fall in the steam pressure, and decrease the speed in response to a rise in the steam pressure.

5 The fuel supply in this system of regulation may be controlled in unison with the compressor aggregate in which case, the fuel pump 125 may be directly coupled to the compressor aggregate. With such arrangement the amount of fuel supplied to the combustion chambers will rise and fall together with the amount of air supplied to the combustion chambers. The fuel pump may also be driven separately and the control of the fuel supply to the combustion chambers may be regulated in accordance with the steam pressure in any of the ways explained in connection with regulation system (2) described above.

4. Combined mixture and quantity regulation—Control of air and fuel supply and of number of cycles

15 In this regulating system, regulation is effected by varying the amount of air delivered to the combustion chambers as well as the fuel delivered and the number of combustion cycles per unit time.

25 The number of combustion cycles is arranged to be increased with the increase of the load, being small for a small load, and likewise the speed of the compressor is caused to increase with the increase of the load, and together with it, the fuel supply is also controlled so as to increase the amount of fuel delivered to the combustion chambers as the amount of air and the number of cycles are increased.

35 The compressor aggregate may be regulated in the same way as in the arrangement described under (3). The cycle control aggregate and the fuel control aggregate may likewise be regulated in the ways described in connection with the regulating systems (1), (2) and (3) described above. Thus, the compressor aggregate is controlled in response to the steam pressure regulator 201 so as to cause the speed to rise and fall in response to the drop or rise of the steam pressure. The cycle aggregate and its cam shaft 131 may be either directly coupled to the compressor aggregate so as to go through changes of speed of the same character as the speed changes of the compressor aggregate, or the cycle aggregate may be driven by a separate motor as explained before, and the speed of this motor regulated by the steam pressure as described before. The fuel pump may be connected either to the cycle aggregate or to the compressor aggregate, or driven separately and controlled in accordance with the steam pressure in the way described in the prior arrangements.

60 In the diagram Fig. 5, there is given a detail diagrammatic view of the principal regulating elements of the arrangement shown in Fig. 1, connected to carry out the combined regulation of the air quantity, fuel supply and cycle number referred to here. As explained before, the principal regulating device is the steam pressure regulator 201. Upon drop of the steam pressure the diaphragm 206 of the steam pressure regulator moves upwardly under the action of the control spring 207, depending on its adjustment. This movement of the diaphragm is transmitted to the spindle 209, which in turn actuates the steam admission valves of the auxiliary steam turbine 123 of the compressor aggregate, opening the admission valves and delivering more 75

steam to the auxiliary turbine. This increases the number of revolutions of the auxiliary turbine and of the compressor 131 coupled thereto, so that more air is compressed by the compressor and a larger weight of air is delivered to the combustion chambers of the several boiler units.

Simultaneously with the increase of the compressor speed there is also increased the speed of the cycle control aggregate and of the fuel pumps either through direct coupling of the same to the compressor aggregate or through independent drives controlled in accordance with the rise and fall of the steam pressure. As a result, the increase of the compressor speed will also be accompanied by an increase of the combustion cycles and of the fuel delivered by the fuel discharge nozzles to the combustion chambers. Through the combustion of the increased quantity of fuel and combustion air delivered to the combustion chambers, more heat is generated and supplied to the heat exchangers, thereby converting more of the water into steam, causing the steam pressure to rise again. At the same time, the energy of the exhaust gases discharged from the heat exchangers has become larger and more energy is delivered to the gas turbine 122. This enables the gas turbine to deliver the additional power necessary for operating the compressor at the higher speed and increased capacity, until a stationary condition has been reached and the steam pressure has been brought up again to its normal value, whereupon the steam pressure regulator 201 causes the valves 162, 163 of the auxiliary steam turbine to be brought back to normal position.

In case of a diminution of the steam consumption from the boiler plant, the steam pressure in the collector 14 increases. This causes the steam pressure regulator 201 to operate in opposite direction, closing the valves 162, 163 of the auxiliary steam turbine and reducing the speed of the compressor and of the cycle aggregate and the fuel pumps. In the course of such negative control action the steam pressure regulator 201 may also bring about the opening of the gas exhaust valves 152, reducing the power output of the gas turbine and thereby bringing down the delivery of gas by the compressor until the amount of the combustion energy freed in the combustion chamber has been so far reduced as to restore stationary conditions and the desired steam pressure.

In general, no special regulation of the water circulating pump will be necessary as the quantity of water circulated through the heat exchangers will usually be a multiple of the corresponding quantity of steam generated by the boiler units. The circulating water circuit is preferably so arranged as to include the hollow spaces 33, 47 and 57 of the combustion chamber. The circulating pump may be driven as explained above, although in most cases a separate drive for the water circulating pump will be preferable.

The feed water pump is preferably driven by a separate motor as shown in Fig. 1A and its operation is in accordance with the consumption of steam of the system, or in accordance with the level of the water in the separator 14, or in accordance with any of the usual methods for regulating the feed water supply.

The regulating action of the steam regulator 201 is transmitted to the steam inlet valves 162, 163 by an oil control line 149 through the action of the slide valve 212 referred to before, depending on the position of the diaphragm 206 as de-

termined by the pressure of the steam and the adjustment of the spring 207, this slide valve will open more or less so as to vary the oil pressure in the oil control line 149. The oil control line 149 is also connected to the actuating unit 182 of the controller rheostat 181 of the driving motor 180 of the cycle control aggregate 130. The oil control line 149 is also connected to the regulating valve 219 of the speed governor 202 of the compressor aggregate which is so set that upon increase of the speed of the compressor aggregate above the predetermined value, the sleeve valve 219 opens and releases the oil pressure in the oil control line 149, causing all the steam admission valves 162, 163 to close, irrespective of the operation of the steam regulator valve 201. In addition, the second sleeve valve 213 on the steam pressure regulator 201 serves to regulate the oil pressure in the further oil control line 150 to which is connected the gas outlet valve 152 of the gas turbine. The valve 213 is so adjusted that upon excessive increase of the steam pressure, the pressure from the oil control line 150 is increased, causing the actuating unit 154 of the gas outlet valve 152 to open the valve and reduce the supply of operating medium to the gas turbine 122.

In regulating the operation of the system, the diaphragm 206 will move the spindle 209 up or down, thereby varying the oil outlet openings of the sleeve valves 212, 213. Depending on the quantity of the outflowing oil passing through these valves, the pressure in the oil control systems to which they are connected, respectively, will be higher or lower, so as to cause the actuating pistons of the actuating units 164, 165, 166, 182 of the various control devices to assume different positions. The springs of these actuating devices are so adjusted as to cause the valves controlled thereby to close or open in sequence one after another. In like way, the spring of the actuating unit 182 is so adjusted that depending on the pressure of the oil in the oil control line 149, more or less of the resistance of the rheostat 181 will be cut in the circuit of the driving motor 180 of the cycle control aggregate.

As a result of the foregoing arrangement, the auxiliary turbine 123 and the cycle control aggregate 130 will operate at different speeds, depending on the pressure of the steam on the diaphragm 206. By suitable choice and adjustment of the springs of the actuating units of the steam inlet valves 162, 163 and of the motor control rheostat 181, and of the resistance steps of the latter, it is possible to provide for uniform or variable relationship of the variation of the steam turbine speed and the cycle aggregate speed, thereby providing for an adjustment of the relationship of the air pressure and the quantity of air delivered by the compressor, and the number of operating cycles.

In the arrangement described above the speed governor 202 on the compressor aggregate is employed principally for safety purposes so that upon excessive increase of the compressor speed the oil pressure from the oil control line 149 is released, causing all the valves 162, 163 to close. The gas outlet valve 152 may likewise be controlled from the same common oil control line 149 and arranged to release the valve 152 to open position when the oil pressure in the oil control line 149 is released by the action of the governor 202 in case of excessive speed of the compressor aggregate.

5. *Combined mixture quantity regulation—Control of cycle number during low power range—Control of compressor during high power range*

This system of regulation involves a number of distinct features which are of great importance in the practical operation of steam power plants of the type under consideration. In the design of such plants a number of factors require special consideration. The amount of compression suitable for the explosion type boilers of the present invention is subject to certain limitations, partially of economical and partially of operative nature. Similarly, the number of cycles per unit time is to a certain extent limited by the dimensions of the combustion spaces. The charging pressure at which the exhaust gas energy is sufficient to drive the compressor which supplies the compressed air of the combustion chambers and may be regarded as the economically preferred charging pressure. By the term "exhaust gas energy" I designate the energy remaining in the combustion gases after depriving the same of all the heat energy available for steam generation.

The minimum charge pressure is on the other hand to a certain extent limited by the requirement that the gas remnants shall be reliably driven out from the combustion spaces by the incoming charge, and that the fresh charge shall become thoroughly mixed through. The number of charges per unit time, or in other words, the number of cycles, is dependent on the time which is practically required for filling the combustion chamber, for the complete combustion of the charge, and for the thorough removal of the burned combustion gases from the combustion spaces.

Care must also be taken to always secure a good combustible mixture while avoiding admission of over-rich mixtures within the combustion chambers. The best operation is secured by maintaining the most favorable ratio of fuel to the combustion air throughout the entire range of loads under which the system is to operate.

A distinct feature of my invention is the provision of a regulating system whereby, under the maintenance of a substantially constant ratio of the fuel to the air, the fuel-air mixture is regulated in response to the steam pressure or steam supply; and that this regulation is carried out in such manner that the decreasing steam pressure or reduction of steam supply shall first produce an increase of the charge cycles, i. e., the number of charges per time unit, until a certain maximum number of charging cycles has been reached, whereupon the speed of the compressor or the pressure of the charges shall be increased, the additional power input into the compressor being supplied by an auxiliary power source. In other words, during the low power range, when steam consumption is small, the regulating action of the system is carried on principally by varying the number of charging cycles per unit of time, increasing the number of cycles if the steam demand rises, and decreasing the number of cycles if the steam demand falls. While operating in this low power range, the compressor aggregate is maintained at substantially constant speed and the compressed air is delivered with substantially constant pressure to the combustion chambers.

On the other hand, during the high power range where the steam consumption is relatively large, the system operates with a substantially constant, practicably high number of cycles and

the regulating action is carried on by varying the speed of the compressor to increase the compression with the increase of the steam consumption, and decrease the compression with the drop of the steam consumption.

In operating the steam power plant in accordance with this regulating system, the circulating water pump may be directly coupled to the compressor aggregate comprising the compressor 121 with the exhaust gas turbine 122 and the auxiliary steam turbine 123. The principal elements entering into the regulating mechanism of this system are shown in more detail in Fig. 6 of the drawings. The cycle aggregate 130 is driven by its motor 180 separately from the compressor aggregate 120, the speed of each aggregate being arranged to be controlled separately from the other.

To carry out this system of regulation I use two oil control lines 148 and 149, these two oil control lines being in addition to the oil control line 142 which serves to transmit the control action of the cycle control aggregate 130 to the inlet and outlet valves 37 and 107 of the individual boiler units. Two main regulating devices are employed in this system, namely, the steam pressure regulator 201, and the compressor speed regulator 202, the oil control line 149 being in the first instance controlled by the steam regulator 201 through its connection to the slide valve 212 of the latter, and the oil control line 148 being controlled in the first instance through the compressor speed regulator 202 through its connection by its slide valve 217 to the oil control line 148. The several regulating and control devices of the system are so adjusted as to carry out the regulating action in the following manner:

The gas turbine 122 and the compressor 121 are so designed with relation to the boiler units that under steady conditions, for the higher loads up to full load, the exhaust gases from the boiler units are by themselves sufficient to supply to the exhaust turbine the energy necessary for driving the compressor at a speed at which the latter is capable of compressing the combustible mixture or air to the pressure at which it is to be delivered to the combustion chambers. During this regulating range the speed of the compressor aggregate 120 is such that the slide valve 217 actuated by the compressor speed regulator 202 permits sufficient oil from the oil control line 148 to escape so that the pressure within the oil control line 148 is not sufficient to lift the actuating piston of the actuating unit 166 to open the steam admission valve 163, the spring of the actuating unit holding the steam admission valve closed.

The diaphragm of the steam regulator 201 assumes under steady operating conditions its normal position corresponding to the pressure at which the steam boiler is intended to be operated. The slide valve 212 actuated by the steam pressure regulator is so adjusted that under normal steady operating conditions when the steam pressure is at the value at which it is desired to be maintained, the slide valve 212 will choke off sufficient oil outflow from the oil control line 149 to raise the pressure in the oil control line to a value at which it will hold near the upper position the piston of the actuating unit 182 of the electro motor 180 driving the cycle control aggregate 130. Under these conditions substantially the entire resistance of the control rheostat 181 will be cut out so that the motor 180 and with it the cycle aggregate will operate near its upper

speed limit. The boiler plant will then operate with almost its highest number of cycles.

If, under these conditions, the load decreases and the steam pressure in the collector 15 accordingly increases, the diaphragm of the pressure regulator will move the slide valve 212 to a position where the oil outflow through the slide valve is increased, thereby decreasing the pressure in the oil control line 149. This will cause the piston of the actuating unit 182 to be moved downward, cutting in resistance of the regulating rheostat 181 and thereby reducing the speed of the motor 180 driving the cycle control aggregate. This will reduce the number of combustion cycles per unit time and the fuel supply to the boiler units. As a consequence, the steam generation will be decreased in accordance with the reduced steam demand.

It may happen that under these operating conditions the quantity and pressure of the exhaust gases delivered by the boiler unit will not be sufficient to drive the gas turbine and the compressor aggregate at the speed necessary for charging the combustion chambers, notwithstanding the reduced quantity of compressed air necessary at the smaller cycle number. In such case, the compressor speed regulator 202 actuates the slide valve 217 connected thereto so as to choke the outflow of oil from the oil control line 148 and increase the oil pressure thereunder, thus causing the actuating piston of the actuating unit 166 of the steam admission valve 163 to open and admit sufficient steam to the auxiliary turbine 123 to drive the compressor aggregate at the required minimum number of revolutions.

If the load is increased or if a steady overload above the normal load is imposed upon the system, the steam pressure in the steam collector 15 drops and the steam pressure regulator 201 will then cause its slide valve 212 to further choke off the oil outflow from the oil control line 149. This causes the oil in the oil control line 149 not only to hold the actuating piston of the actuating unit 182 in the position at which the cycle aggregate runs with a maximum number of cycles, but also actuates the actuating units 164, 165 to open the steam admission valves 162 of the auxiliary steam turbine in series one after another. As a result, the speed of the compressor aggregate increases, thereby increasing the quantity and the pressure of the fuel-air mixture delivered to the combustion chambers, there being no further increase of the number of combustion cycles.

With the increase of the thermal energy released in the boilers more heat is transferred to the water that is circulated through the boiler units, causing the steam pressure to increase so that, as a result, the steam pressure regulator 201 is again brought to its normal position, reducing thereby oil pressure in the oil control line 149 and causing the excessive steam inlet valves 162 to close again. In the main, this auxiliary steam serves principally to increase the number of revolutions of the compressor aggregate to a value at which the compressor will deliver to the boiler a combustible mixture of sufficient quantity and pressure to supply the thermal energy necessary for producing the amount of steam required under the increased load. Once this higher number of revolutions of the compressor aggregate has been reached, the additional volume and pressure of the exhaust gases supplied to the gas turbine may be either fully or at least largely sufficient in order to maintain the compressor aggregate

at its proper operating condition. Accordingly, even at large overloads, the steam admission valves 162 of the auxiliary turbine may be closed to a more or less extent, once the steady state has been reached, small load variations being taken care of by the cycle aggregate and regulation of the speed of the latter in the manner described before.

In order to prevent the compressor aggregate from running away or from being over-regulated, the compressor speed regulator 202 has the additional slide valve 219 which is connected to the oil control line 149 of the steam admission valves 162 so adjusted that upon excessive speed of the compressor aggregate the oil from the oil control line 149 is released, causing the oil admission valves 162, 163 to close.

Instead of making the regulating arrangements described herein operate through the medium of oil control lines and hydraulic actuating devices, either mechanical or electrical or any other suitable system of regulating apparatus may be used in accordance with the principles disclosed herein. In like way, instead of using electrical motors for auxiliary drives, other driving apparatus may be employed. Similarly, the auxiliary steam turbine may be replaced by an equivalent mechanism, for instance, an electric motor, the latter being regulated in accordance with the same principles as applied to the regulation of the auxiliary steam turbine. In like way, a steam turbine could be used for driving the cycle aggregate, and the speed of the steam turbine may be regulated in accordance with the same principles as the electric motor therefor described above.

In order to make the plant self-sufficient, a special steam collector may be provided for the auxiliary steam, particularly as the quantity of steam required is relatively small. The system adapts itself, however, also for use of steam from a distinct source for the auxiliary apparatus. It is of advantage to arrange the steam collector as an accumulator or to connect it with a separate container that can be used as an accumulator in order to always have available sufficient steam for starting the compressor and circulating pump aggregates. The accumulating capacity of such containers can be made relatively small since the steam generation of the new boiler is an immediate one, and in the presence of water from an accumulator, which is nearly the temperature of evaporation, the load on the steam generator may be imposed at once.

In the last described exemplification it has been assumed that the fuel-air mixing ratio is maintained substantially constant. This can be readily provided for by regulating the fuel supply in accordance with the operation of the compressor aggregate and the cycle aggregate, for instance, the fuel supply may be controlled by a regulating device 204, a Venturi meter for instance, depending on the flow conditions of compressed air in the compress air line 155, the regulating device serving to operate slide valves 241. Through the medium of this slide valve, the oil pressure in the oil control line 150 is regulated, this pressure being utilized to regulate the actuating unit 189 of the fuel pump motor controller 188 or the fuel distribution control unit 128. The fuel supply may also be controlled in response to the pressure of the oil control line 149 in accordance with the action of the steam pressure regulator 201. Since the oil pressure in this oil control line 149 will to a large extent respond to the conditions determining the amount of heat input necessary to

maintain the boiler system in the desired operating condition, the regulating action of the oil pressure in this line will be very effective for the regulation of the fuel supply.

5 In the preferred arrangement utilizing the regulating system of the type described above, the compressor aggregate is arranged so as to operate with substantially constant compressor speed up to about $\frac{3}{4}$ of the load on the steam generator, 10 the cycle aggregate operating during this range with a variable number of cycles depending on the load. In case of steam demand above this limit, the cycle aggregate operates at its maximum number of cycles and the speed of the compressor aggregate is varied. The several elements 15 are so arranged that up to a certain position, the steam pressure regulator regulates the number of revolutions of the motor driving the cycle aggregate, while a speed regulator operated by the 20 shaft of the compressor aggregate is holding the speed of the compressor aggregate at the desired mean value. In case of further increase of the load, and drop in the pressure in the steam collector, the cycle aggregate operates with the maximum number of cycles and the compressor speed 25 is caused to be increased through the action of the steam pressure regulator, admitting more steam to the auxiliary turbine.

With the foregoing arrangement, the fuel pump 30 and supply may be operated either from the compressor aggregate or from the cycle aggregate, or by a separate motor. Where the fuel pump is driven by the cycle aggregate, the regulating devices for controlling the fuel flow are so arranged 35 that after the cycle aggregate has reached its highest number of revolutions, additional fuel oil is supplied to the boiler units in accordance with the increased delivery of compressed air to the boiler units. This may be done, for instance, 40 by causing the actuating unit 189 or actuating unit 128 to be operated by the oil control line 150, the pressure in the latter being regulated in accordance with the degree of compression by the regulator 204, or by the speed of the compressor 45 through the action of the compressor speed governor 202. Where the fuel supply pump is driven by the compressor, provision must be made for decreasing the amount of fuel delivered to the combustion chambers while the 50 cycle aggregate is running below its maximum speed. This may be effected for instance, by mounting on the shaft of the cycle aggregate a governor like the governor 202 of the compressor aggregate and causing the regulating action of 55 this governor to control the actuating unit 185 of the variable speed transmission mechanism 184 so as to increase the speed of the fuel pump with the increased number of cycles, and vice versa.

60 6. Regulation through control of number of active boiler units

Where the number of boiler units is more than three, additional control of the rate of the steam generation may be obtained by controlling the 65 number of boiler units that are effective. Thus, for instance, the steam pressure regulator may be arranged to control a separate oil control line which in turn is arranged so as to cut in or out of action, one or more of the boiler units of the 70 system, depending on the steam pressure. With this type of regulation, any one or all of the different systems of regulation described hereinabove under (1)–(5) may be utilized and combined.

75 In regulating systems utilizing a variable num-

ber of combustion chambers it is of great advantage to provide a separate oil pump for each combustion chamber so that on cutting in or out the combustion chamber, its associated oil pump is 5 simultaneously cut in or cut out. Such arrangement of separate oil pumps for each combustion chamber is also advantageous with the regulating systems where the number of combustion chambers is not varied. Where a common fuel pump 10 is used with a variable number of combustion chambers, suitable pressure equalizing means for maintaining the proper oil pressure under all conditions of operation have to be provided. Likewise, suitable air compression equalizing 15 means have to be provided so that in connecting or cutting out additional boiler units, the units that are in operation will receive the combustible mixture charge at the proper pressure.

B. REGULATION FOR CONSTANT SPEED OF THE MAIN TURBINE 20

In this system of regulation the steam turbine and the boiler plant are operated as a unit and the rate of steam generation is regulated directly 25 in accordance with the operating condition of the turbine. My improved boiler system and boiler construction make possible this type of regulation by reason of the fact that the rate of steam generation in my boilers follows directly the variation of the fuel or combustible mixture supply 30 to the boilers, so that the operation of the turbine governor may be directly utilized for controlling the supply of fuel to the steam generators.

With such arrangement, all the various means ordinarily used in connection with steam turbines 35 for regulating the same may be dispensed with and the regulator of the turbine is arranged to operate directly the control mechanisms of the steam generators. In such operating systems, the pressure of the steam may vary within very 40 wide limits. The steam quantity is determined by the steam consumption capacity of the steam turbine.

With constant steam inlet openings, the steam consumption of the turbine is determined by the 45 pressure and the temperature of the steam and the latter in turn, by the quantity of combustion material released by the regulator. This type of regulation of the operation of the steam boilers may be carried out in accordance with the various 50 systems explained under (A), the turbine speed governor 203 being merely substituted as a regulating element in place of the steam pressure regulator 201. In other words, wherever in the various systems of regulation (1)–(6) described under (A) hereinabove, the regulating 55 action is carried on by the steam pressure regulator 201, in the presently described system of regulation the regulating action is effected through the governor of the main turbine 10. 60 The regulating devices are so connected that upon increase of the turbine speed above the desired value, the steam generation is reduced, and upon decrease of the turbine speed below its set value, the steam generation is increased. 65

The regulating systems and the arrangements described above may also be combined with the regulating systems and arrangements described 70 in my co-pending applications Serial No. 333,453, filed January 18, 1929, 343,745 and 343,746, both filed on March 1, 1929, referred to before.

The various apparatus and mechanisms and details of operation to which I referred in describing the invention are intended only as ex- 75

emplifications, as many modifications and variations thereof might be utilized for carrying out my invention. I accordingly desire that the appended claims be given a broad construction commensurate with the scope of the invention within the art.

I claim:

1. The method of regulating the generation of steam in explosion type steam generators provided with compressors for compressing gas which forms with fuel a combustible mixture which comprises maintaining the compressor speed substantially constant, regulating the number of combustion cycles, and regulating the fuel supply in accordance with the variations of the number of cycles.

2. The method of regulating the steam generation in a steam generator of the explosion type which comprises increasing the compression of the combustion charges in response to decrease of the pressure of the generated steam, and decreasing the compression in response to increase of said pressure independently of the control of the number of combustion cycles per unit time.

3. The method of regulating the steam generation in a steam generator of the explosion type which comprises increasing the compression of the combustion charges in response to increase of the load in the steam consuming device energized by the generated steam, and decreasing the compression in response to decrease in said load independently of the control of the number of explosion cycles per unit time.

4. In a steam generator, a combustion chamber, means including a compressor for supplying a combustible mixture under pressure to said chamber, means for periodically exploding said combustible mixture in said chamber to produce combustion gases of high temperature and higher pressure than said combustible mixture, a heat exchanger holding a fluid to be heated and having heat exchange surfaces in the form of a plurality of relatively narrow gas discharge ducts immersed in the fluid and connected to said chamber for discharging combustion gases therefrom, said heat exchanger being so arranged relatively to said combustion chamber that the high pressure of the produced combustion gases imparts to said gases a high velocity through said ducts at which the major portion of the heat present in the produced combustion gases is transferred to the fluid surrounding said ducts while said gases flow through said ducts at said high velocity to generate steam, and regulating means for regulating the number of combustion cycles per minute in response to the variations of the steam load while maintaining the combustible mixture under substantially constant pressure.

5. In a steam generator, a combustion chamber, means including a compressor for supplying a combustible charge under pressure to said chamber, means for periodically exploding said combustible charge in said chamber to produce combustion gases of high temperature and higher pressure than said combustible charge, a heat exchanger holding a fluid to be heated and having heat exchange surfaces in the form of a plurality of relatively narrow gas discharge ducts immersed in the fluid and connected to said chamber for discharging combustion gases therefrom, said heat exchanger being so arranged relatively to said combustion chamber that the high pressure of the produced combustion gases imparts to said gases a high velocity through said ducts at which the major portion of the heat

present in the produced combustion gases is transferred to the fluid surrounding said ducts while said gases flow through said ducts at high velocity to generate steam, and regulating means for regulating the compression of the combustible charge in response to the variations of the steam load while maintaining the number of combustion cycles per minute substantially constant.

6. In a steam generator, a combustion chamber, means including a compressor for supplying a combustible charge under pressure to said chamber, means for periodically exploding said combustible charge in said chamber to produce combustion gases of high temperature and higher pressure than said combustible charge, a heat exchanger holding a liquid to be heated and having heat exchange surfaces in the form of a plurality of relatively narrow gas discharge ducts immersed in the liquid and connected to said chamber for discharging combustion gases therefrom, said heat exchanger being so arranged relatively to said combustion chamber that the high pressure of the produced combustion gases imparts to said gases a high velocity through said ducts at which the major portion of the heat present in the produced combustion gases is transferred to the liquid surrounding said ducts while said gases flow through said ducts at high velocity to generate steam, and regulating means responsive to the variations of the steam load for regulating the number of combustion cycles per minute during one load range of said generator and for regulating the compression of the combustible charge during another load range of said generator.

7. In a steam generator, a combustion chamber, means including a compressor for supplying a combustible charge under pressure to said chamber, means for periodically exploding said combustible charge in said chamber to produce combustion gases of high temperature and higher pressure than said combustible charge, a heat exchanger holding a liquid to be heated and having heat exchange surfaces in the form of a plurality of relatively narrow gas discharge ducts immersed in said liquid and connected to said chamber for discharging combustion gases therefrom, said heat exchanger being so arranged relatively to said combustion chamber that the high pressure of the produced combustion gases imparts to said gases a high velocity through said ducts at which the major portion of the heat present in the produced combustion gases is transferred to the liquid surrounding said ducts while said gases flow through said ducts at high velocity to generate steam, and regulating means responsive to the variations of the steam load for regulating the number of combustion cycles during the low load range of said generator and for regulating the amount of compression of the combustible charge during the high load range of said generator.

8. In a steam generator, a combustion chamber, means including a compressor for supplying a combustible charge under pressure to said chamber, means for periodically exploding said combustible charge in said chamber to produce combustion gases of high temperature and higher pressure than said combustible charge, a heat exchanger holding a liquid to be heated and having heat exchange surfaces in the form of a plurality of relatively narrow gas discharge ducts immersed in the liquid and connected to said chamber for discharging combustion gases therefrom, said heat exchanger being so arranged rela-

tively to said combustion chamber that the high pressure of the produced combustion gases imparts to said gases a high velocity through said ducts at which the major portion of the heat present in the produced combustion gases is transferred to the fluid surrounding said ducts while said gases flow through said ducts at high velocity to generate steam, and regulating means responsive to the variations of the steam load for regulating the number of combustion cycles during the low load range of the generator until a predetermined number of cycles is reached corresponding to about three-fourths of the normal load of said generator and to regulate the amount of compression of the combustible charge during the high load range, while maintaining the number of combustion cycles substantially constant.

9. The method of regulating the steam generation of a steam generator in which means including a compressor supply a combustible charge to a closed combustion chamber subjecting it to periodical combustion therein, and the pressure of the combustion gases in the chamber is applied for driving the hot combustion gases through heat exchange ducts surrounded by a steam generating liquid at high velocities of about 200 meters per second or more at which the major part of the heat contained in the gases is delivered to said liquid and steam generated therefrom, comprising regulating the rate of the explosion cycles during one range of operation, and regulating the speed of the compressor during another range of operation while maintaining the rate of the combustion cycles substantially constant.

10. The method of regulating the steam generation of a steam generator in which means including a compressor supply a combustible charge to a closed combustion chamber subjecting it to periodical combustion therein, and the pressure of the combustion gases in the chamber is applied for driving the hot combustion gases through heat exchange ducts surrounded by a steam generating liquid at high velocities of about 200 meters per second or more at which the major part of the heat contained in the gases is delivered to said liquid and steam generated therefrom, comprising increasing the number of the explosion cycles per minute in response to steam load increase until a predetermined maximum number of cycles per minute has been reached while maintaining the compressor speed substantially constant, increasing the compression of the charge in response to further steam load increase while maintaining the number of cycles per minute substantially unchanged, and vice versa.

11. The method of regulating the steam generation of a steam generator in which means including a compressor supply a combustible charge to a closed combustion chamber subjecting it to periodical combustion therein, and the pressure of the combustion gases in the chamber is applied for driving the hot combustion gases through heat exchange ducts surrounded by a steam generating liquid at high velocities of about 200 meters per second or more at which the major part of the heat contained in the gases is delivered to said liquid and steam generated therefrom, comprising increasing the number of the explosion cycles per minute in response to steam load increase until a predetermined maximum number of cycles per minute has been reached, increasing the speed of the compressor in response to further steam load increase while maintaining

the number of cycles per second substantially unchanged, and vice versa.

12. The method of regulating the steam generation of a steam generator in which means including a compressor supply a combustible charge under compression to a closed combustion chamber subjecting it to periodical combustion therein, and the pressure of the combustion gases in the chamber is applied for driving the hot combustion gases through heat exchange ducts surrounded by a steam generating liquid at high velocities to transfer heat to said liquid and generate steam, comprising regulating the rate of the explosion cycles during one range of operation and regulating the compression of the charge during another range of operation while maintaining the rate of the combustion cycles substantially constant.

13. The method of regulating the steam generation of a steam generator in which means including a compressor supply a combustible charge under compression to a closed combustion chamber subjecting it to periodical combustion therein, and the pressure of the combustion gases in the chamber is applied for driving the hot combustion gases through heat exchange ducts surrounded by a steam generating liquid at high velocities to transfer heat to said liquid and generate steam, comprising regulating the rate of the explosion cycles during the low steam load range and regulating the compression of the charge during the high steam load range while maintaining the rate of the combustion cycles substantially constant.

14. The method of regulating the steam generation in steam generators in which means including a compressor supply a combustible charge under compression to a combustion chamber subjecting it to periodical combustion therein and the pressure developed by the combustion is utilized to impart to the gases of combustion high velocities through heat exchange ducts surrounded by a steam generating liquid at which the major part of the heat contained in the gases is delivered to said liquid while the gases flow at high velocity, which comprises increasing the number of combustion cycles per minute with increasing load until a predetermined number of cycles per minute has been reached, and upon further increase of the load, regulating the compression of the combustible charge of said combustion chamber while maintaining the number of cycles per minute substantially constant.

15. The method of regulating the steam generation of a steam generator in which means including a compressor supply a combustible charge under compression to a closed combustion chamber subjecting it to periodical combustion therein, and the pressure of the combustion gases in the chamber is applied for driving the hot combustion gases through heat exchange ducts surrounded by a steam generating liquid at high velocities to transfer heat to said liquid and generate steam, comprising regulating the rate of the explosion cycles during the low steam load range while maintaining the compression of the charge substantially constant and regulating the compression of the charge during the high steam load range while maintaining the rate of the combustion cycles substantially constant.

16. The method of regulating the steam generation of a steam generator in which means including a compressor supply a combustible charge under compression to a closed combustion chamber subjecting it to periodical combustion

therein, and the pressure of the combustion gases in the chamber is applied for driving the hot combustion gases through heat exchange ducts surrounded by a steam generating liquid at high velocities to transfer heat to said liquid and generate steam, comprising increasing the number of the explosion cycles per minute in response to steam load increase until a predetermined maximum number of cycles per minute has been reached, increasing the speed of the compressor in response to further steam load increase while maintaining the number of cycles per second substantially unchanged, and vice versa.

17. The method of regulating the steam generation of a steam generator in which means utilizing a compressor compress a gaseous body and produce therewith a combustible charge of substantially raised pressure supplied to a closed combustion chamber, subjecting it therein to combustion under raised pressure, applying the pressure of the hot combustion gases for imparting to said gases a velocity of a predetermined high range through heat exchange ducts surrounded by steam generating fluid to transfer heat to said fluid and generate steam, and applying the cooled down combustion gases discharged from said ducts to impel a gas turbine for driving said compressor, comprising increasing the speed of said compressor in response to decrease of the steam pressure of said generator and decreasing the speed of said compressor in response to an increase in said steam pressure.

18. The method of regulating the steam generation of a steam generator in which means utilizing a compressor compress a gaseous body and produce therewith a combustible charge of substantially raised pressure supplied to a closed combustion chamber, subjecting it therein to combustion under raised pressure, applying the pressure of the hot combustion gases for imparting to said gases a velocity of a predetermined high range through heat exchange ducts surrounded by steam generating fluid to transfer heat to said fluid and generate steam, applying the cooled down combustion gases discharged from said ducts to impel a gas turbine for supplying a part of the driving power required by said compressor for imparting said high velocity to said combustion gases, and an auxiliary motor driving said compressor for supplying thereto additional power required for driving said compressor, comprising increasing the speed of said auxiliary motor in response to decrease of the steam pressure and decreasing the speed of said

compressor in response to an increase in said steam pressure.

19. A steam generator comprising a pressure-proof combustion chamber, means including a compressor for supplying to said combustion chamber a combustible charge of substantially raised pressure, and subjecting said charge to combustion to produce hot combustion gases of a predetermined pressure range, heat exchange means for holding a steam generating fluid and having heating surfaces constituting ducts connected to said chamber proportioned to apply the pressure in said chamber for driving said hot gases from said chamber through said ducts at a predetermined high velocity range to heat said fluid and generate steam, a gas turbine connected in series with said ducts to apply the energy of the gases discharged from said ducts for driving said compressor, and regulating means for decreasing the speed of said gas turbine in response to the increase of the steam pressure of said generator and for increasing the speed of said gas turbine in response to decrease of the steam pressure of said generator.

20. A steam generator comprising a pressure-proof combustion chamber, means including a compressor for supplying to said combustion chamber a combustible charge of substantially raised pressure, and subjecting said charge to combustion to produce hot combustion gases of a predetermined pressure range, heat exchange means for holding a steam generating fluid and having heating surfaces constituting ducts connected to said chamber proportioned to apply the pressure in said chamber for driving said hot gases from said chamber through said ducts at a predetermined high velocity range to heat said fluid and generate steam, a gas turbine connected in series with said ducts to apply the energy of the gases discharged from said ducts for driving said compressor with an amount of power insufficient to supply to the charge the pressure required for imparting to the gases said high velocity, auxiliary motor means supplying to said compressor additional driving power in an amount sufficient to enable the compressor to produce the pressure required for imparting to the gases the high velocity, and regulating means for decreasing the speed of said auxiliary motor means in response to the increase of the steam pressure of said generator and for decreasing the speed of said auxiliary motor means in response to increase of the steam pressure of said generator.

WALTER GUSTAV NOACK.