

[54] **METHOD AND APPARATUS FOR INCREASING POWER OUTPUT AND/OR THERMAL EFFICIENCY OF A GAS TURBINE POWER PLANT**

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[51] Int. Cl. **F02c 3/06, F02c 7/10**

[58] Field of Search. **60/39.18 R, 39.18 A, 39.18 B, 60/39.18 C, 39.02, 39.67; 415/179**

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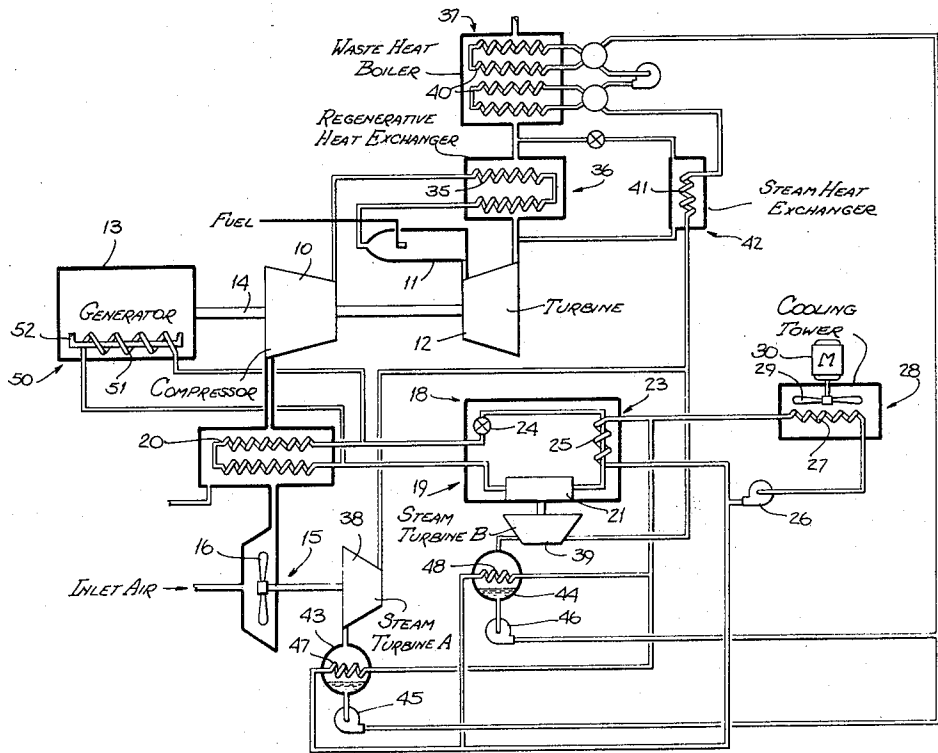
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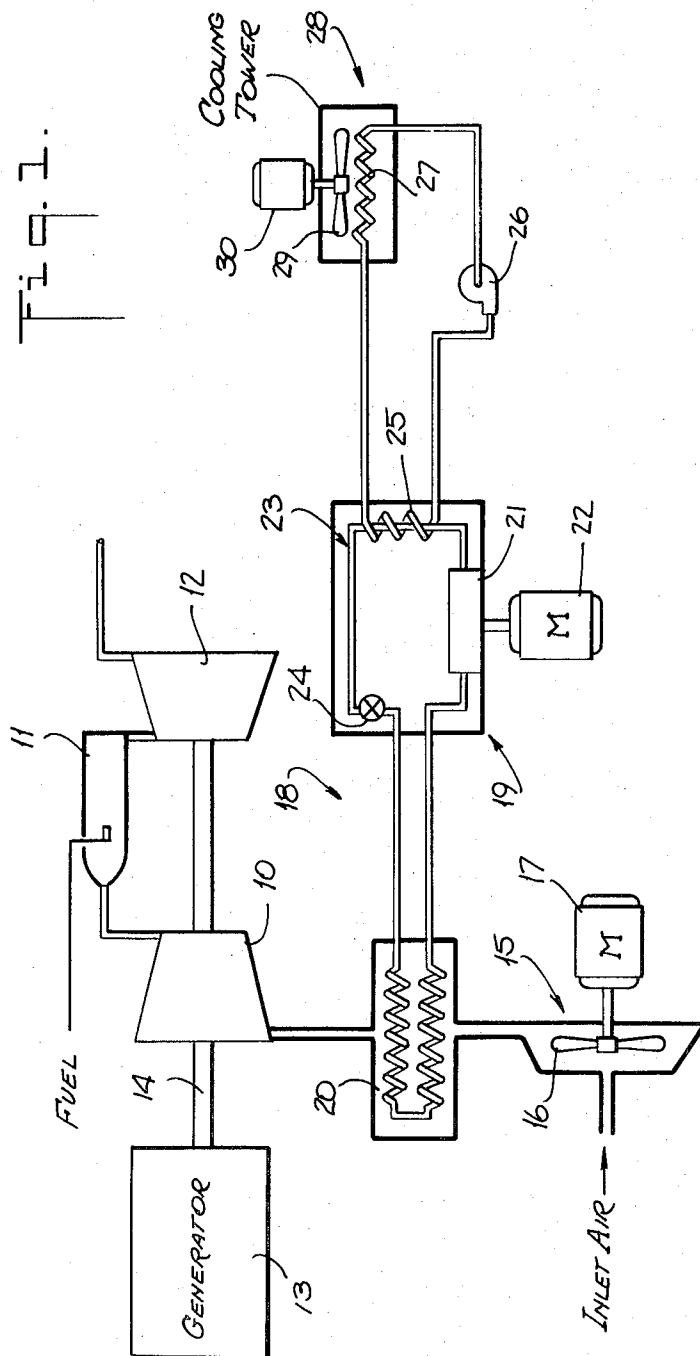
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[57] **ABSTRACT**

A gas turbine power plant having a modified gas turbine cycle (Brayton cycle) wherein the compressor inlet air is super-chilled before it enters the compressor. Superchilling, as defined herein, means to supercharge the inlet air to increase the pressure thereof to a pressure level moderately greater than the atmospheric pressure and to chill the supercharged air to decrease the temperature thereof, the preferred temperature level being in the vicinity of about 40° Fahrenheit. A heat recovery cycle is provided to supply the energy necessary to superchill the compressor inlet air.

6 Claims, 9 Drawing Figures





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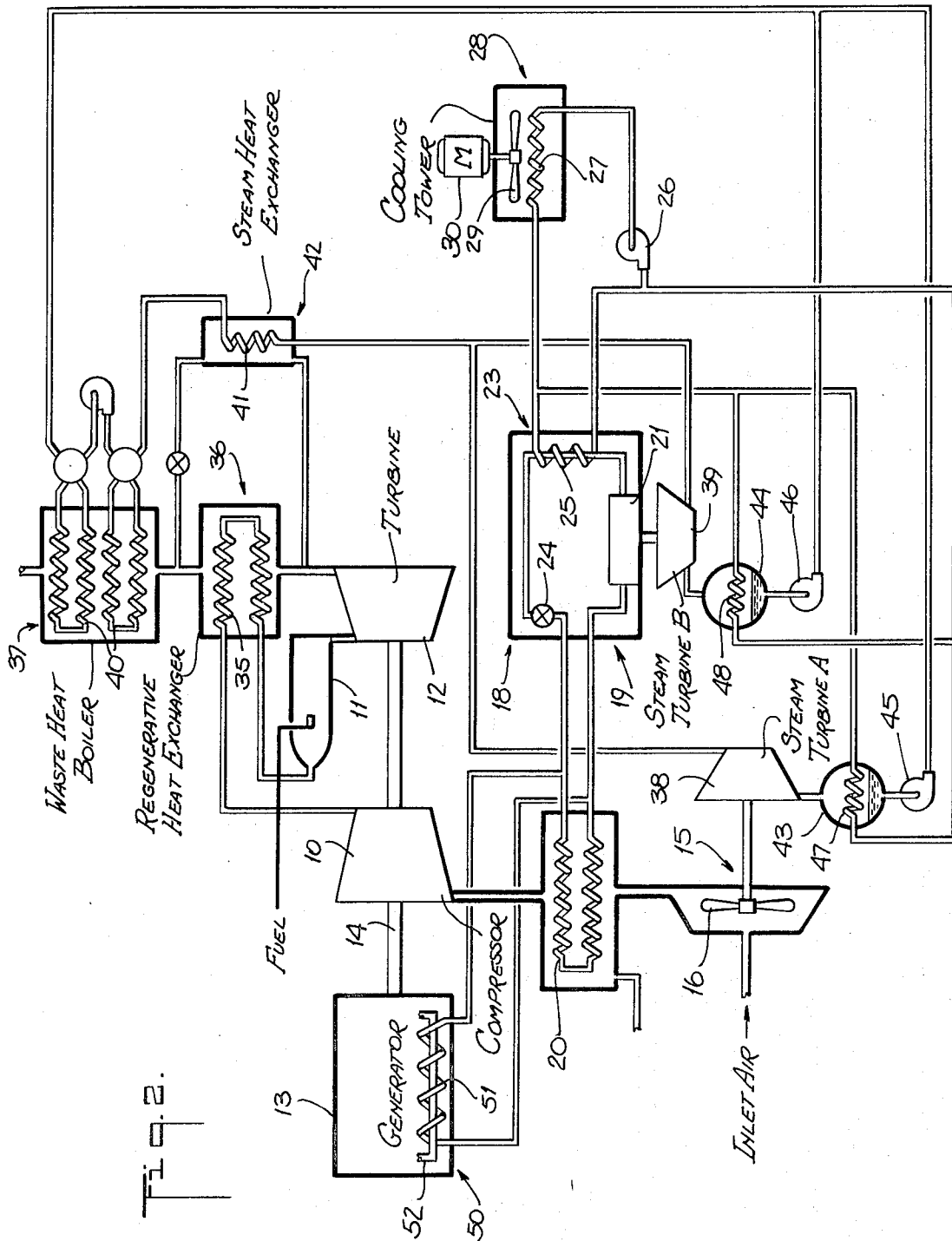
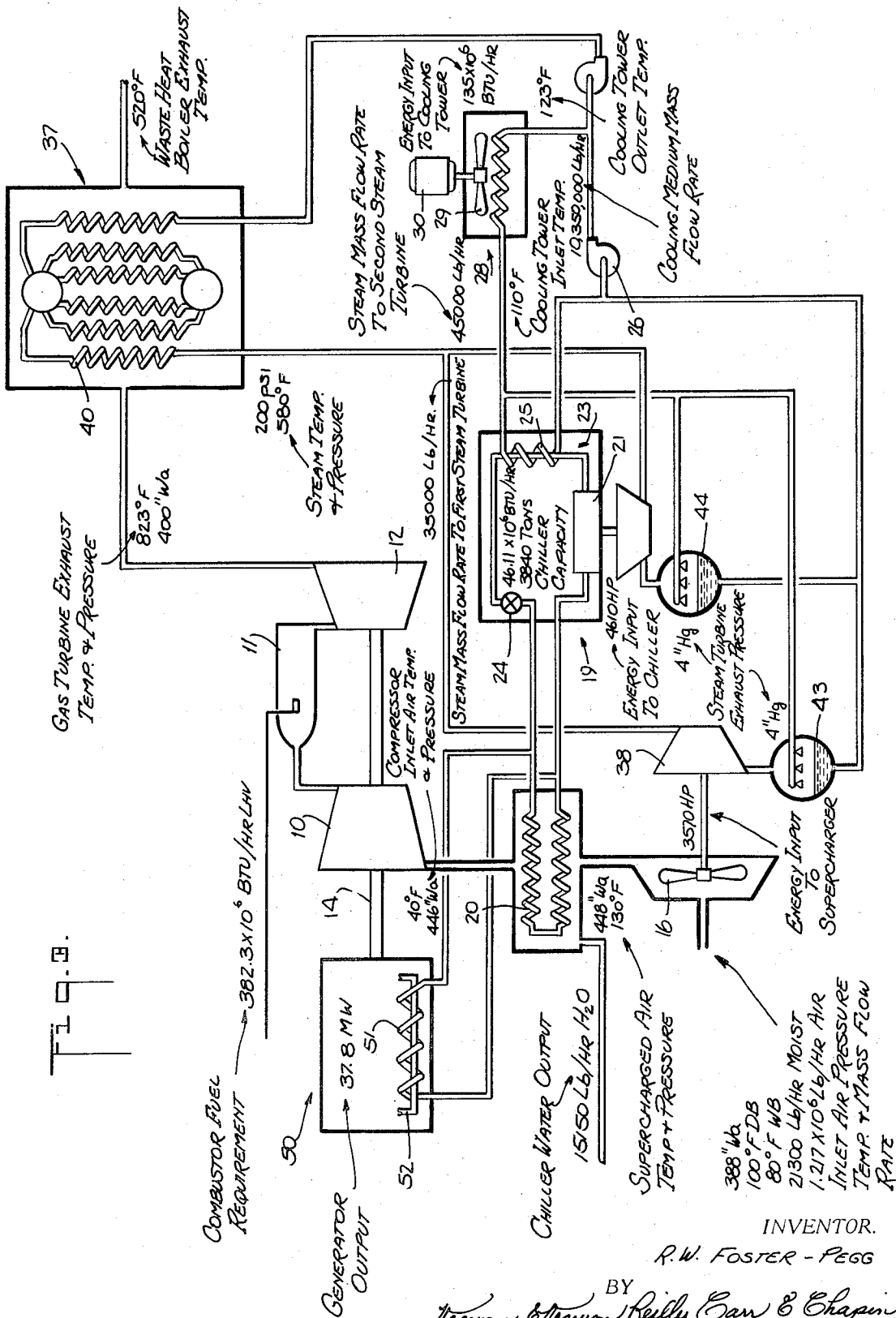


FIG. 2.

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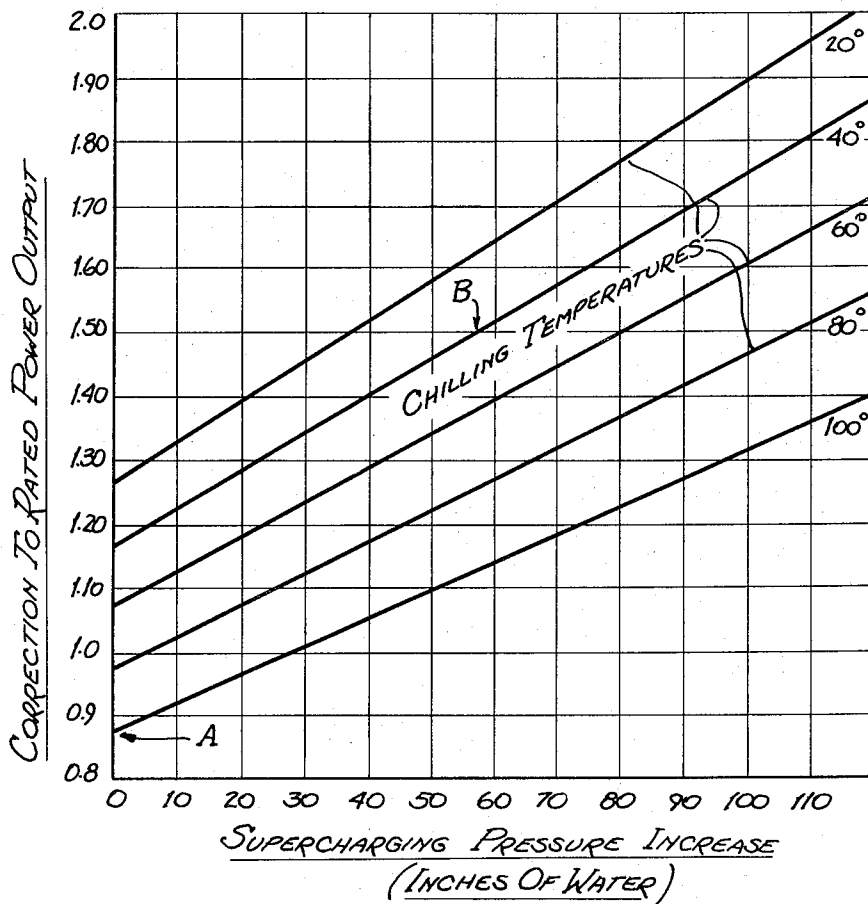


Fig. 4.

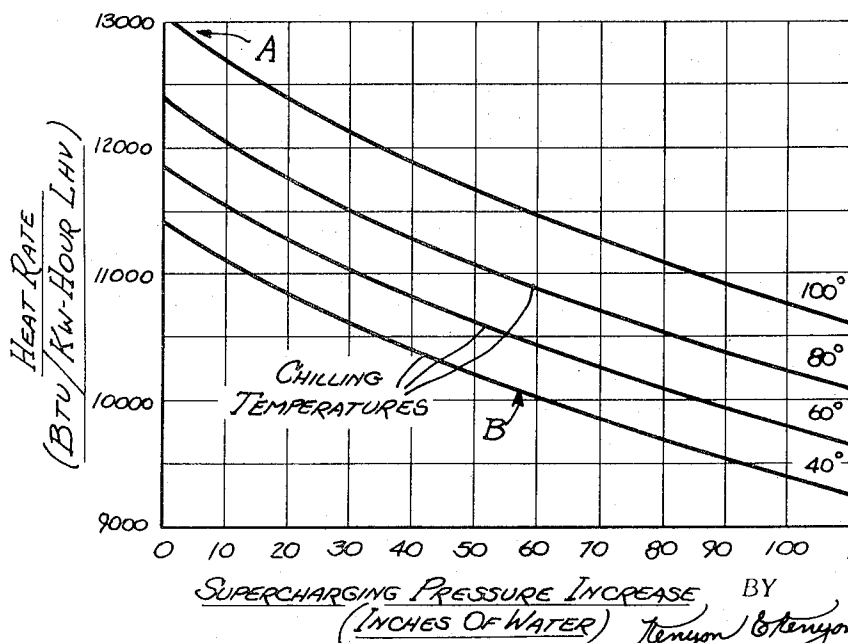


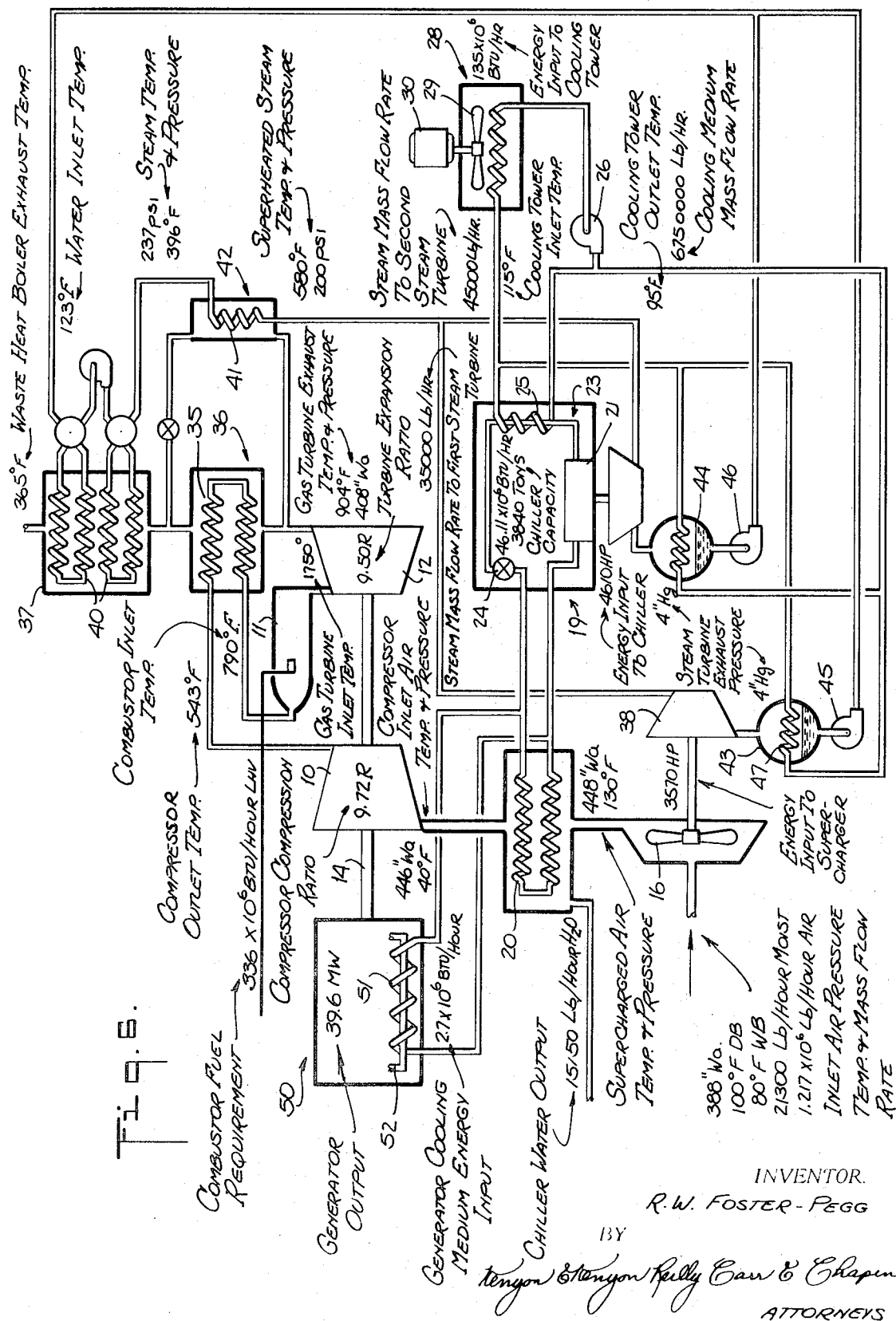
Fig. 5.

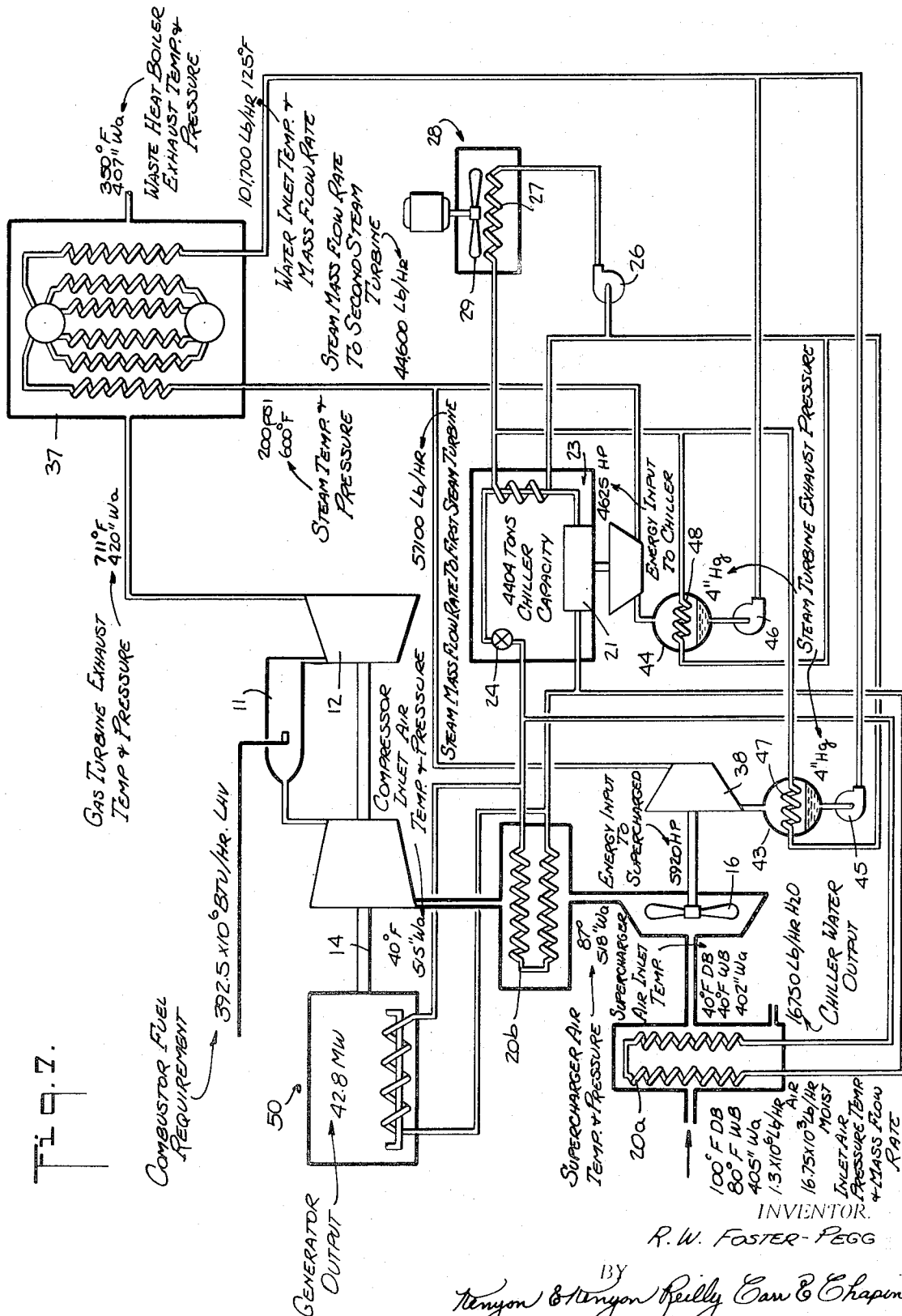
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SUPERCHARGING PRESSURE INCREASE BY
(INCHES OF WATER)

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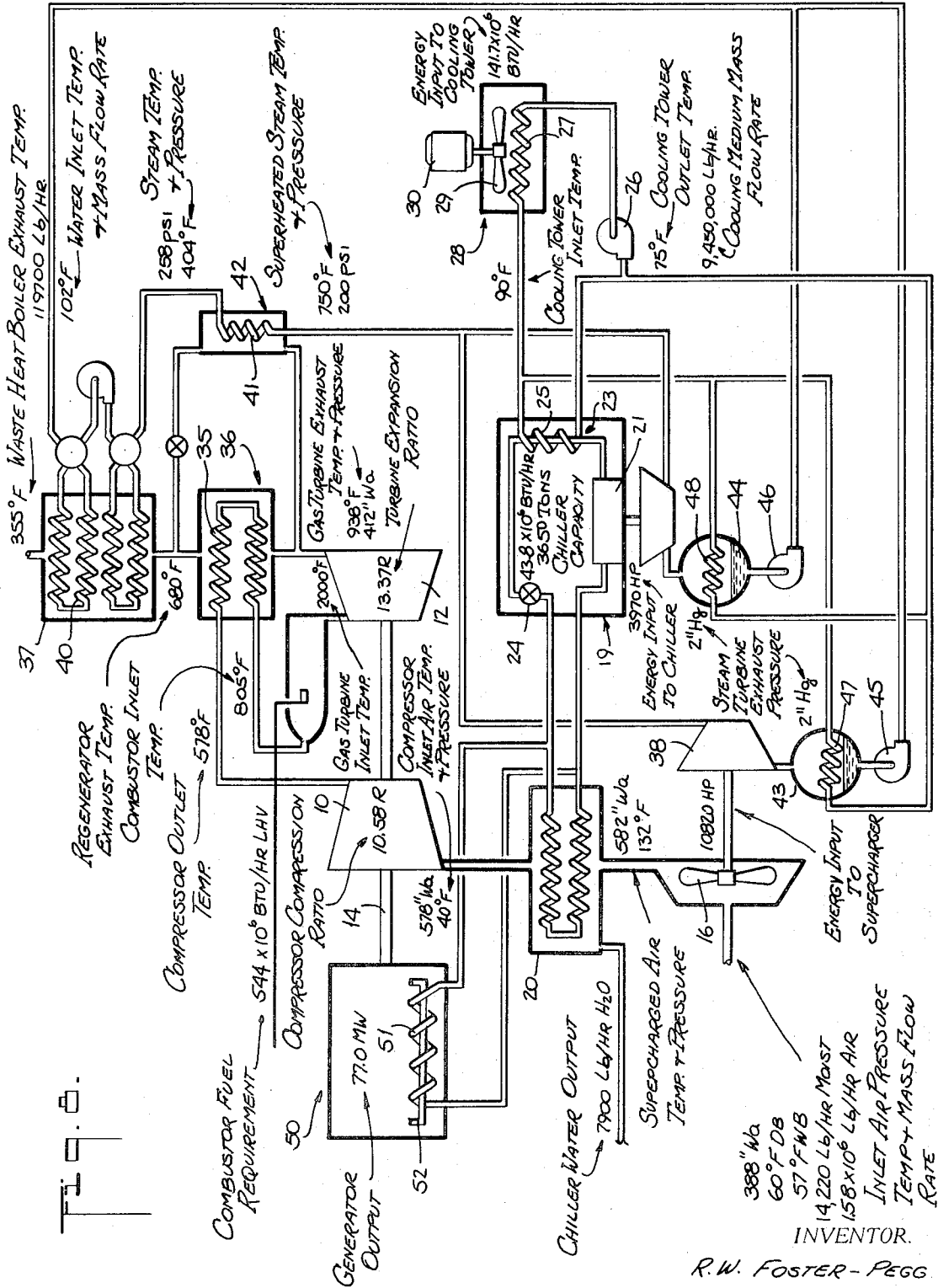


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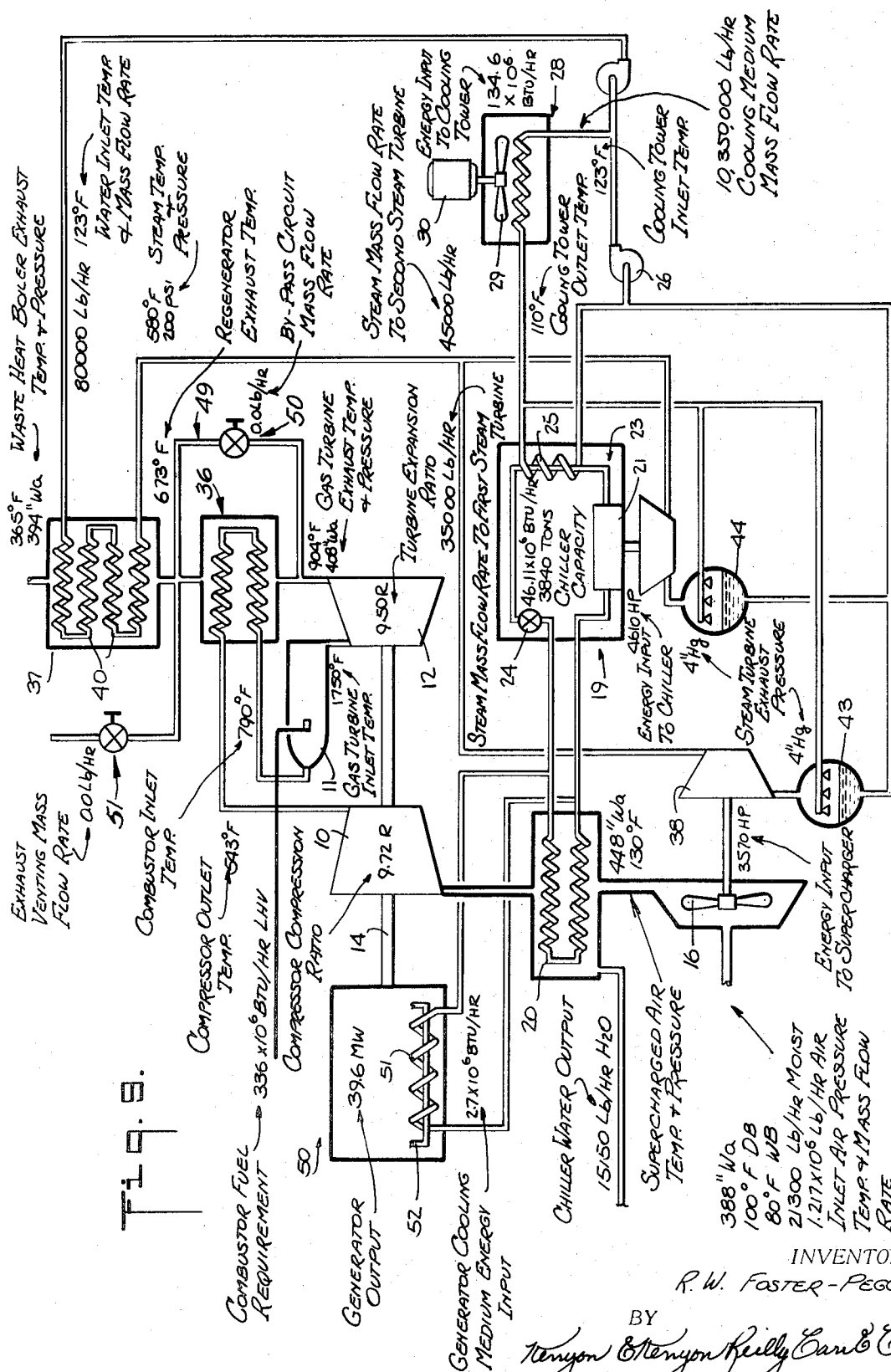
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METHOD AND APPARATUS FOR INCREASING POWER OUTPUT AND/OR THERMAL EFFICIENCY OF A GAS TURBINE POWER PLANT

BACKGROUND OF THE INVENTION

The present invention broadly relates to a modified gas turbine cycle for a gas turbine power plant. More particularly, the invention relates to a modified gas turbine cycle wherein the compressor inlet air is superchilled to increase the power output and/or the thermal efficiency of the gas turbine power plant.

Gas turbine power plants have been used for many years to generate electrical power, particularly during periods when demand for electrical power is greatest. The peak demand periods generally occur during the hottest weather when the ambient temperature of the air is high. The high temperature of the compressor inlet air at these times significantly reduces the performance of a gas turbine power plant by decreasing the power output and/or thermal efficiency of the turbine. Consequently, during periods of more moderate or normal ambient air temperatures, the power required of the stationary gas turbine may be substantially below that which the turbine is capable of producing at these conditions so that adequate capacity is available when the ambient temperature of the air is high.

Electrical utilities and gas turbine manufacturers have considerable incentives to increase the power output and/or thermal efficiency of stationary gas turbine power plants, and much effort has been expended to reap the rewards occasioned by each increase therein. Thus, stationary gas turbine power plants occasionally include various means for modifying the basic gas turbine cycle such as intercoolers, regenerators and recuperators which increase the power output and/or thermal efficiency of the gas turbine power plant. In addition, limited use has been made of supercharging the compressor inlet air and cooling the supercharged air to increase the power output of the gas turbine power plant. However, at the present time such uses extend only to supercharging with electric motor driven fans and to cooling with evaporative coolers. For example, see Foster-Pegg, R.W., "Supercharging of Gas Turbines by Forced Draft Fans with Evaporative Intercooling," American Society of Mechanical Engineers, Paper No. 65-GTP-8 (1965). Thus, the prior art does not disclose supercharging compressor inlet air with waste heat energy from the gas turbine exhaust gases. Further, chilling the compressor inlet air to low temperatures is also known. For example, see U.S. Pat. No. 2,322,717 for Apparatus For Combustion Turbines issued June 22, 1943. However, chilling of the compressor inlet air has not been adopted by electrical utilities and gas turbine manufacturers, except when a means for chilling the intake air is already available or is being installed for another purpose.

At present, no gas turbine power plant has been installed with a chilling means provided for the primary purpose of chilling the compressor inlet air. Thus, the prior art does not disclose chilling the compressor inlet air with a refrigeration system having a compressor driven by waste-heat energy from the exhaust gases of the gas turbine. Further, the prior art does not include supercharging and chilling the compressor inlet air.

Despite the incentives to increase the power output and/or thermal efficiency of gas turbine power plants and the efforts that have been expended in this regard, present gas turbine power plants generally remain uneconomical for continuous base load electrical power generation when compared to steam turbine power plants or combined steam and gas turbine power plants.

SUMMARY OF THE INVENTION

The method and apparatus of the present invention provide a gas turbine power plant having a modified gas turbine cycle wherein the compressor inlet air is superchilled prior to entering the compressor of the gas turbine. Superchilling the compressor inlet air significantly increases the power output and/or thermal efficiency of the gas turbine. Thus, the present invention reduces the energy production cost of gas turbines so that a gas turbine power plant becomes competitive with a steam turbine power plant for base load power generation.

A gas turbine power plant is provided having a basic gas turbine cycle comprising the following steps: compressing the inlet air from the atmosphere in a compressor; heating the compressed air in a combustor; and expanding the heated, compressed air through a turbine.

According to one embodiment of the present invention, the power output and/or the thermal efficiency of the basic gas turbine cycle described above are significantly improved by the additional step of superchilling the ambient inlet air before it enters the compressor of the gas turbine power plant. Superchilling, as used herein, means supercharging the inlet air to the compressor of the gas turbine to increase the pressure thereof to a pressure level moderately greater than the atmospheric pressure by means of a low pressure ratio device and chilling the supercharged inlet air to reduce the temperature thereof to a temperature at least as low as the temperature that could be obtained with an evaporative cooler cooling the supercharged air under ambient conditions then present. Chilling is accomplished by the direct transfer of heat from the supercharged inlet air to the refrigerant of a refrigeration system.

The term "refrigerant" is used herein in a broad and not a restriction sense of the word. The term "refrigerant" includes all fluids (such as liquids, vapors, and gas) to which heat from the inlet air can be transferred to chill the air. Thus, the term "refrigerant" is not limited to those liquids which produce refrigeration by their evaporation from a liquid to a gaseous under reduced pressure. By way of example, the term "refrigerant" can include liquid, such as a brine, which serves as an intermediate refrigerant between a primary refrigerant used to cool the fluid and the inlet air which is chilled by the direct transfer of heat to the fluid. Further, the term "refrigerant" includes ice which may be used to chill the inlet air directly or to cool an intermediate refrigerant such as a brine.

The compressor inlet air is preferably supercharged to increase the pressure thereof in accordance with a supercharging pressure ratio in the range of pressure ratios extending from about 1.1 to about 1.75. One preferred low pressure ratio device for increasing the pressure of the compressor inlet air is a fan device, for example, a conventional single stage, dual flow centrifugal blower. Supercharging pressure ratios above those

obtainable with a single stage fan device can be obtained by two stages of supercharging with such a fan device.

Generally speaking, since ambient air usually contains some moisture, the lower temperature limit for the chilling of the supercharged gas is a temperature in the vicinity of the temperature at which concomitant chilling of the moisture in the inlet air could form ice accumulations on heat transfer surfaces used to chill the inlet air. To avoid ice accumulations, the temperature of a heat transfer surface used to chill the inlet air should be maintained at a temperature at least as high as the freezing temperature of the moisture in the inlet air. Thus, as the chilling temperature level of the inlet air approaches the freezing temperature of the moisture in the inlet air, an extensive heat transfer surface is required to chill the air. Accordingly, a chilling temperature level in the vicinity of the range of temperatures extending from about 35 degrees Fahrenheit to about 40° Fahrenheit is preferred.

However, lower chilling temperatures are possible. For example, a means for removing the ice formed on the heat transfer surfaces can be provided thus enabling the compressor inlet air to be chilled to a temperature significantly below the preferred range of temperatures. Further, if a significant degree of moisture is not present in the compressor inlet air, the chilling temperature can also extend considerably below the preferred range of temperatures.

The energy required to chill the inlet air is increased by moisture contained in the air. Since the supercharged inlet air is generally chilled to a temperature below the dew point of the inlet air, moisture in the inlet air in excess of the saturated moisture content of the air at the chilling temperature will be condensed in the chilling means. Accordingly, under humid conditions, the total cooling requirement for the chilling means significantly exceeds the sensible heat cooling that would be required for dry air alone.

The compressor inlet air can be chilled both before and after the inlet air has been supercharged, or the inlet air can be chilled only after it has been supercharged. Although chilling both before and after supercharging can result in increased capital expenditures, it can be advantageous under certain circumstances. Initial chilling of the inlet air reduces the power required to supercharge a given inlet air mass flow rate, and thus reduces the total power required to chill the inlet air before it enters the compressor since the heat input to the inlet air caused by supercharging is reduced. The reduction of power to supercharge the inlet air results from the lower temperature of the air entering the supercharging means and from the decreased mass flow rate through the supercharging means caused by the moisture condensed from the inlet air during the initial chilling thereof. Further, the reduced power required to charge the inlet air to a given pressure permits a higher supercharge pressure to be obtained when a heat recovery cycle, to be discussed hereinafter, is provided to drive the supercharging means and the chilling means.

According to another embodiment of the present invention, a heat recovery cycle is provided to supply the energy necessary to superchill the compressor inlet air. For example, a waste-heat boiler can be provided to generate steam by utilizing the waste-heat in the turbine exhaust gases. The steam is subsequently ex-

panded through a first and a second steam turbine. The output shaft of the first steam turbine is coupled to drive the low pressure ratio device for supercharging the compressor inlet air. The output shaft of the second steam turbine is coupled to drive a compression refrigeration unit for chilling the compressor inlet air.

As will be more fully illustrated below, significant and heretofore unforeseen benefits result from superchilling the compressor inlet air. Superchilling the compressor inlet air significantly increases the air mass flow rate to the compressor of the gas turbine power plant at a fixed volume flow rate by increasing the pressure and decreasing the temperature of the inlet air. Superchilling also increases the gas turbine inlet pressure thereby increasing the expansion ratio across the gas turbine. The increased air mass flow rate through the gas turbine power plant and the increased expansion ratio across the turbine provide a significant increase in the power output of the gas turbine power plant. Further, the lower compressor inlet air temperature permits the gas turbine power plant to be operated at near optimum power output irrespective of the ambient air temperature. When the heat recovery cycle is provided to superchill the compressor inlet air, an additional significant increase in the power output results as well as an improvement in the thermal efficiency of the gas turbine.

Accordingly, it is an objective of the present invention to provide a gas turbine power plant having increased power output and/or thermal efficiency.

Another object is to increase the power output and/or the thermal efficiency of the gas turbine power plant when the ambient temperature of the air is high.

Still another object is to provide a gas turbine power plant wherein ambient inlet air is superchilled before it enters the compressor of the gas turbine for increasing the power output and/or the thermal efficiency of the turbine cycle.

A further object is to provide a gas turbine power plant wherein waste-heat in the turbine exhaust gases is utilized to supply the energy for superchilling the compressor inlet air.

A still further object is to provide a gas turbine power plant for driving an electric generator wherein the compressor inlet air is superchilled before it enters the compressor, and the electric generator cooling medium is chilled for simultaneously increasing the power output and/or the thermal efficiency of the gas turbine and the generating capacity of the electric generator.

A still further object is to provide a gas turbine power plant wherein the waste-heat in the turbine exhaust gases is utilized to supply the energy for superchilling the compressor inlet air and for chilling the electric generator cooling medium.

These and other objects and advantages of the gas turbine power plant of the present invention will become more apparent from the following description, when read in conjunction with the accompanying drawings, wherein corresponding parts of each figure have corresponding numbers.

FIG. 1 is a schematic diagram of one embodiment of the present invention wherein the compressor inlet air is supercharged and subsequently chilled before the air enters the compression stage of the gas turbine.

FIG. 2 is a schematic diagram of another embodiment of the present invention wherein the supercharger

and chiller are driven by waste-heat energy recovered from the turbine exhaust gases.

FIG. 3 is a schematic diagram of still another embodiment of the present invention showing selected operating characteristics for a complete gas turbine cycle adjacent the individual components.

FIG. 4 is a graph showing the power output of the embodiment of the gas turbine power plant illustrated in FIG. 3 as a function of the degree of superchilling of the compressor inlet air.

FIG. 5 is a graph showing the heat rate of the embodiment of the gas turbine power plant illustrated in FIG. 3 as a function of the degree of superchilling of the compressor inlet air.

FIG. 6 is a schematic diagram of the embodiment of the gas turbine power plant of FIG. 2 showing selected operating characteristics for a complete gas turbine cycle adjacent the individual components.

FIG. 7 is a schematic diagram of still another embodiment of the present invention wherein the compressor inlet air is chilled before and after it is supercharged.

FIG. 8 is a schematic diagram of the embodiment of the gas turbine power plant illustrated in FIG. 2 showing a second set of selected operating characteristics for a complete gas turbine cycle adjacent the individual components.

FIG. 9 is a schematic diagram of still another embodiment of the present invention showing selected operating characteristics for a complete gas turbine cycle adjacent the individual components.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to the drawings, one preferred embodiment of the improved gas turbine power plant is schematically illustrated in FIG. 1. The improvement of the present invention is schematically shown in conjunction with a conventional open-cycle, single shaft gas turbine power plant. The gas turbine power plant comprises a compressor 10 for compressing the inlet air from the atmosphere, a combustor 11 for heating the compressed air, and a turbine 12 for expanding the heated, compressed air. The turbine 12 is operably coupled to drive the compressor 10 and an electric generator 13 by means of shaft 14.

According to the present invention, the power output and/or the thermal efficiency of the gas turbine described above are significantly increased by superchilling the inlet air before it enters the compressor of the gas turbine.

Thus, referring to FIG. 1, a supercharging means 15 is provided which comprises a low pressure ratio device, conveniently shown as a fan 16, driven by an electric motor 17. Inlet air is drawn through the fan 16 thereby increasing the pressure thereof to a pressure level moderately greater than the atmospheric pressure. Once the inlet air has been supercharged, it is ducted to a chilling means 18.

The chilling means 18 is conveniently shown as a compression refrigeration unit 19 which comprises an evaporator coil 20 in which a liquid refrigerant boils at a low temperature, a compressor 21 driven by an electric motor 22 for raising the pressure and temperature of the gaseous refrigerant from the evaporator coil 20, a condenser 23 in which the refrigerant from the compressor 21 discharges heat to a secondary cooling medium such as water, and an expansion valve 24 for ex-

panding the liquid refrigerant from the high pressure level in the condenser 21 to the low pressure level in the evaporator coil 20. The supercharged inlet air is ducted across the evaporator coil 20 where the air is chilled as heat from the air is transferred to the expanded gaseous refrigerant therein.

The secondary cooling medium is circulated through the coil 25 of the condenser 23 by a circulating pump 26 where the gaseous refrigerant condenses to a liquid and releases heat to the cooling medium. The cooling medium subsequently circulates through a cooling coil 27 of a cooling tower 28 where the cooling medium discharges heat to air circulated across the cooling coil 27 by a cooling fan 29, driven by an electric motor 30.

To illustrate the increased performance of a gas turbine power plant obtained by superchilling the compressor inlet air, the power output and thermal efficiency of various embodiments of the gas turbine power plant are described below. The ideal gas turbine power plant assumed for some of the comparisons is a 25 megawatt gas turbine power plant having an ideal gas turbine cycle (Brayton Cycle) rated at a compressor inlet temperature of about 80°F. and a compressor inlet pressure of about 392 inches of water which corresponds to the pressure at about a 1,000 foot elevation. The ideal cycle has no inlet or exhaust pressure losses. Assuming a compressor inlet mass flow rate of about 1×10^6 pounds of air per hour, a combustor fuel requirement of about 300×10^6 Btu/Hr LHV (Lower Heating Value), and a gas turbine exhaust temperature and pressure of about 895°F. and about 392 inches of water, respectively, the ideal gas turbine power plant would produce about 25 megawatts (rated power output) at a heat rate of about 12,000 Btu/Kwh LHV.

The performance of the ideal 25 megawatt gas turbine power plant is significantly reduced when the gas turbine is operated under warm weather conditions. If the ideal gas turbine were operated at high ambient inlet air temperatures of about 100°F. dry bulb and 80°F. wet bulb at about a 1,000 foot elevation with an inlet pressure loss of about 2 inches of water and an exhaust pressure loss of about 4 inches of water, the ideal gas turbine would produce about 87.5 per cent of the rated power output (21.6 megawatts) at a heat rate of about 13,000 Btu/Kwh LHV. The power output and heat rate calculations are based upon a compressor mass flow rate of about 0.954×10^6 pounds of air per hour, a combustor fuel requirement of about 281.7×10^6 Btu/hr LHV, and a turbine exhaust pressure of about 396 inches of water. The ideal 25 megawatt gas turbine having the above assumed inlet and exhaust pressure losses will hereinafter be referred to as the standard gas turbine.

Superchilling the compressor inlet air of the standard gas turbine described above, as shown in FIG. 1, significantly increases the power output and/or thermal efficiency of the gas turbine. For example, assume the following operating conditions: inlet air temperatures, pressure and mass flow rates of about 100°F. dry bulb, about 80°F. wet bulb, about 390 inches of water, about 1.223×10^6 pounds of air per hour and about 21.4×10^3 pounds of water per hour; a combustor fuel requirement of about 382.3×10^6 Btu/hr LHV; and a turbine exhaust temperature of about 823°F. Supercharging the compressor inlet air of the standard gas turbine to moderately raise the pressure thereof from about 390 inches of water to about 448 inches of water with

a motor driven fan 16, and chilling the supercharged compressor inlet air to about 40°F. with a motor driven compression refrigeration unit 19 increases the net power output of the standard gas turbine to about 125 per cent of rated power output (31.17 megawatts) at a heat rate of about 12,300 Btu/Kwh LHV. The motor driven fan 16 would require about 2,750 kilowatts to supercharge the inlet air. The refrigeration unit 19 would require about 3,892 kilowatts to chill the supercharged air. Thus, of the about 37.8 megawatts of electrical power generated by the standard gas turbine, about 6,642 kilowatts are consumed by the superchilling thereby providing a net power output of about 31.17 megawatts.

Another preferred embodiment of the present invention is schematically illustrated in FIG. 2. As described above, the supercharging means 15 and the chilling means 18 of the present invention are shown in conjunction with a conventional open cycle, single shaft gas turbine power plant. However, in FIG. 2, the compressed air from the compressor 10 is heated in a regenerator by waste heat in the turbine exhaust gases. Thus, the compressed air is ducted through the coil 35 of a regenerative heat exchanger 36 before it enters the combustor 11. Most of the turbine exhaust gas is ducted across the heat exchanger coil 35 for heating the compressed air passing therethrough.

According to the present invention, the power output and/or thermal efficiency of the gas turbine power plant are still more significantly increased by the addition of a heat recovery cycle wherein residual waste-heat in the turbine exhaust gases is recovered and converted into mechanical energy for driving the supercharging means 15 and the chilling means 18. Thus, referring to FIG. 2, a closed steam cycle is provided wherein a waste-heat boiler 37 generates steam from the residual waste-heat in the turbine exhaust gases. The steam generated thereby is expanded through a first steam turbine 38, operably coupled to drive the supercharging means 15, and a second steam turbine 39, operably coupled to drive the chilling means 18.

The turbine exhaust gases from the regenerative heat exchanger 36 are ducted through the waste-heat boiler 37. The residual waste-heat in the exhaust gas generates steam from water pumped through the coils 40 of the waste-heat boiler 37. The steam generated thereby is subsequently circulated through the coil 41 of a steam heat exchanger 42 where the waste-heat in the remainder of the turbine exhaust gases superheats the steam. The remainder of the exhaust gases is then ducted through the waste-heat boiler 37 to supplement the waste-heating by the turbine exhaust gases from the regenerative heater 36.

A portion of the superheated steam generated by the waste-heat boiler is expanded in the first steam turbine 38 which is operably coupled to drive the fan 16 of the supercharging means 15. The remainder of the superheated steam is expanded in the second steam turbine 39 which is operably coupled to drive the compressor 21 of the compression cycle refrigeration unit 19. The steam discharged from the steam turbines 38 and 39 is condensed in condensers 43 and 44 and is recycled to the waste-heat boiler 37 by return pumps 45 and 46. The cooling medium for the condensers 43 and 44 is conveniently provided from the secondary cooling medium for the condenser coil 23. Thus, the circulating pump 25 also circulates the cooling medium from the

cooling coil 26 through the condenser coil 23 of the refrigeration unit 19 and through the coils 47 and 48 of the condensers 43 and 44.

The performance of the gas turbine power plant is still further improved when the electric generating capacity of the electric generator is increased to complement the increased shaft output of the gas turbine power plant. An electric generator cooling means 50 is provided to chill the generator cooling medium. The cooling means 50 comprises a generator cooling coil 51 disposed within the electric generator 13 in a heat-transfer relationship with the generator cooling circuit 52. The liquid refrigerant from the chilling means 18 is circulated through the coil 51 to substantially chill the generator cooling medium flowing in the circuit 52. As illustrated in FIG. 3, the generator cooling coil 51 and the evaporator coil 20 are connected in parallel between the expansion valve 24 and the compressor 21. The liquid refrigerant expands through the expansion valve 24 and is circulated through the generator cooling coil 51 where the refrigerant boils to chill the generator cooling medium.

As noted above, superchilling the compressor inlet air according to the present invention significantly increases the power output and/or thermal efficiency of a gas turbine power plant. For example, another embodiment of the present invention is illustrated in FIG. 3. The embodiment of the gas turbine power plant of FIG. 3 is similar to the gas turbine power plant illustrated in FIG. 2, however, the compressed air from the compressor 10 is not regeneratively heated by a regenerative heat exchanger 36.

The operating characteristics of the gas turbine power plant listed in FIG. 3 are calculated on the basis of a compressor inlet pressure loss of about 2 inches of water and a turbine exhaust pressure loss of about 4 inches of water. An additional inlet pressure loss of about 2 inches of water is assumed for the chilling means 19 and an additional exhaust pressure loss of about 4 inches of water is assumed for the waste heat boiler 37. Thus, the gas turbine of FIG. 3 having the compressor inlet air supercharged to increase the compressor inlet pressure by about 58 inches of water and chilled to reduce the temperature of the compressor inlet air to about 40°F. would produce about 151 per cent of the rated power output (37.8 megawatts) at a heat rate of about 10,130 Btu/Kwh LHV.

Now, referring to FIGS. 4 and 5, the performance of the standard gas turbine power plant described above is compared with the performance of the gas turbine power plant of FIG. 3 under varying levels of supercharging and chilling. The performances are compared for high ambient inlet air temperatures of about 100°F. dry bulb and about 80°F. web bulb at about a 1,000 foot elevation. An inlet pressure loss of about 2 inches of water and an exhaust pressure loss of about 4 inches of water are assumed for the standard gas turbine power plant. An additional inlet pressure loss of about 2 inches of water is assumed for the chilling means and an additional exhaust pressure loss of about 4 inches of water is assumed for the heat recovery cycle.

The performance of the standard gas turbine power plant is represented in FIGS. 3 and 4 by the points marked A on the 100°F. line (no chilling) corresponding to zero pressure increase (no supercharging). As indicated therein, the standard gas turbine power plant would produce about 87.5 per cent of the rated power

output (21.6 megawatts) at a heat rate of about 13,000 Btu/Kwh LHV. In comparison, the performance of the gas turbine power plant of FIG. 3 is indicated by the points marked B.

The increase in power output and/or thermal efficiency of a gas turbine power plant having superchilled compressor inlet air is still more significant when the gas turbine cycle includes regenerative heating of the compressor outlet air, as shown in FIG. 6. The embodiment of the present invention illustrated in FIG. 6 is the same as the embodiment illustrated in FIG. 2. Referring to FIG. 6, selected characteristics of the gas turbine power plant for one set of operating conditions are listed adjacent the individual components thereof. The same ambient conditions and pressure losses assumed for the calculations presented in FIGS. 4 and 5 were applied to the calculations for FIG. 6. The power output for the superchilled gas turbine power plant of FIG. 6 is about 39.6 megawatts at a heat rate about 8,480 Btu/Kwh LHV.

By way of comparison, the power output of a conventional regenerative gas turbine power plant would be about 26.1 megawatts at a heat rate of about 9,850 Btu/Kwh LHV. These calculations are based upon the following conditions: air temperature, pressure and mass flow rates of about 80°F., about 388 inches of water, and about 0.96×10^6 pounds of air per hour and about 13.3×10^3 pounds of water per hour, respectively; a compressor compression ratio of about 9.0 and a turbine expansion ratio of about 7.8; a compressor outlet temperature of about 543°F., a combustor inlet temperature of about 839°F. and a turbine inlet temperature of about 1750°F. a combustor fuel requirement of about 257.3×10^6 Btu/Hr LHV; gas turbine exhaust temperature and pressure of about 963°F. and about 404 inches of water, respectively; and regenerator exhaust temperature and pressure of about 743°F. and about 396 inches of water, respectively.

As indicated above, the compressor inlet air can also be chilled both before and after the air is supercharged. An embodiment of the gas turbine power plant having such dual chilling is illustrated in FIG. 7. The chilling means 18 comprises a compression refrigeration unit 19, as described above with respect to FIG. 1, but having a first evaporator coil 20a and a second evaporator coil 20b. The inlet air is initially drawn across the first evaporator coil 20a where the air is chilled as heat from the air is transferred to the coil. The inlet air is next drawn through the fan 16 where the air is supercharged. The supercharged inlet air which has been heated by the work applied to the air by the fan 16 is again chilled as it is ducted across the second evaporator coil 20b before it enters the compressor 10.

Selected characteristics for the gas turbine power plant for one set of operating conditions are listed in FIG. 7. The power output for the gas turbine is about 42.8 megawatts at a heat rate of about 9,180 Btu/Kwh LHV.

A further example of the significant increase in thermal efficiency and/or power output obtained through superchilling is illustrated in FIG. 8. The embodiment of the gas turbine power plant illustrated therein is similar to the embodiment of FIG. 2; however, the individual components are considerably larger to accommodate the increased air mass flow rate necessary to generate sufficient power to drive the larger generator 13. Although the same pressure losses are assumed for the

calculations listed in FIG. 8, it will be noted that the assumed ambient air temperatures are lower than the ambient air temperatures assumed above. At these conditions, the superchilled gas turbine power plant would produce about 77.0 megawatts of power at a heat rate of about 7,060 Btu/Kwh LHV.

The heat rate of the superchilled gas turbine power plant of FIG. 8 is remarkably low for a gas turbine power plant and it is highly competitive with the heat rates obtained with steam turbine power plants.

Superchilling the compressor inlet air with energy provided by a heat recovery cycle and regeneratively heating the compressed air before it enters the combustor can utilize substantially all of the waste-heat in the turbine exhaust gases. As a practical matter, maximum regenerator efficiency would commonly be on the order of about seventy-five per cent so that some waste-heat would always be available for the heat recovery cycle. The waste-heat boiler can include an economizer and/or a combination low pressure boiler and deaerator.

Maximum efficiency of superchilling will generally occur at maximum regenerator efficiency with as much of the residual waste-heat leaving the regenerator being utilized for the heat recovery cycle. The ability of superchilling to operate at maximum efficiency can be coupled with high power capability by selectively increasing the energy input to the recuperative cycle thereby increasing the degree of superchilling. Thus, referring to FIG. 9, still another embodiment of the present invention is illustrated wherein a regenerator bypass circuit 49 is provided to duct the turbine exhaust gases around the regenerator 36 and directly into the waste-heat boiler 37 of the heat recovery cycle thereby increasing the energy available for superchilling the inlet air. An adjustable bypass valve 50 in the circuit 49 permits the power output and/or thermal efficiency of the gas turbine power plant to be modulated in a controlled manner.

Power output of the gas turbine power plant below the point of maximum efficiency can also be effectively modulated. Preferably, this modulation would be accomplished by reducing the degree of supercharging, for example, throttling the first steam turbine 38 coupled to drive the fan 16, while maintaining the same degree of chilling. Still lower power outputs could be obtained by controlled exhausting to the atmosphere of the turbine exhaust gases from the regenerator 36 to reduce the energy input to the heat recovery cycle thus reducing the degree of chilling. With complete exhausting of the turbine exhaust gases, the gas turbine power plant operates in the conventional manner without any supercharging or chilling of the compressor inlet air.

Accordingly, referring again to FIG. 9, a second adjustable bypass valve 51 is provided to permit selective venting to the atmosphere of the turbine exhaust gases from the regenerative heat exchanger 36 and/or the bypass circuit 49. Adjustment of the second bypass valve 51 provides selective reduction of the energy available to the heat recovery cycle which in turn reduces the degree of superchilling of the compressor inlet air.

By way of further example, the operating characteristics for the embodiment of the gas turbine power plant of FIG. 9 are listed therein. The superchilled gas turbine power plant would produce about 39.6 megawatts of power at a heat rate of about 8480 Btu/Kwh LHV.

Other means are available to supercharge the ambient inlet air. For example, fan 16 can be replaced by a moderately low pressure rise compressor or blower. Similarly, other means are also available to chill the inlet air. For example, the compression refrigeration unit 19 can be replaced by an absorption refrigeration unit. Generally speaking, in an absorption refrigeration unit, the compressor 21 and motor 22 in the compression refrigeration unit would be replaced by an absorber, a generator, a pump, a heat exchanger and a reducing valve. Waste-heat in the turbine exhaust gases would provide the heat input to the absorption refrigeration unit.

Although the gas turbine power plant of the present invention has been described as a power source for an electrical generator, it is to be understood that the improved gas turbine power plant has other applications. For example, the gas turbine power plant has application as a natural gas pipeline compressor drive.

The embodiments of the gas turbine power plant described above are for the purpose of illustrating the broader aspects of the present invention, and the advantages attendant therein. Other modifications and variations of the embodiments will be apparent to those skilled in the art, and they may be made without departing from the spirit and scope of the present invention.

What is claimed is:

1. An improved gas turbine having increased performance, the gas turbine including a compressor for receiving inlet gas and compressing the same, means for heating the compressed gas, and a turbine for expanding the heated compressed gas, the improvement comprising:

- a. means for supercharging the inlet gas before it is received by the compressor; and
- b. means for chilling the supercharged inlet gas before it is received by the compressor, the chilling means including a refrigerant for the direct transfer of heat from the supercharged gas thereto; and
- c. means for regeneratively heating the compressed gas from the compressor with waste-heat from the exhaust gases of the turbine before the compressed gas passes to the compressed gas heating means; and
- d. means for recovering a portion of the waste-heat energy from the exhaust gases of the turbine and for converting the waste-heat energy into energy for driving the supercharging means and for driving the chilling means; and
- e. means for selectively controlling the portion of the waste-heat energy converted to energy for driving the supercharging means and for driving the chilling means for providing selective control over the amount of energy available to superchill the inlet gas thereby providing for selective control of the performance of the gas turbine.

2. An improved gas turbine having increased performance, the gas turbine including a compressor for receiving inlet gas and compressing the same, means for heating the compressed gas, and a turbine for expanding the heated compressed gas, the improvement comprising:

- a. means for supercharging the inlet gas before it is received by the compressor; and
- b. means for chilling the supercharged inlet gas before it is received by the compressor, the chilling

means including a refrigerant for the direct transfer of heat from the supercharged gas thereto; and

c. means for recovering a portion of the waste-heat energy from the exhaust gases of the turbine and for converting the waste-heat energy into energy for driving the supercharging means and for driving the chilling means; and

d. means for selectively controlling the portion of the waste-heat energy converted to energy for driving the supercharging means and for driving the chilling means for providing selective control over the amount of energy available to superchill the inlet gas thereby providing for selective control of the performance of the gas turbine.

3. An improved gas turbine having increased performance, the gas turbine including a compressor for receiving inlet gas and compressing the same, means for heating the compressed gas, and a turbine for expanding the heated compressed gas, the improvement comprising:

- a. means for supercharging the inlet gas before it is received by the compressor; and
- b. means for chilling the supercharged inlet gas before it is received by the compressor; and
- c. means for recovering a portion of the waste-heat energy from the exhaust gases of the turbine and for converting the waste-heat energy into energy for driving the supercharging means and for driving the chilling means; and
- d. means for selectively controlling the portion of the waste-heat energy converted to energy for driving the supercharging means and for driving the chilling means for providing selective control over the amount of energy available to superchill the inlet gas thereby providing for selective control of the performance of the gas turbine.

4. An improved gas turbine according to claim 3, further comprising means for regeneratively heating the compressed gas from the compressor with waste-heat from the exhaust gases of the turbine before the compressed gas passes to the compressed gas heating means.

5. A method for increasing the performance of a gas turbine, the gas turbine including a compressor for receiving inlet gas and compressing the same, means for heating the compressed gas, and a turbine for expanding the heated compressed gas to extract work therefrom, the method comprising the steps of:

- a. supercharging the inlet gas by a fan or a blower device before it is received by the compressor by increasing the pressure of the inlet gas in accordance with a supercharging pressure ratio in a range of pressure ratios extending from about 1.1 to about 1.75;
- b. chilling the supercharged inlet gas before it is received by the compressor by the direct transfer of heat from the supercharged gas to the refrigerant of a refrigeration system;
- c. regeneratively heating the compressed gas from the compressor with waste-heat from the exhaust gases of the turbine;
- d. recovering a portion of the waste-heat energy from the exhaust gases of the turbine and converting the waste-heat energy into energy for superchilling the compressor inlet air; and
- e. selectively controlling the portion of the waste-heat energy converted to energy for superchilling

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the compressor inlet air thereby selectively controlling the performance of the gas turbine.

6. An improved gas turbine having increased performance, the gas turbine including a compressor for receiving inlet gas and compressing the same, means for heating the compressed gas, and a turbine for expanding the heated compressed gas, the improvement comprising:

- a. fan or blower means for supercharging the inlet gas before it is received by the compressor by increasing the pressure of the inlet gas in accordance with a supercharging pressure ratio in a range of pressure ratios extending from about 1.1 to about 1.75;
- b. means for chilling the supercharged inlet gas before it is received by the compressor, the chilling means including a refrigerant for the direct transfer of heat from the supercharged gas thereto;

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- c. means for regeneratively heating the compressed gas from the compressor with waste-heat from the exhaust gases of the turbine before the compressed gas passes to the compressed gas heating means;
- d. means for recovering a portion of the waste-heat energy from the exhaust gases of the turbine and for converting the waste-heat energy into energy for driving the supercharging means and for driving the chilling means; and
- e. means for selectively controlling the portion of the waste-heat energy converted to energy for driving the supercharging means and for driving the chilling means for providing selective control over the amount of energy available to superchill the inlet gas thereby providing for selective control of the performance of the gas turbine.

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