United States Patent [19]

Davoud

[11] **3,716,990**

[45] **Feb. 20, 1973**

[54] CONDENSABLE VAPOR POWER PRODUCING SYSTEM

[75] Inventor: **John Gordon Davoud,** Richmond, Va.

[73] Assignee: **Edwin Cox Associates,** Richmond, Va.

[22] Filed: **May 12, 1971**

[21] Appl. No.: **142,468**

[52] **U.S. Cl.**.....**60/94,** 60/36

[51] **Int. Cl.**.....**F01k 25/00,** F01k 23/00

[58] **Field of Search**.....60/36, 39, 40, 64, 65, 93, 60/94

[56] **References Cited**

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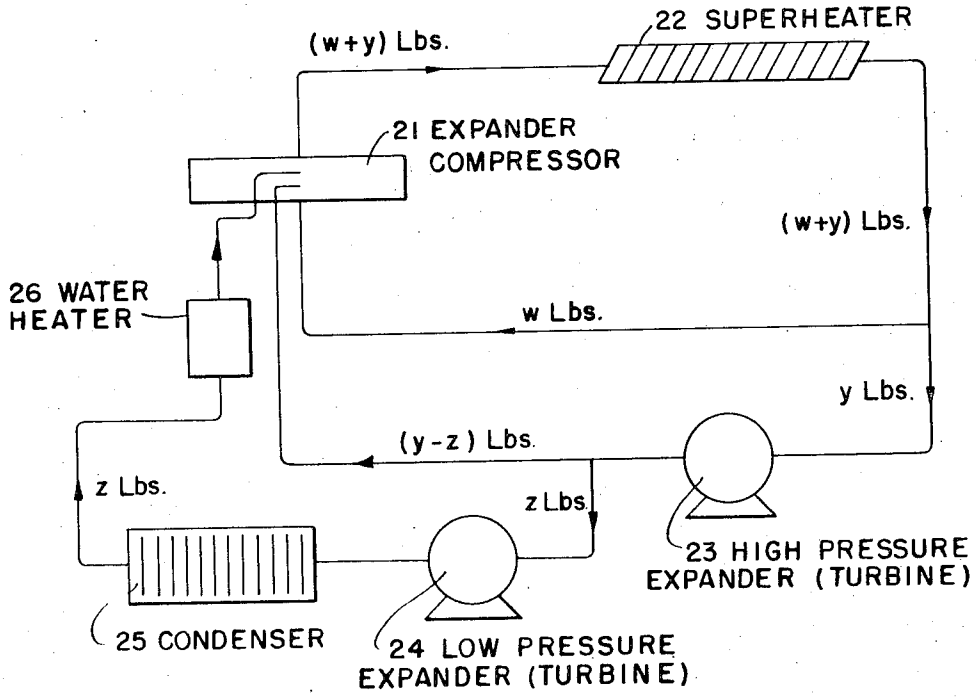
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Primary Examiner—Martin P. Schwadron
Assistant Examiner—Allen M. Ostrager
Attorney—Stowell & Stowell

[57] **ABSTRACT**

Steam or other condensable vapor heated to a predetermined temperature at a predetermined pressure and having a given entropy is expanded in part in a work producing zone to a lower pressure and then condensed while the remaining part is expanded in a second zone; and following expansion, water is added thereto, with or without addition, or following withdrawal, of a portion of steam at the same state as the second portion of steam after expansion, to form a weight of steam greater than that introduced into the second zone but at a lower entropy, using the work of expansion in the second zone to compress the steam expanded in the second zone, plus the further portion of added steam, when such addition of steam is made, or minus the portion removed, when such removal is made, plus the added water, to the original predetermined pressure.

24 Claims, 19 Drawing Figures



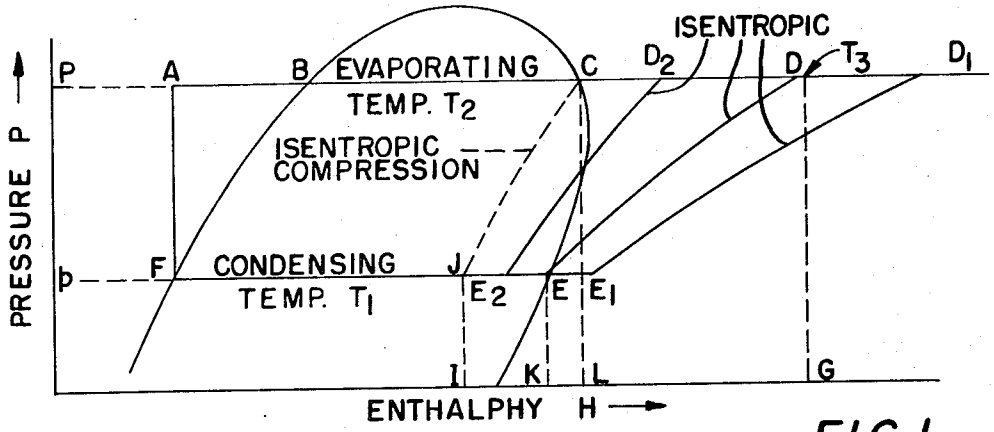


FIG. 1.

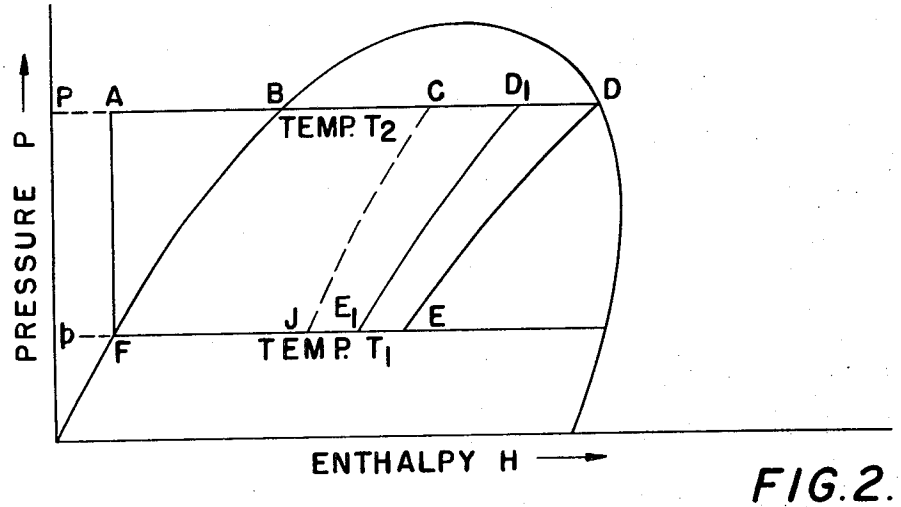


FIG. 2.

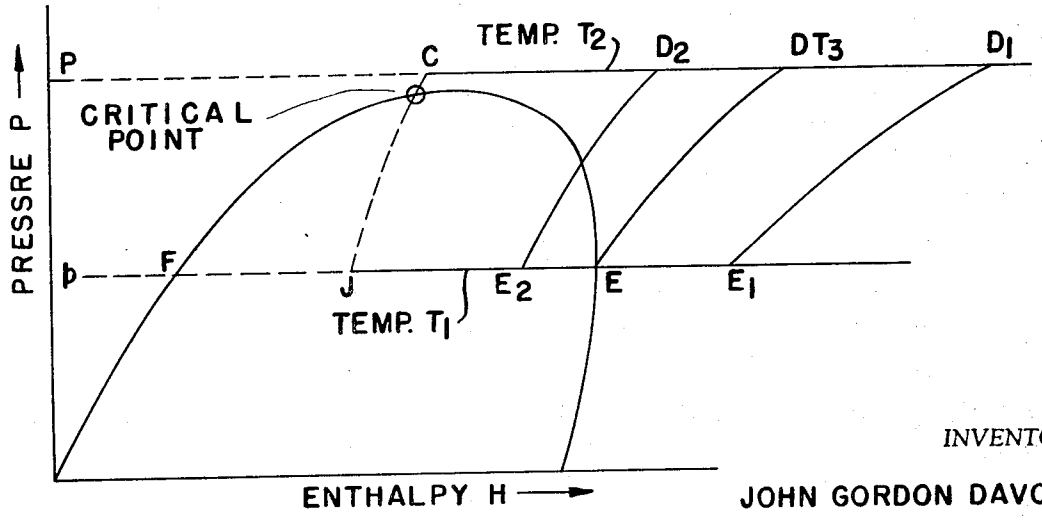


FIG. 3.

INVENTOR
JOHN GORDON DAVOUD

BY Stevens & Stevens

ATTORNEY

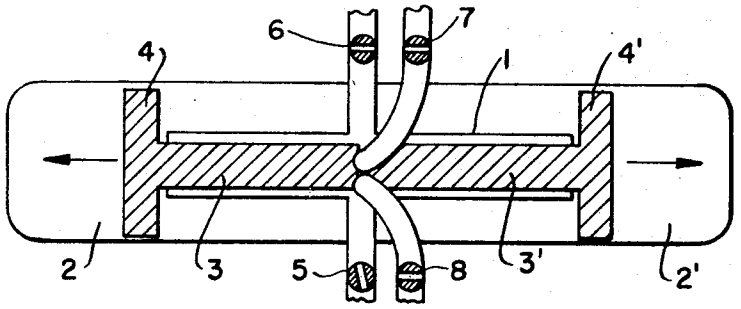


FIG. 4A.

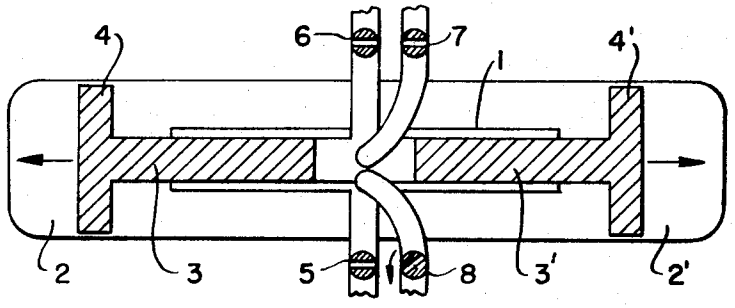


FIG. 4B.

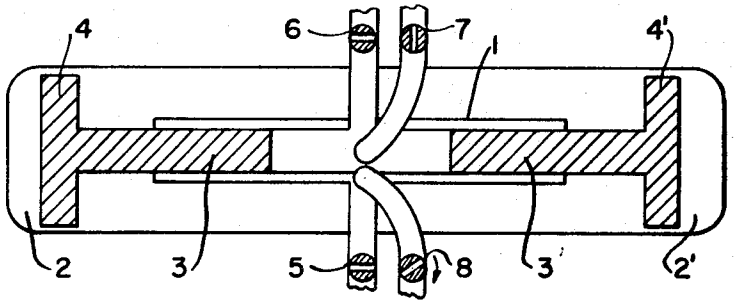


FIG. 4C.

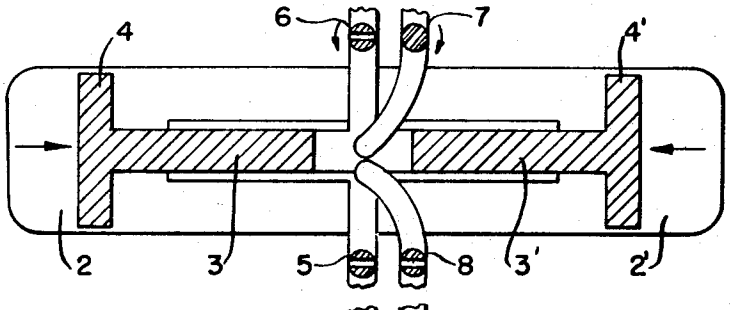


FIG. 4D.

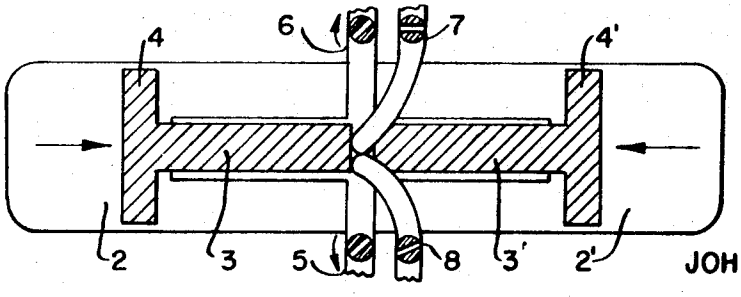


FIG. 4E.

INVENTOR

JOHN GORDON DAVOUD

BY

ATTORNEY

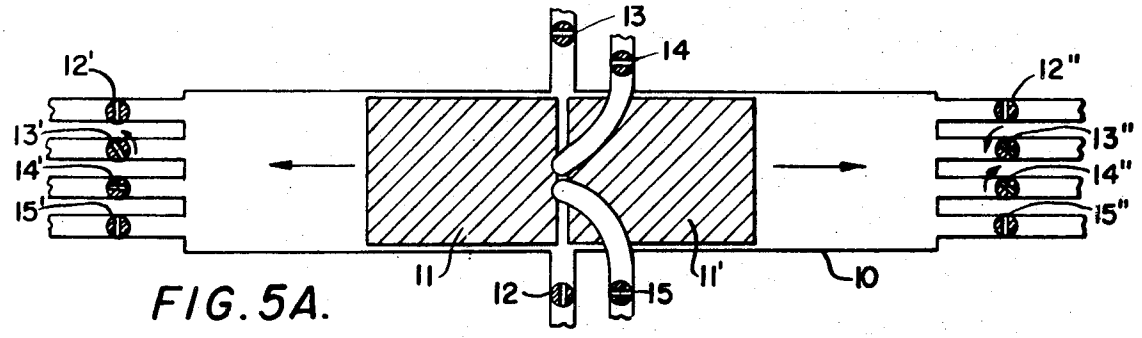


FIG. 5A.

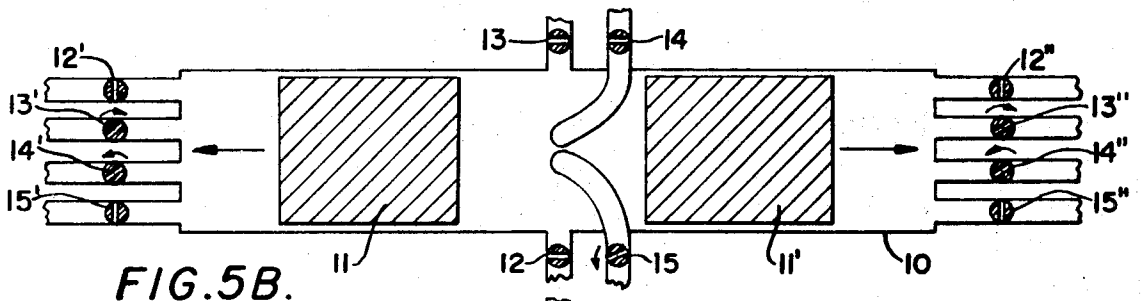


FIG. 5B.

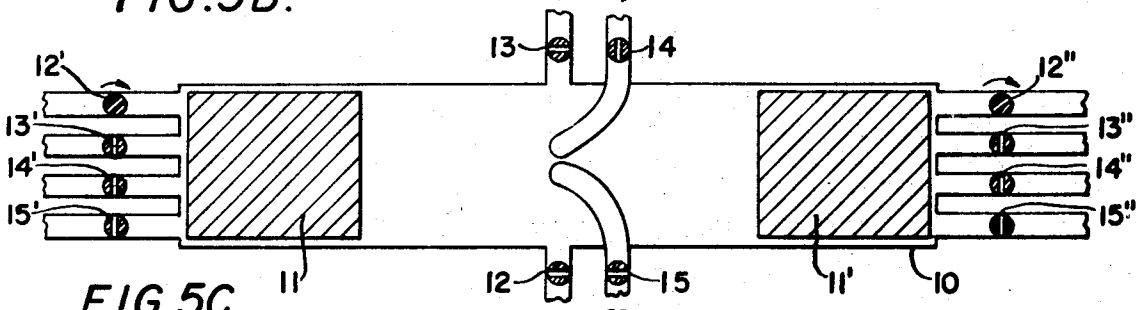


FIG. 5C.

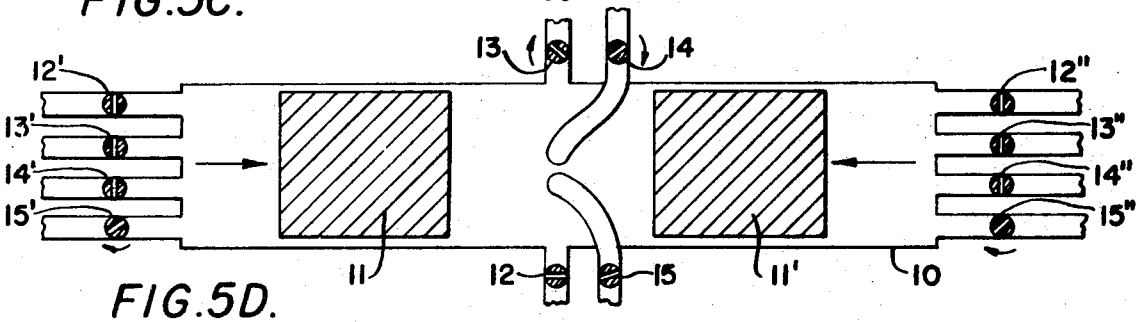


FIG. 5D.

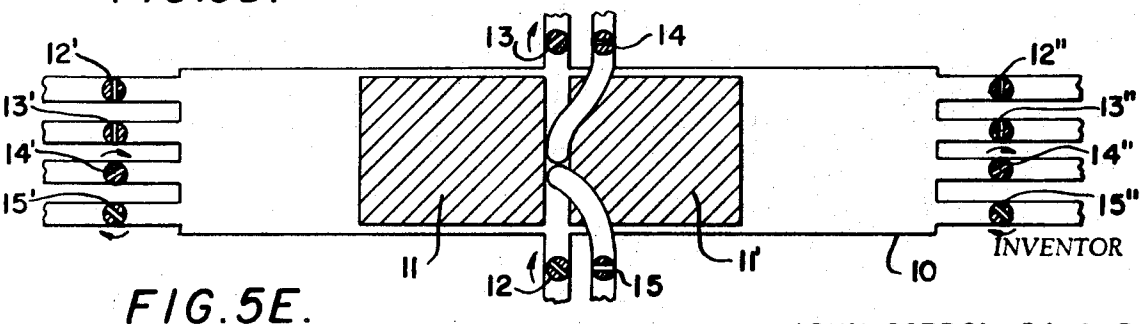


FIG. 5E.

INVENTOR

JOHN GORDON DAVOUD

BY Stauffer & Stauffer

ATTORNEY

FIG. 6.

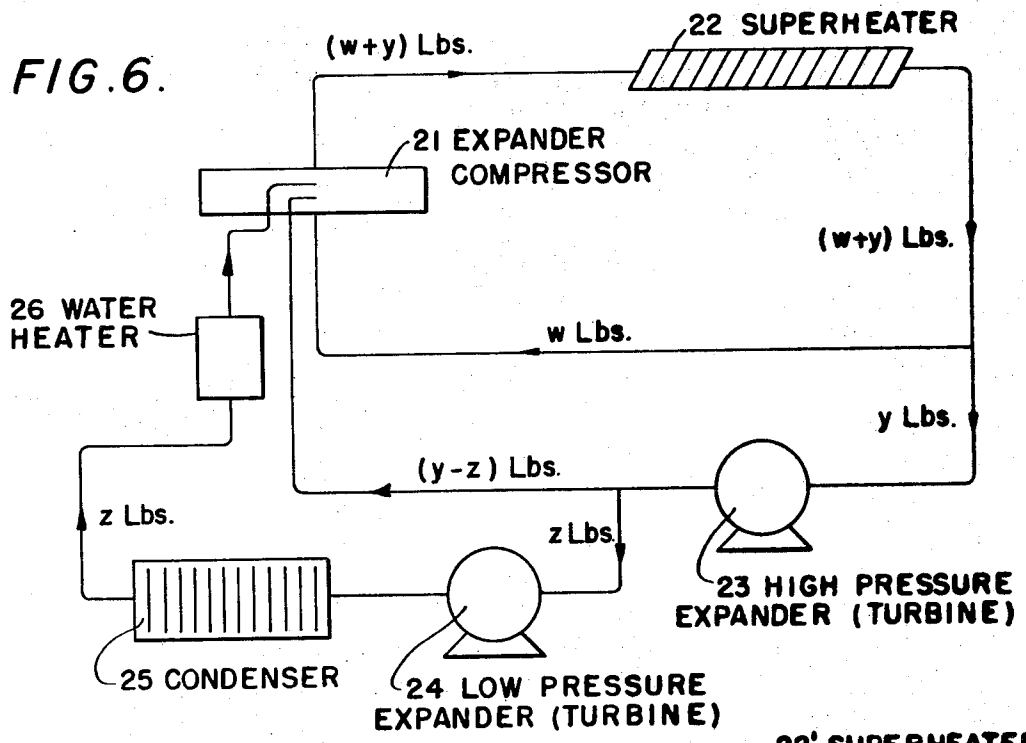
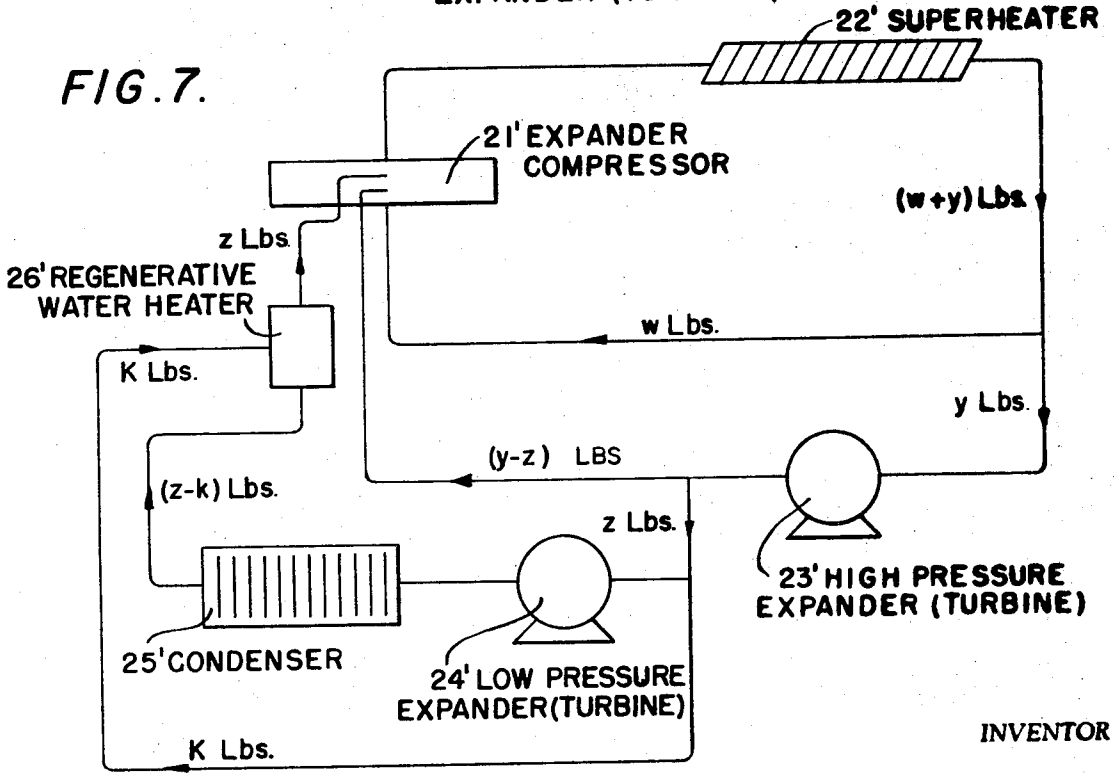


FIG. 7.



INVENTOR

JOHN GORDON DAVOUD

BY Stowell & Stowell

ATTORNEY

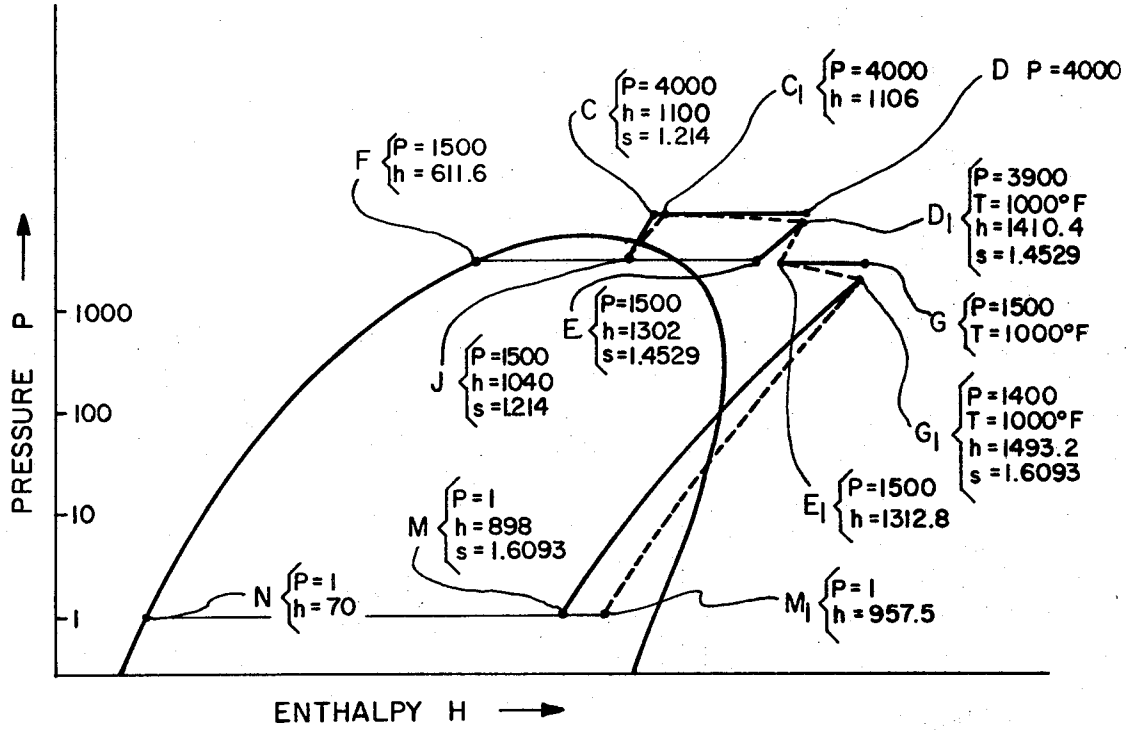


FIG. 8

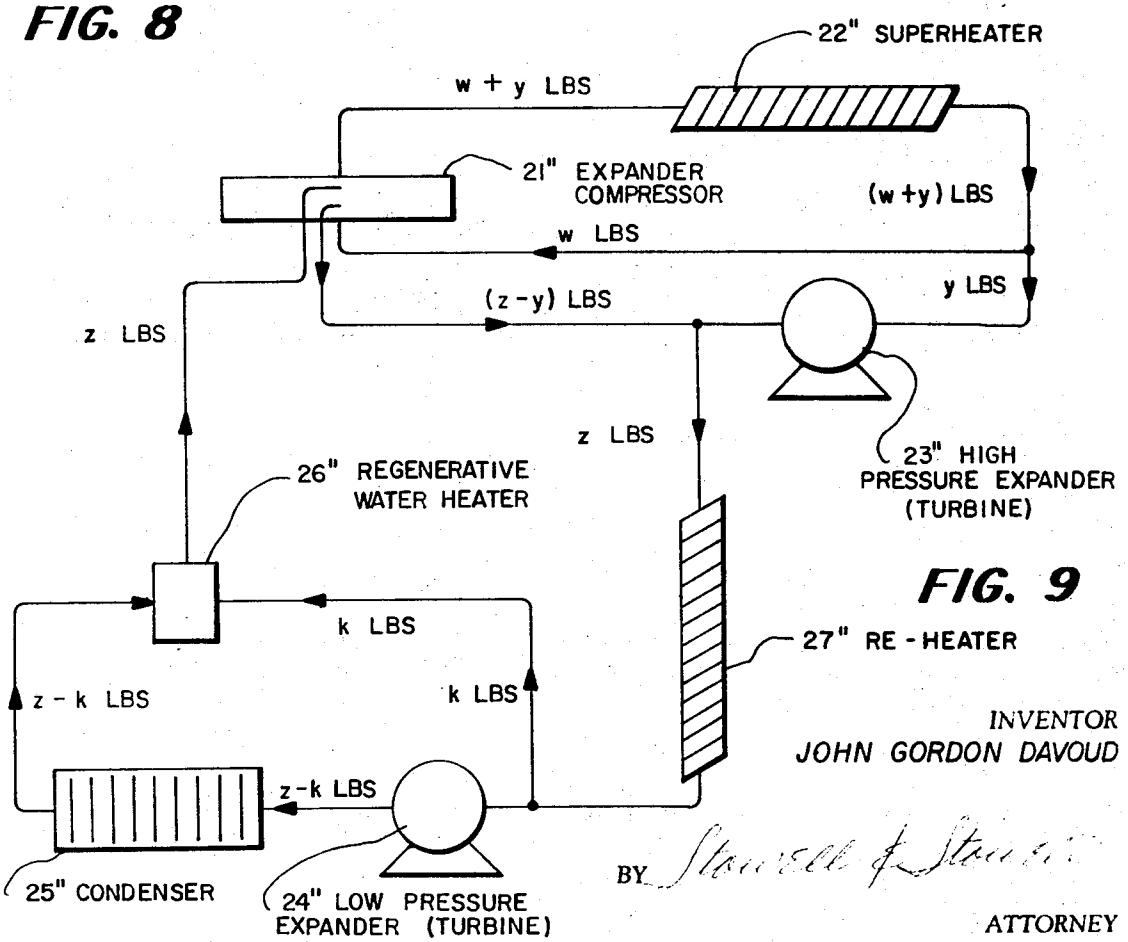


FIG. 9

INVENTOR
JOHN GORDON DAVOUD

BY Stowell & Stowell

ATTORNEY

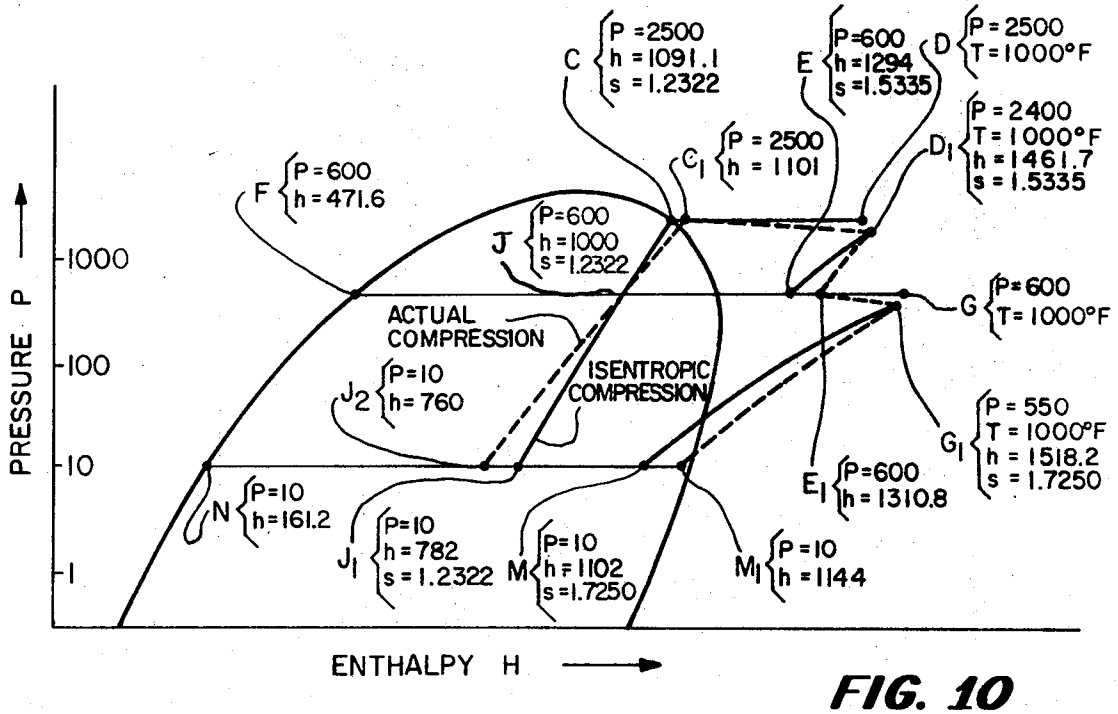
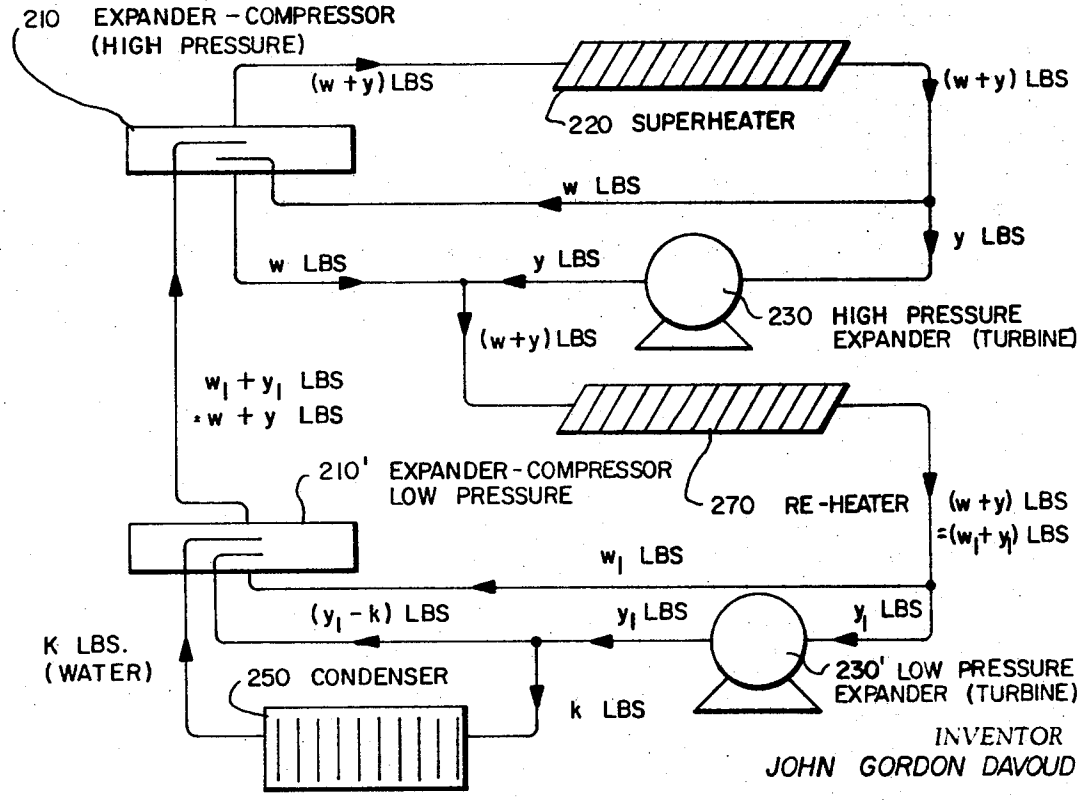


FIG. 10



INVENTOR
JOHN GORDON DAVOUD

FIG. 11

BY Stowell & Stowell

ATTORNEY

As there is a progressive increase in the slopes of the lines of constant entropy of steam with decreasing entropy, more work can be gained, for example, by expanding a given weight of superheated vapor from a higher pressure P to a lower pressure p, at high entropy, (DE, D_1E_1, D_2E_2), than is required to compress the same weight of (liquid plus vapor) back from p to P at lower entropy (JC). From quantitative diagram similar to FIG. 1, it will be seen that conditions can be found where the difference in slope between two pressures P and p approaches or exceeds 2, i.e., much more work is theoretically released by expanding altogether, largely or partially, in the superheat region between two given pressures, than is required to compress the same weight of (vapor plus liquid) in the region of mixtures, from lower to higher pressure.

On the diagram in FIG. 1, this can be seen as the difference between KG (expansion work along the isentropic line DE) and IL (work of compression along the isentropic JC).

This principle has been applied to the gainful production of useful work in the patent application entitled "External Combustion Power Producing System" — Davoud et al., filed July 24, 1970, Ser. No. 58,099.

The advantages disclosed in said application can be adapted to the generation of steam. To do this, it is necessary to arrange a system of expander and compressor where the work of expansion of steam at a given entropy is nearly, or ideally exactly balanced by the work of compression of a larger weight of steam having a lower entropy. As indicated in examples set forth hereinafter, it is then possible, from the expansion of a given weight of superheated steam, say w pound, to produce by compression $(w + y)$ pounds of dry saturated steam; or of steam in the critical region, or of wet steam, when "y" is a number greater than zero.

Thus, in FIG. 1,

D–E Expand w lb. superheated steam from pressure P and temperature T_3 to p and T_1. Work out = KG.

E – J Add z pounds of water and $(y-z)$ lbs of steam, having state E, at temperature T_1. The point J lies on the line of constant entropy which passes through C; C has pressure P, and can lie, as shown, on the saturated vapor line.

J – C Compress $(w + y)$ lbs. of wet steam along the line of constant entropy from pressure p to P.

Conditions can be chosen, i.e., w, y, and z, and the location of the point D with respect to C, and the pressures P and p, so that the work of expansion of w exactly equals the work required for compression of $(w + y)$.

Then

C – D Superheat $(w + y)$ lbs. of dry saturated steam from saturation temperature corresponding to pressure P, to temperature T_3.

Then — at D — steam is separated into two portions; — w lb. of superheated steam at D, is required for expansion to provide work to compress $(w + y)$ lb. of wet steam from J to C; and y pounds of superheated steam at D is available to do useful work, — for example, to expand through a turbine.

In the above description, the point C and the point E were described as both being on the saturated vapor line; i.e., the expansion DE and the compression JC both end on the saturated vapor line. Other conditions can be chosen. In FIG. 1, D_1 is chosen so that considerable superheat remains at the end of expansion; D_2E_2 shows an isentropic expansion starting in the superheat and ending in the region of mixtures.

Similarly, compression can be arranged so that the point C lies on the line of saturated vapor, within the region of mixtures, in the region of superheat, below critical pressure, or in the critical region.

FIG. 2 shows how both expansion and compression can take place within the region of mixtures. FIG. 3 shows an arrangement where expansion takes place at a pressure greater than or close to critical pressure, from a point in the region of superheat, to a point which can lie in the superheat, on the saturated vapor line, or within the region of mixtures, and compression takes place along a line of constant entropy which passes through, or close to, the critical point. Other locations within the steam dome can be chosen for the line JC in FIG. 3.

In each of the three cases represented in FIGS. 1 to 3, when conditions are chosen in such a way that the work of expansion is considerably greater than the work of compression, it is found that more steam is required at the end of expansion to bring the heated and pressurized condensate at F in FIGS. 1, 2 and 3, to the desired entropy J, at a point along the constant temperature and pressure line FE, than is normally required to provide the necessary work out in the expander-compressor.

A preferred source of the added steam required is to expand (in a separate expander) (see FIGS. 1, 6 and 7) that portion of steam available to do useful work, in at least two stages, the first of which is over the same pressure range as the operating range of the expander-compressor. At the end of this stage of expansion, the steam is separated into two portions; one portion is added to the steam which has expanded in the expander-compressor, both portions of steam being then at the same state. The other portion of steam which has already done useful work, is allowed to expand to some lower pressure, through a suitable expander, doing further useful work. The numerical examples which are included hereinafter, and the applications of the principles described heretofore, and shown in FIGS. 6 and 7, are based on the method described herewith in this paragraph. A special case, for which condition can be selected, is when the portion of steam added to the steam already expanded, in the expander-compressor, is zero; i.e., only water is added.

In actual practice, as will be shown below, it is more usual to have to withdraw steam from the expander-compressor at the end of expansion.

In the theoretical cases described below, where both steam and water are added to the expanded steam before compression, the sequence of events is,

At maximum pressure and temperature — State point D, FIG. 1, D FIG. 2, D_1 FIG. 3, there are $w + y$ lbs. of steam. w lb. is required to operate the expander-compressor; y lbs. are available to do useful work. y lbs. are expanded through an expander between P and p, — the same operating range as the expander-compressor. At point E in FIGS. 1 and 2, or E_1 in FIG. 3, it is separated into two portions, $(y - z)$ and z. (see FIGS. 6 and 7). z lbs., with or without reheat, is allowed to expand to some lower pressure; it is condensed, re-heated

CONDENSABLE VAPOR POWER PRODUCING SYSTEM

Considerable effort and money have been and are continuing to be spent to introduce the Rankine steam cycle to ground traveling vehicles as a replacement for engines operating on the conventional Otto and Diesel cycles, particularly in view of the reduced air pollution from the external combustion system used for heat input in the Rankine cycle.

Also, it is desirable to increase the thermal efficiency of the steam power system in electric power generation, and thereby reduce both air and thermal pollution, while at the same time saving fuel.

It is a particular object of the present invention to provide a technically feasible and commercially adaptable external combustion power producing cycle which may be used as the power producing cycle for vehicular and stationary engines, and electric power generation, which would have improved air pollution abatement and possess substantially greater efficiency than the known Rankine cycle and in addition, would have the following advantages:

A. A cycle that may be applied to all known combinations of reciprocating, turbine, and positive displacement rotary engines and compressors;

B. A cycle which has a significant improvement in thermal efficiency and therefore corresponding reduction in fuel usage compared with gasoline fuel internal combustion spark ignition engines and conventional Rankine cycle steam engines;

C. A cycle which will function with large reduction, in condenser load and proportionate reduction in condenser dimensions and weight, compared with Rankine system of similar power;

D. A cycle which would be readily adapted to automotive propulsion systems, having increased efficiency and reduced pollution; in addition, a system which would be amenable to mass production, and have acceleration and driving response comparable to present automotive pleasure cars, together with high reliability, low maintenance costs, reasonable life expectance and acceptable start-up time.

E. A cycle which is adaptable to electric power generation in thermal stations, and which would result in more efficient use of fuel, decreased thermal pollution and, where fossil fuels are the heat source, reduced air pollution.

The invention will be described in respect to steam as the condensable fluid; however, it will be apparent that other condensable vapors may be employed in the system and the invention will be described in reference to the accompanying drawing wherein:

FIG. 1 is a phase diagram for water, wet and superheated steam, in which the pressure of the steam is plotted as Y-axis against the heat content or enthalpy along the X-axis;

FIG. 2 is a further phase diagram illustrating that steam can be expanded from a predetermined pressure and then recompressed to such pressure solely within the region of mixtures;

FIG. 3 is a phase diagram illustrating further variation of this invention;

FIGS. 4A through 4E illustrate one form of apparatus for carrying out expansion and compression of steam in a common cylinder;

FIGS. 5A through 5E illustrate another form of free piston steam expansion and compression means suitable for carrying out a portion of the system of the invention;

FIG. 6 is a diagrammatic showing of one form of power producing apparatus within the scope of the present invention;

FIG. 7 is a diagrammatic showing similar to FIG. 6 of another form of power producing apparatus within the scope of the invention;

FIG. 8 is phase diagram showing state-points for an application of the invention where departure from isentropic expansion and compression occur.

FIG. 9 is a diagrammatic showing similar to FIGS. 6 and 7 of a power producing system adjusted for non-isentropic expansion and compression;

FIG. 10 is a phase diagram showing application of the invention to more than one stage of expansion and compression; and

FIG. 11 is a diagrammatic showing similar to FIGS. 6, 7 and 9 of a power producing system involving two expansion and two compression stages, with non-isentropic expansion and compression.

The thermodynamic properties of steam and other condensable vapors as applied to their use in heat engines can be most simply understood by reference to a phase diagram for water, wet and superheated steam, in which the pressure of the steam is plotted as Y-axis against the heat content or enthalpy along the X-axis of a diagram as illustrated in FIG. 1 of the drawings. While the diagram FIG. 1 is for water, it will be similar for other fluids containing polar groups, of relatively low molecular weight, such as ammonia, carbon dioxide, methanol, ethanol, and trifluorethanol.

In heat engines and compressors, both expansion and compression of the working fluid take place, theoretically, at constant entropy. Thus, the course of an expansion between two pressures P and p can be shown by a line of constant entropy, such as DE, D_1E_1, D_2E_2 or CJ, in FIG. 1. The work out, or work done by the expansion, is given by the length of the projection of the line of constant entropy, for example DE, between P and p, on the X-axis, — shown by KG in the diagram.

Similarly, compression from E to D would take place in the reverse direction along the same isentropic line, and the intercept KG would in this case be the work done on the system — or work in, or compressor work.

On a p-H diagram such as in FIG. 1, the entropy at any given pressure decreases with decreasing enthalpy. Thus, any point on the constant entropy line JC will have a lower entropy than any point on the constant entropy line DE, D_1E_1, or D_2E_2.

Between any given pressure, P and p, the slope of the isentropic line is given by

$$\frac{(P-p)}{(\text{Intercept on } x-\text{axis})},$$

i.e., in FIG. 1, the slope of isentropic line JC is $(P-p)/IL$; of the isentropic line DE is $(P-p)/KG$.

Thus, between any given pressures, the slope of the isentropic line is a reciprocal measure of work done by the system in expansion, or of work done on the system in compression; that is, the greater the slope, the less the work out in expansion, or work in during compression.

under pressure along the saturated liquid line, preferably by regenerative means, to point F (FIGS. 1, 2 and 3).

$(w+y-z)$ lbs. of steam at E (FIGS. 1 and 2) or at E_1 (FIG. 3) is added to the expanded steam which operates the expander-compressor, while the piston is still moving, in such a way that the pressure of the steam therein does not change. In this way, there will then be at the point of maximum volume in the expander-compressor $(w+(y-z))$ lbs. of steam, at pressure p. In the case shown by FIGS. 1 and 2, this steam will be at temperature T_1; z lbs. of water, also at temperature T_1, at point F, is then added to $(w+y-z)$ of steam. The resultant $(w+y)$ lbs. of wet steam at state J, is compressed by the return stroke of the piston or pistons, to C. $(w+y)$ lbs. of steam are then brought by a steam separator and boiler in FIG. 2, or a superheater, in FIG. 1, to point D, and the cycle repeats.

In the case shown by FIG. 3, when the first expansion stage ends in the superheat, water at F will be at a lower temperature than steam at E_1. In this case, water should be added at such a rate that the pressure remains constant until the steam is saturated, and no further reduction in temperature occurs. This can be done by adjusting the rate of admission of water to the expanded steam in the initial stages of the compression stage.

From the above, it is seen that a means must be provided for dividing the steam used for useful work into at least two portions after the first expansion stage, which corresponds to the operating pressure range of the expander-compressor; and of adding one of the portions to the expanded steam before admission of water.

From the above, it is clear that a portion of the steam is expanded simply to provide work to operate a compressor and in principle any combination of expander and compressor — reciprocating, turbine or positive displacement rotary — could be used. No net useful work comes out of this combination; and the cheaper, and simpler it is, and the higher the efficiency, the better.

An advantageous arrangement is to expand and compress the steam in the same cylinder, and to provide a minimum of mechanical complexity, this can to great advantage be done by using a free-piston expander-compressor.

Useful simplification results when the expansion of steam at a given entropy and the compression of a greater quantity of steam at a lesser entropy, takes place in the same cylinder. A diagrammatic sketch of one form of such an expander-compressor is described as Example 1, and shown in FIG. 4, A to E, in which

1 is a cylinder for expansion and compression of steam;
2, 2′ are bounce chambers, containing air;
3,3′ are pistons for expansion and compression which operate in cylinder 1;
4,4′ are bounce chamber pistons;
5 is an inlet valve and inlet for high pressure superheated steam;
6 is an outlet valve and outlet leading to a superheater;
7 is an inlet valve and inlet for admission of a metered weight of water into cylinder 1;
8 is an inlet for expanded steam at pressure p, (as shown in FIGS. 1, 2 and 3) into the cylinder.

In operation, the sequence of events is:

Start of expansion — FIG. 4A.

Steam pistons 3,3′ are almost touching, and moving outward.

Valve 5 (Inlet for superheated steam) open.

Valve 6 (Outlet to superheater) closed.

Valve 7 (Inlet of water) closed.

Valve 8 (Inlet for expanded steam at pressure p, — closed.

Half-way through expansion stroke — FIG. 4B.

Pistons moving outward. Air in bounce cylinders 2—2′ being compressed.

Valves 5, 6 and 7 — closed.

Valve 8 (Inlet for steam at pressure p — starting to open.

End of expansion — FIG. 4C.

Valve 5 (Inlet for superheated steam) closed.

Valve 6 (Outlet for superheater) closed.

Valve 7 (Inlet for water) open.

Valve 8 (Inlet for steam at pressure p) — closing.

Half-way through compression of steam — FIG. 4D.

Pistons moving inward due to expansion of compressed air in bounce cylinders.

Valve 5 (Inlet for superheated steam) closed.

Valve 6 (Outlet to superheater) closed; just starting to open.

Valve 7 (Inlet for water) closing.

Valve 8 (Inlet for steam at pressure p) — closed.

End of compression stroke — FIG. 4E.

Valve 5 (Inlet for superheated steam) — opening.

Valve 6 (Outlet to superheater) closing.

Valve 7 (Inlet for water) closed.

Valve 8 (Inlet for steam at pressure p) closed.

Cycle repeats.

In another form of free piston steam compressor, described in Example 2, and shown in FIG. 5, A to E, instead of using bounce cylinders to bring about the inward motion of the two pistons, steam is used to move the pistons in both directions; such an expander-compressor is double-acting.

In FIG. 5 A to E.

10 is a cylinder;

11, 11′ are pistons moving in cylinder 10;

12, 12′, 12″ are inlet valves, for high pressure steam;

13, 13′, 12″ are outlet valves and outlets leading to a superheater;

14, 14′, 14″ are inlet valves for admission of a metered weight of water into cylinder 10;

15, 15′, 15″ are inlet valves and inlets for steam at pressure p, FIGS. 1, 2 and 3.

In operation, the sequence of events may be:

Start of Outward Expansion — FIG. 5A.

Pistons are almost touching. The space at each end of the cylinder contains $(w+y-z)$ lbs. of expanded steam.

Valve 12 (Inlet valve for high pressure steam to central part of cylinders) open, to admit 2 w lbs. of high pressure steam.

Valves 12′, 12″ (Inlet valves for high pressure steam at ends of cylinder) closed.

Valves 13, 13′, 13″ (Outlet valves to superheater) closed.

Valve 14 (Inlet valve for water to center of cylinder) closed.

Valves 14′, 14″ (Inlet valves for water at ends of cylinder) open. z lbs. of water is injected at each end of cylinder 1 while the pistons are moving outward.

Valve 15, 15', 15'' (Inlet valves for steam at pressure p)— closed.

Half-way through outward stroke — FIG. 5B.

Valves 12, 12', 12'' (Inlets for high pressure steam) closed.

Valves 13 (Outlet from central part of cylinder to superheater) — closed.

Valves 13', 13'' (Outlet valves from ends of cylinder to superheater) starting to open.

Valve 14 (Inlet valve for water to central part of cylinder) — closed.

Valve 14', 14'' (Inlet valves for water to ends of cylinder) — starting to close.

Valve 15 (Inlet for steam at pressure p to central part of cylinder) — starting to open.

Valve 15', 15'' (Inlet valves for steam at pressure p to ends of cylinder)— closed.

End of outward stroke — FIG. 5C.

Central part of cylinder now contains $2w + 2(y - z)$ lbs. of expanded steam.

Valve 12 (Inlet valve for high pressure steam to central part of cylinder)— closed.

Valve 13 (outlet valve from central part of cylinder to superheater) — closed.

Valve 14 (Inlet valve to central part of cylinder for water) open 2z lbs. of water is injected during inward stroke of pistons.

Valve 15 (Inlet valve for steam at pressure p to central part of cylinder) — closed.

Valves 12', 12'' (Inlet valves for high pressure steam to ends of cylinder) opening — nearly wide open.

Valves 13', 13'' (Outlet valves from ends of cylinder to superheater) closed.

Valves 14', 14'' (Inlet valves for water to ends of cylinder) closed.

Valves 15', 15'' (Inlets for steam at pressure p to ends of cylinder) — closed.

Half-way through inward stroke — FIG. 5D.

Valve 12 (Inlet valve for high pressure steam to central part of cylinder) closed.

Valve 13 (Outlet valve from central) part of cylinder to superheater) just starting to open.

Valve 14 (Inlet valve for water to central part of cylinder) — starting to close.

Valve 15 (Inlet for steam at pressure p to central part of cylinder) — closed.

Valves 12', 12'' (Inlet valves for high pressure steam at ends of cylinder) — closed.

Valves 13', 13'' (Outlet valves from ends of cylinder to superheater) — closed.

Valves 14', 14'' (Inlet valves for water at ends of cylinder) — closed.

Valves 15', 15'' (Inlets for steam at pressure p to ends of cylinder) — opening.

End of Inward Stroke — FIG. 5E.

Valve 12 (Inlet valve for high pressure steam to central part of cylinder) — opening — nearly open.

Valve 13 (Outlet valve from central part of cylinder to superheater) closing — nearly closed.

Valve 14 (Inlet valve to central part of cylinder for water) — closed.

Valve 15 (Inlet for steam at pressure p to central part of cylinder) — closed.

Valves 12', 12'' (Inlet valves for high pressure steam to ends of cylinder) — closed.

Valves 13', 13'' (Outlet valves from ends of cylinder to superheater) — closed.

Valves 14', 14'' (Inlet valves to ends of cylinder for water) — opening — nearly open.

Valves 15', 15'' (Inlets for steam at pressure p to ends of cylinder) — closing; nearly closed.

Cycle repeats.

Various methods known in the art can be used for maintaining the pistons in the correct position relative to each other; for example, the non-mechanical method taught in U.S. Pat. No. 3,127,881; and the simple mechanical method taught in U.S. Pat. Nos. 3,369,530 and 3,369,738.

Ways in which the steam-operated free piston steam compressor would fit into a power-producing cycle is described below as Examples 3, 4 and 5, and shown in block diagrams schematically in FIGS. 6 CONDITIONS 7.

EXAMPLE 3 — Steam Conditions For Steam-Operated Free Piston Compressor — FIG. 1, Isentropic Expansion D–E

Temperature T_2 of steam leaving superheater 22 — 1,000° F

Pressure P of steam leaving superheater 22 —1,000 psia

Expand w lbs. of steam in the free piston expander-compressor 21 (FIG. 6) to the saturated vapor line.

The above conditions correspond to isentropic expansion DE illustrated in FIG. 1 of the drawing. The pressure at the end of this expansion stage (p in the drawing of FIG. 1) is about 54 psia.

Expand y lbs. of steam at 1,000° and 1,000 psia, (as it leaves superheater 22(FIG. 6) through a turbine or other form of expander, to p (54 psia).

Divide y lbs. of steam at pressure $p = 54$ lbs. into two portions — z lbs. and $(y - z)$ lbs. — without altering the state of either portion.

At the end of the expansion of w lbs. of steam in the expander-compressor, continue, by movement of the pistons, the expansion of the chamber containing this expanded steam to draw $(y - z)$ lbs. of steam at pressure $p = 54$ psia into the space, — also without altering the state of the w lbs. of steam already in the space, or the $(y - z)$ lbs. of steam drawn in by the continuing movement of the piston or pistons.

Expand z lbs. of steam at pressure p and temperature T_1 as illustrated in FIG. 1, through a turbine or other form of expander, to a lower pressure and temperature — for example 1 psia, doing further useful work. Condense all this steam (z lbs.); heat by suitable means — external heating, or regeneratively, the condensate under pressure, to the temperature T_1 illustrated in the FIG. 1 of the drawings.

Add z lbs. of condensate at temperature T_1 (corresponding to pressure $p = 54$ psia) to $w + (y - z)$ lbs of steam at pressure $p = 54$ psia in the expander-compressor, while compressing the total $w + (y - z) + z = (w + y)$ lbs. back to pressure P = 1,000 psia by the return stroke of the piston or pistons in the expander-compressor.

This corresponds to isentropic compression line JC in FIG. 1 of the drawings, and under the conditions described, results in dry saturated steam at 1,000 psia.

For convenience, in calculation, let $w = 1$ lb. Work out from expansion of 1 lb. of steam = 329.5 BTU Work of compression of 1 lb. of steam along line JC in FIG. 1, where P = 1,000 psia and p =54 psia =212 BTU. Then 1 lb. of steam at 1,000° + 1,000 psia expanding to 54 psia can compress 329.1/212 = 1.555 lbs. of steam of composition J as shown in FIG. 1 to the point C, (which in this example is 1,000 psia on the saturated vapor line).

The enthalpy of the points F, J and E are:

F — 255.1 BTU/lb.

J — 980 BTU/lb.

E — 1,176 BTU/lb.

This enables the quantity z to be defined.

As shown above, $1 + y = 1.555$ lbs.

$$\therefore y = 0.555$$

$$\text{and } z\,(255.1) + (1.555 - z)\,(1{,}176) = 1.555\,(980)$$

$$\text{From which } z = 0.331$$

That is, 0.555 lbs. of steam is available to do useful work through the expansion stage 1,000 to 54 psia; and 0.331 lbs. is available to do further useful work by expansion from 54 psia to 1 psia.

Total useful work out = 0.555 (329.5) + 0.331 (1,176 − 924)

= 183 + 83.5 = 266.5 BTU.

Heat required (superheater): 1.555 (1,505 − 1,192) =486 BTU

Heat required (water heating): 0.331 (255 − 69.7) = 61.3 BTU

Total heat required =547.3 BTU

Efficiency e = 266.5/547.3 = 48.5 percent

A further increase in efficiency is brought about by using some of the steam (z lbs. at point E in FIG. 1) for water heating. This is best accomplished in stages in the manner known as regenerative heating.

It is also advantageous, in the present invention to reheat the steam (z lbs.) to maximum temperature before further expansion. As many stages of reheat can be used as may be advantageous, in the manner known to the art.

By the use of a small compression ratio, and a maximum pressure in or near the critical region, a given weight of superheated steam expanding in the free piston expander-compressor **21** will compress several times its weight of wet steam. A numerical example of a power cycle using such conditions is described below as Example 4. The conditions for the expander-compressor **21** are shown in a non-quantitative way by the isentropic expansion D_1E_1, and compression JC in FIG. 3.

EXAMPLE 4. — STEAM CONDITIONS FOR STEAM OPERATED COMPRESSOR — FIG. 3

For Point D_1, $p = 4{,}000$ psia, Temperature = 1,000°, h (enthalpy) = 1,406.8 BTU/lb.

For Point E_1, P = 1,500 psia, h = 1,293 BTU/lb.

For Point J, p = 1,500 psia, h = 879 BTU/lb.

For Point C, P = 4,000 psia, h = 910 BTU/lb.

For Point F, p = 1,500 psia, h = 611.6 BTU/lb.

Then w lbs. of superheated steam at Point D_1 expanding to E_1 will compress $(w + y)$ lbs. of wet steam from J to C. For convenience, let $w = 1$.

Work out from expansion of 1 lb. from D_1 to E_1 = 114 BTU/lb.

Work of compression of 1 lb. from J to C = 31 BTU/lb.

Then to determine y (amount of steam available for useful work)

$$y = \left(\frac{114}{31} - 1\right) = (3.67 - 1) = 2.67 \text{ lbs.}$$

Thus, for every pound of steam required to operate the compressor, 2.67 lbs. is available to do useful work, through the expansion stage 4,000 to 1,500 psia. This is the quantity y, (as in Example 3 above).

In the same way as shown in Example 3, from the enthalpy figures for points F, J, and E_1, the value of z can be calculated.

$$z(611.6) + (3.67 - z)\,1{,}293 = 3.67\,(879)$$

$$682.4z = 1{,}520$$

$$z = 2.23 \text{ lb.}$$

Useful work out is obtained by expanding 2.67 lbs. from 4,000 to 1,500 psia (y), and further expanding 2.23 lbs. (z) to some lower pressure, for example, 1 psia.

Then: Total useful work out = 2.67 (114) + 2.23 (1,293 − 808) = 1,391.

Heat required for superheater = 3,67 (1406.8 − 910) = 1825 BTU

Heat required for water heating = 2.23 (611.6 − 69.7) = 1,208 BTU

Total heat in = 3.033 BTU

$$\text{Efficiency } e = 1{,}391/3{,}033 = 46 \text{ percent}$$

Examples 3 and 4 require a feedwater heater, shown as **20** in FIG. 6, with external heating.

A favored way to carry out such heating is to withdraw a portion of steam from the turbine **23'** (FIG. 7) or other form of expander used for production of useful mechanical work, after expansion, for example, to the pressure p as shown in FIG. 3 — and use the steam so withdrawn for heating the condensate. The effect of doing this, under the conditions described above for Example 4, is given below, and described as Example 5.

EXAMPLE 5 — CONDITIONS FOR OPERATION OF FREE PISTON STEAM OPERATED STEAM EXPANDER-COMPRESSOR — AS IN EXAMPLE 4

1 lb. of superheated steam will compress 3.67 lbs. of wet steam along isentropic JC (FIG. 1) from 1,500 to 4,000 psia.

Expand 2.67 lbs. of steam from 4,000 psia to 1,500 psia.

Allow k lb. of steam to expand from 1,500 psia to 1 psia.

Use $(2.67 - k)$ lbs. of steam at 1,500 psia to heat the condensate.

$$(3.67 - k)\,1{,}293 + k\,(70) = 3.67\,(879)$$

$$\text{and } k = 1.243 \text{ lbs.}$$

Then: Work out from expansion through turbine or turbines =

2.67 × 114 + 1.243 (1,293 − 804) = 304 + 608 = 912 BTU

Heat required (for superheater **22**) = 3.67 (1,407 − 910) = 1,825 BTU.

Efficiency e 912/1,825 = 50 percent. The application of Example 5 to a power producing system is shown schematically in FIG. 7.

Even higher efficiencies can result if this principle is extended to removal of steam in a number of stages, in the way known as regenerative heating.

As shown in FIG. 2, the free piston compressor can be operated by steam under conditions which fall within the region of mixtures. A cycle based on such a mode of operation is described below as Example 6. The method of calculation is the same as for Example 5; mode of partition, and results are given.

EXAMPLE 6 — CONDITIONS FOR OPERATING COMPRESSOR — SEE FIG. 2

Expansion — Line DE —
Point D (on saturated vapor line)
P = 1,000 psia
Temperature = 544.6° F.
Enthalpy h = 1,192 BTU/lb.
Point E p = 50 psia
h = 975 BTU/lb.
Point J p = 50 psia
h = 690 BTU/lb.
Point C P = 1,000 psia
h = 802 BTU/lb.

At point D, 1 lb. of steam is required to operated compressor; 0.936 lb. are available for expansion to some lower pressure, e.g., 1 psia, to do useful work.

Using the nomenclatures developed in Examples 3 through 5:

$$y = 0.936$$

z = 0.761 (when all steam condensed at lowest pressure is heated externally to saturation temperature corresponding to p — FIG. 2 — in this example, p — 50 and T_1 — 281°.

Work out = (0.936 × 217) + 0.761 (975 − 776) = 357.5 BTU.

Heat in by boiler (point C to point D — , FIG. 2).

$$1.936\,(1192 - 802) = 755 \text{ BTU}$$

Heat in (water heating) = 0.761 (250 − 70) = 137 BTU

Efficiency e = 357.5/755 + 137) = 40 percent

As in Example 5, the condensate can be advantageously heated with steam. Using the same nomenclatures as in Example 5 — if k lbs. of steam are allowed to expand from 50 to 1 psia, and $(y - k)$ are used for heating the condensate, —

$$(1.936 - k)\,975 + (70) = 1.936\,(690)$$

$$K = 0.609 \text{ lbs.}$$

Work out = 0.936 × 217 + 0.609 × 199 = 324.4
Heat in = 755 BTU/lb.
Efficiency e = 324.4/755 = 43 percent.

Examples 3 through 6 above show isentropic expansion and compression; and in all cases that it is necessary to add steam at the end of expansion in the free piston expander-compressor to bring the heated and pressurized condensate at F in FIGS. 1, 2 and 3, to the desired state-point J.

In actual practice, when account is taken of such factors as departure from isentropy during expansion and compression, and pressure drop through superheaters, the ratios of the quantities w, y and z (using the previously developed nomenclature) can alter in such a way that it is necessary to withdraw steam from the expander-compressor, before addition of water and recompression. This is liable to occur in the majority of actual cases.

A numerical example of a cycle with re-heat, useful for power generation, is described below as Example 7, and a pH diagram is shown as FIG. 8, with numerical data for the operative state-points shown on the diagram.

EXAMPLE 7 — STEAM CONDITIONS FOR STEAM-OPERATED FREE PISTON COMPRESSOR — FIG. 8

Solid Line D_1E — Isentropic expansion
Dotted Line D_1E_1 — Actual expansion
Solid Line JC — Isentropic compression
Dotted Line Jc_1 — Actual compression

Lines D_1E_1 and Jc_1 represent 10 percent departure from true isentropic expansion (shown by D_1E) and compression (shown by JC).

Line C_1D_1 reflects a 100 psia fall in pressure through the high pressure superheater; and E_1G_1 reflects a 100 psia fall in pressure through the re-heat stage.

Line G_1M_1 represents a 10 percent departure from true isentropic expansion, i.e., it is only 90 percent efficient.

Referring to FIG. 8:
High pressure expansion — Line D_1E_1
point D_1 — P = 3,900 psia, Temperature — 1,000°F, h (enthalpy) = 1,410.4 BTU/lb.
For point E_1, P = 1,500 psia, h = 1,312.8 BTU/lb.
For point J, h = 1040 BTU/lb.
For point C_1, h = 1,106 BTU/lb.
For point F, h = 611.6 BTU/lb.
Work out from expansion of 1 lb. of steam from D_1 to E_1 = 97.6 BTU/lb.
Work of compression of 1 lb. from J to C_1 = 66 BTU/lb.

To determine y (amount of steam available for useful work), —

$$y = 97.6/66 - 1 = 0.482 \text{ lbs.}$$

From the enthalpy figures for points F, J and E_1 the value of z, the amount of steam available at point E_1, for further useful work, is found as follows, —

$$z \times 611.6 + (1.482 - z) \times 1{,}312.8 = 1.482 \times 1{,}040$$

z = 0.579 lbs.; i.e., 0.579 − 0.482 = 0.097 lbs. of steam can be withdrawn from the expander-compressor at the end of expansion, and is available for further expansion to some lower pressure.

Then: Useful work out is obtained by expanding 0.482 lb. from 3,900 to 1,500 psia; and 0.482 + 0.097 = 0.579 lbs. for expansion or regenerative heating, after further addition of heat.

From the enthalpy figures for points G_1, N and F, the value of k is found as follows:

$$k\,(70) + (0.579 - k)\,(1{,}493.2) = 0.579 \times 611.6$$

k = 0.358 lbs. — available for expansion from 1,400 psia to 1 psia.

Then: Total useful work out is:

$$0.482 \times 97.6 + 0.358 \times 536.7 = 239.4 \text{ BTU/lb.}$$

Total heat required:
For superheater = 1.482 (1,410 − 1,106) = 450 BTU/lb.
For reheat: 0.579 (1493.2 − 1312.8) = 104.5 BTU/lb.
Efficiency $e = 239.4/(450 + 104.5) = 43.3$ percent

Operated in this way, the expander-compressor shown in FIGS. 4A to 4E would require modification to valve **8**, and that shown in FIGS. 5A to 5E would require modification to valves **15**, **15'** and **15''**. These valves would serve to remove a measured amount of steam at state E_1, without altering the state of the portions removed, which can produce further work, or of the portion remaining. Valvular means by which this can be accomplished for single and double acting systems are described in detail in the aforementioned patent application, Ser. No. 58,099, Davoud, et al.

A way in which this principle can be applied to the generation of power is shown in FIG. **9**.

In power generation, high efficiency can result from using more than one stage of both expansion and compression, i.e., using two or more expander-compressor stages. A numerical example of a power system of this kind is described below as Example 8, and a *p*H diagram showing the quantitative values of the significant state points is shown as FIG. **10**. Ten percent departure from isentropy in both expansion and compression, and 100 psia pressure drop through the high pressure superheater, and 50 psia through the reheater, are assumed.

EXAMPLE 8 — STEAM CONDITIONS FOR STEAM-OPERATED FREE PISTON COMPRESSOR — FIG. **10**

High Pressure Stage —
Solid Line D_1E — Isentropic expansion
Dotted Line D_1E_1 — actual expansion
Solid Line JC — Isentropic compression
Dotted Line JC_1 — Actual compression
Dotted Line C_1D_1— reflects 100 psia pressure drop through superheater
Dotted Line E, G, — reflects 50 psia pressure drop through reheater.
Using nomenclatures and methods as in previous examples:

$y = 0.495$ lbs. (available for useful work)

At point E_1 all steam used for operating compressor (w lbs.) + y lbs. used for expansion through high pressure stage for useful work, e.g., through a turbine, is reheated. A portion is required to operate the low-pressure expander-compressor; a portion is available for useful work. (When $w = 1$, $y = 0.495$; $w + y = 1.495$ lbs.)
Low pressure stage, —
Solid Line G_1M — Isentropic expansion
Dotted Line G_1M_1 — actual expansion
Solid Line J_1J — isentropic compression
Solid Line J_2J — Actual compression

As all steam (1.495 lbs. used in the system) has to be compressed, the amount of steam at G_1 which is required to operate the lower pressure expander-compressor is: 1.56/1.845 = 0.96 and 1.495 − 0.96 = 0.53 lbs. are available for useful work over the low pressure stage.

At point M_1 39.0 percent of total steam, is condensed. This is most readily done by condensing most of the steam used in the lower pressure turbine, and returning the remainder to the low pressure expander-compressor in the way described in Example 3. In FIG. **11**, the amount of steam condensed is shown as k.
Then: Useful work (high pressure stage) = 0.495 (150.9 = 74.7 BTU
Useful work (low pressure stage) = 0.53 (374) = 198 BTU
Total useful work = 272.7 BTU
Heat required: high pressure superheater 1.495 (1,461.7 − 1,101) = 539 BTU
Heat required: reheat 1.495 (1,518.2 − 1.310.8)= 309 BTU
Total heat required = 848 BTU

Efficiency $e = 272.7/848 = 32.2$ percent

A way in which this method of using free piston expander-compressors can be applied to power generation is shown as FIG. **11**.

It is recognized that various modifications may be made in this system, and the illustrated examples are by way of examples and not by way of limitation.

I claim:

1. In a power producing system of the external combustion type the method including heating a condensable vapor to a predetermined temperature at a predetermined pressure;
separating the heated condensable vapor into first and second portions;
expanding the first portion of the heated condensable vapor in a work producing zone to a lower pressure;
expanding the second portion of the heated vapor in a second zone;
adding the liquid of the condensable vapor to the second portion of the condensable vapor to form a weight of the vapor greater than the weight of the second portion but at a lower entropy;
utilizing the work produced in expanding the vapor in the second zone to compress the vapor expanding in the second zone, after expansion, plus the added liquid, to the original predetermined pressure.

2. The method defined in claim **1** including adding, prior to compression, a further portion of vapor to the said liquid at the same state as the second portion of vapor after its expansion.

3. The method defined in claim **2** wherein the amount of said further portion of vapor added to said liquid is determined by the entropy of the steam at the predetermined state-point at the start of compression and the maintenance of a constant amount of working fluid in the system for a predetermined power output.

4. The method defined in claim **1** including removing, prior to compression, a portion of said second portion of vapor without changing the state of said second portion of vapor.

5. The invention defined in claim **4** wherein the amount of said second portion of vapor removed prior to compression is determined by the entropy of the steam at the predetermined state-point at the start of

compression and the maintenance of a constant amount of working fluid in the system for a predetermined power output.

6. The invention defined in claim 1 wherein the liquid is added during compression of the second portion of the vapor.

7. The invention defined in claim 1 wherein the liquid is added before compression of the second portion of the vapor.

8. The invention defined in claim 1 wherein the liquid is added before and during compression of the second portion of the vapor.

9. In a power producing system of the external combustion type the method including heating steam to a predetermined temperature at a predetermined pressure;

separating the heated condensable vapor into first and second portions;

expanding the first portion of the heated steam in a work producing zone to a lower pressure;

expanding the second portion of the heated steam in a second zone;

adding water to the second portion of the steam plus a further portion of steam at the same state, as the second portion, after expansion, to form a weight of steam greater than the weight of the second portion but at a lower entropy;

utilizing the work produced in expanding the steam in the second zone to compress the steam expanded in the second zone, plus a further portion of steam having the same state as the second portion, after expansion, plus the added water to the original predetermined pressure.

10. The invention defined in claim 1 wherein the work produced in the second zone is substantially equal to the work expended in compression.

11. The method described in claim 1, where the expansion in the second zone takes place in the region of superheat, and the compression takes place in the region of mixtures.

12. The method described in claim 1, where the expansion in the second zone starts in the region of superheat, and ends in the region of mixtures, and compression takes place in the region of mixtures.

13. The method described in claim 1, where expansion in the second zone is from a point above critical pressure to a point in the superheat, and compression is along the line of constant entropy passing through or close to the critical point.

14. The method described in claim 1, where expansion in the second zone is from a point above critical pressure, to a point in the superheat, and compression may be along any line of constant entropy between the saturated liquid and saturated vapor lines.

15. The method described in claim 1, where expansion in the second zone and compression are both within the region of mixtures.

16. In a power producing system of the external combustion type means for heating a condensable vapor to a predetermined temperature at a predetermined pressure;

means for separating the heated condensable vapor into first and second portions;

means for expanding to a lower pressure the first portion of the heated condensable vapor in a work producing cycle;

second means for expanding the second portion of the heated vapor;

means for adding the liquid of the condensable vapor to the second portion of the condensable vapor to form a weight of vapor greater than the weight of the second portion but at a lower entropy;

further means for utilizing the work produced in the second expanding means to compress the vapor expanded therein to the original predetermined pressure.

17. The invention defined in claim 16 including means for adding a further portion of vapor to said liquid at the same state as the second portion of vapor after its expansion prior to compression of said liquid to the original predetermined pressure.

18. The invention defined in claim 16 including means for removing a portion of the second portion of the vapor without changing the state of said second portion of vapor.

19. In a power producing system of the external combustion type means for heating steam to a predetermined temperature at a predetermined pressure;

means for separating the steam in to first and second portions;

means for expanding to a lower pressure the first portion of the heated steam in a work producing cycle;

second means for expanding the second portion of the heated steam;

means for adding water plus an additional portion of steam having the same state as the second portion, after expansion, to the second portion of the steam to form a weight of steam greater than the weight of the second portion but at a lower entropy;

further means for utilizing the work produced in the second expanding means to compress the steam expanded therein plus added steam plus added water, to the original predetermined pressure.

20. In a power producing system of the external combustion type means for heating steam to a predetermined temperature at a predetermined pressure;

means for separating the steam into first and second portions;

means for expanding to a lower pressure the first portion of the heated steam in a work producing cycle;

second means for expanding the second portion of the heated steam;

means for removing a portion of the expanded steam from the second means for expanding steam, without altering the state of the portion removed or of the portion remaining, and means for adding water to the portion of steam remaining to form a weight of steam greater than the weight of the second portion but at a lower entropy;

further means for utilizing the work produced in the second expanding means to compress the steam expanded therein less steam removed plus added water, to the original predetermined pressure.

21. Apparatus defined in claim 16 wherein the second means and the further means comprise a free piston expander-compressor.

22. Apparatus as defined in claim 21 wherein the free piston expander-compressor includes air filled bounce chambers to return the pistons after expansion; and in which expansion takes place by introducing

steam between the pistons, moving them outwards, and compression takes place as the pistons return under the action of the air compressed in the bounce chambers.

23. Apparatus as defined in claim **21** wherein the free piston expander-compressor is double-acting, such that both inward (moving together) and outward (moving apart) motion of the pistons is activated by expanding steam, which expanded steam is then compressed in the expander-compressor, plus the addition of water; and in which expansion and compression take place in the space between the pistons, and in the two spaces between the pistons and the ends of the cylinder.

24. The use of a steam-operated expander-compressor of the free-piston type, to generate steam.

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