Dec. 20, 1960 G. SONNEFELD 2,964,910
METHOD AND SYSTEM FOR THE CARNOTIZATION OF STEAM CYCLIC PROCESSES
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Fig. 1

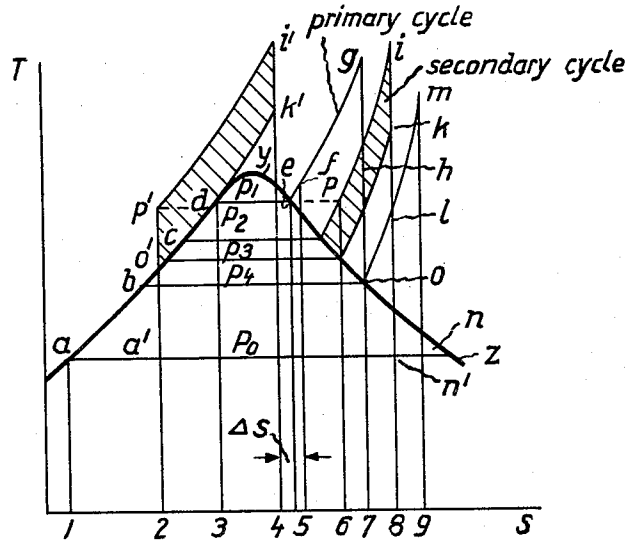
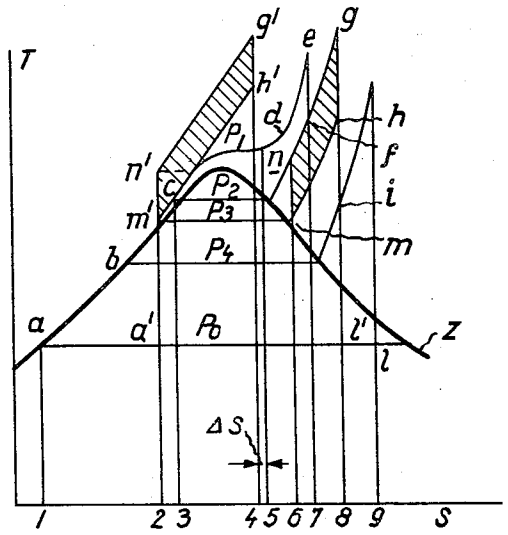


Fig. 2



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METHOD AND SYSTEM FOR THE CARNOTIZATION OF STEAM CYCLIC PROCESSES

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Fig. 3

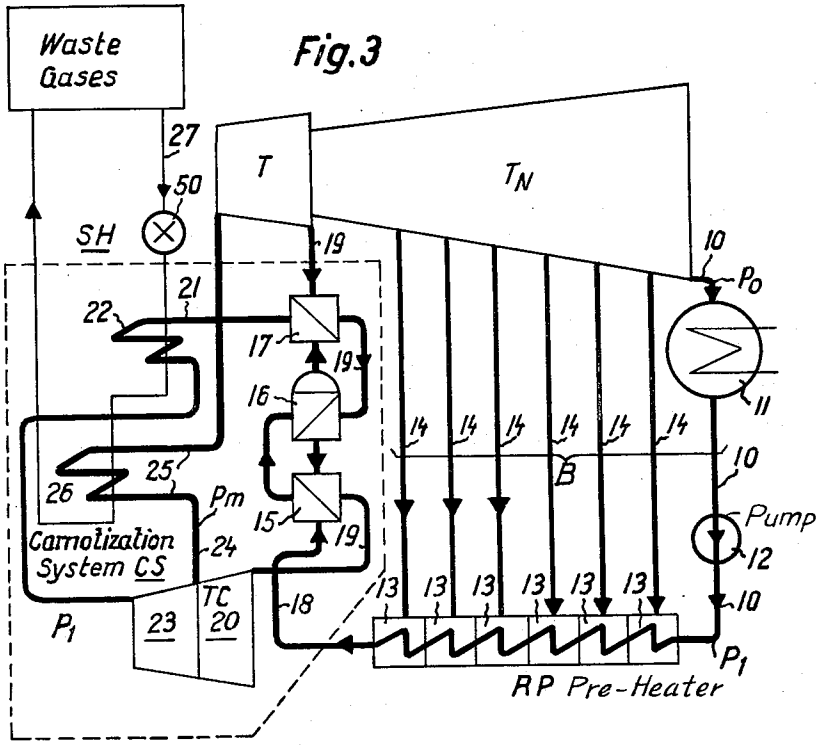
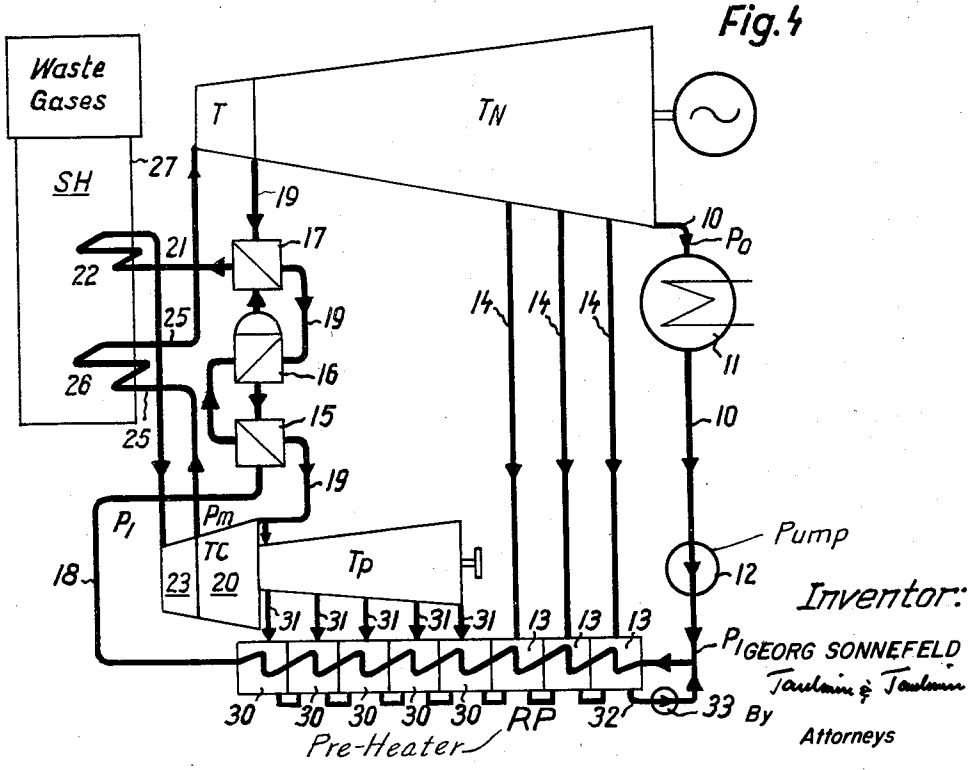


Fig. 4



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METHOD AND SYSTEM FOR THE CARNOTIZATION OF STEAM CYCLIC PROCESSES

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Fig.6

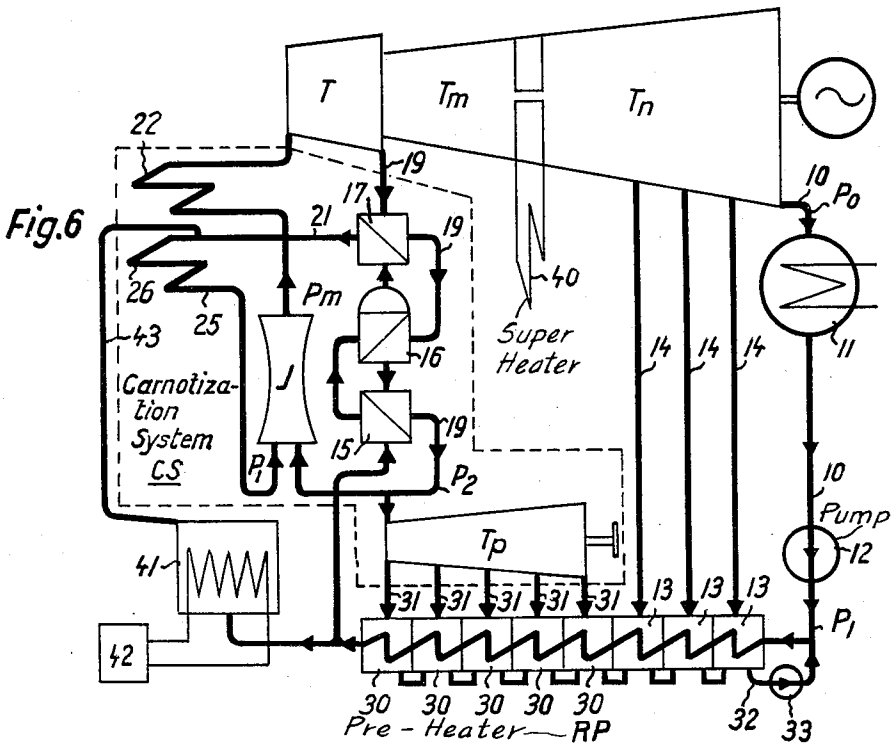
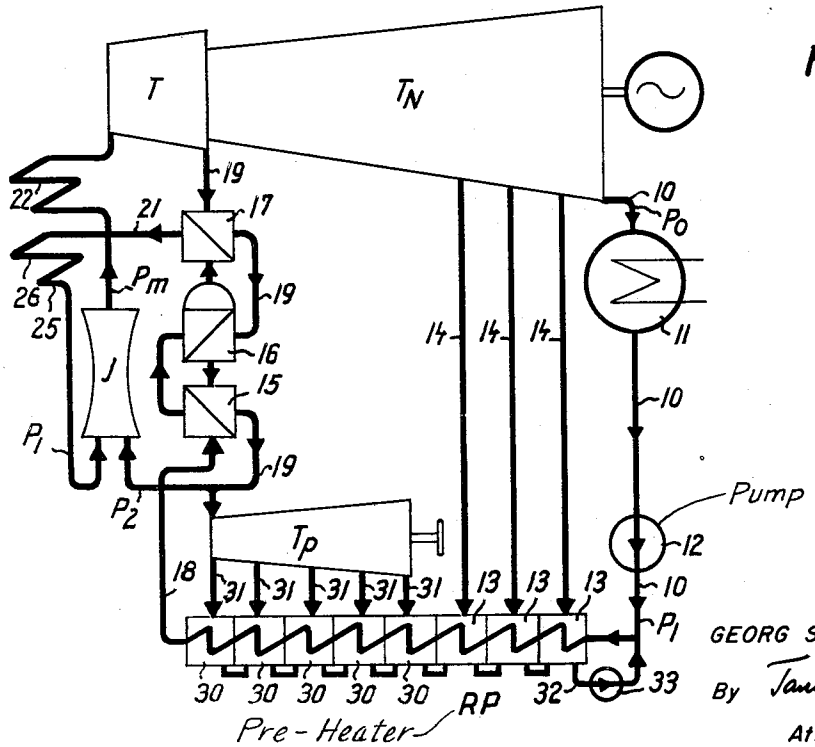


Fig.5



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United States Patent Office 2,964,910 Patented Dec. 20, 1960

2,964,910

METHOD AND SYSTEM FOR THE CARNOTIZATION OF STEAM CYCLIC PROCESSES

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Filed Apr. 11, 1957, Ser. No. 652,186

Claims priority, application Germany Apr. 13, 1956

8 Claims. (Cl. 60—67)

The present invention relates to a method and a system for the carnotization of steam cyclic processes.

It is the general object of my invention to provide for a method and a system of rendering steam cycles more efficient by making an actual cycle process approach the ideal conditions of the "Carnot" cycle to a higher degree than with the known carnotization methods.

It is another object of the invention to provide for an improved method and a system of carnotizing in a multi-stage steam cycle the stages of preheating, evaporation and at least a portion of superheating.

It is a further object of the present invention to provide for a method and a system for equalizing pressures in a steam cycle by bringing gases of different pressures to a common, intermediate pressure level while maintaining the entropy of these stages substantially constant.

It is still another object of the present invention to provide for a method and a system of the nature just indicated, which are simple, inexpensive and more efficient than the known methods and systems.

It is a final object of the present invention to provide for a method and a system of the nature just indicated in which shock losses and oscillations of the working medium are avoided, thereby maintaining the entropy of the corresponding process stages substantially constant.

It is well known that steam cycle processes are rendered more efficient if the steam cycle is "carnotized," i.e. if the conditions prevailing in the particular cycle are caused to more closely approach the conditions of the well known "Carnot" cycle.

If the heat taken in, for instance, by a steam boiler, is designated by Q_1, and the heat given out, for example by the condenser is Q_2, and if, furthermore, T_1, is the highest and T_2 the lowest temperature in the entire process, then the degree of efficiency (η_c) of the ideal "Carnot" process can be expressed by the equation

$$\text{(A)} \qquad \eta_c=(Q_1-Q_2)/Q_1$$

and also

$$\text{(B)} \qquad \eta_c=(T_1-T_2)/T_1$$

In actual steam cycles, the lowest temperature T_2 depends upon the temperature of the coolant for the condenser. The actual value of temperature T_1 is more difficult to ascertain because the total heat taken in during the process is in most cases not supplied at a single cycle step but rather, this heat is supplied to the steam cycle in various stages, for instance as heat of evaporation, by super-heating the steam and by pre-heating the feed water. For that reason, the highest temperature T_1 in the above Equation B must be replaced by a medium temperature T_m which is of course smaller than T_1 and comprises the various temperatures corresponding to the various amounts of heat supplied to the steam cycle. In an actual steam cycle process

the efficiency η can be computed on the basis of the equation:

$$\text{(C)} \qquad \eta=(T_m-T_2)/T_m$$

The ratio T_m/T_1 indicates the degree of carnotization of the process. Only if carnotization is complete, i.e. if the efficiency of an actual process is identical with the efficiency of the corresponding "Carnot" process, the $T_m=T_1$ and the above equation reads

$$\text{(D)} \qquad \eta=(T_1-T_2)/T_1$$

In the circulation of the "Carnot" process the working medium (such as, for instance, steam) must be brought cyclically from the temperature T_1 to temperature T_2 and back from the temperature T_2 to temperature T_1. In an actual process, however, which is to approach the efficiency of the ideal "Carnot" process to the highest degree possible, these changes in temperature of the working medium must be brought about exclusively by changes in the inner energy of the system without any addition of energy to the system from an external energy source or dissipation of energy from the system to the outside. If the inner heat energy, or "enthalpy," is insufficient to assure the completion of the cycle process from T_1 to T_2 and back to T_1 and accordingly additional heat must be supplied to the system from the outside, then the degree of carnotization is necessarily smaller than the value one, i.e. Equation C reads

$$\eta=\frac{T_1-T_2}{T_1}<1$$

The cycle of an actual carnotized process involves a mechanical transformation of energy. Enthalpy is first transformed into work which process is accompanied by an "isentropic" expansion, i.e. entropy remains constant. This work can actuate mechanical transformation means, such as for example, means for compressing the working medium (e.g. steam) whereby the work is re-converted to enthalpy.

According to the Ericson and Stirling cycles the inner heat circulation is performed by circulating air which absorbs and releases heat as the case may be so that there is no mechanical conversion but an exclusive exchange of heat. The adiabatics of the Carnot cycle are thus replaced by constant volume curves in the Stirling cycle, and by constant pressure curves in the Ericson cycle. The efficiency of both cycles is identical with that of the "Carnot" cycle.

A greater degree of efficiency of the steam cycle can thus be accomplished in various manners: the energy cycle may be carried out with the aid of mechanical means or with heat exchange means. It is, of course, also possible to combine both systems and to have the inner circulation performed partly by mechanical reconversion means and partly by heat circulation and heat exchange means. It is quite evident that in multi-stage processes the efficiency of the entire process depends on the number of individual stages that have been subjected to a carnotization. In the case of the steam cycle, the carnotization may be near to perfect or quite imperfect depending on which and how many of the various phases such as, for instance, pre-heating, evaporation and super-heating have been carnotized.

A limited carnotization is known in the art, which carnotization is particularly applied to the pre-heating stage or stages of the process, and which is known as a multi-stage regenerative or bleeding cycle. In the bleeding cycle, steam is drawn from the turbine at one or more pressure stages of the latter and used to heat the feed water (see Marks' "Mechanical Engineers' Handbook," 5th ed., page 326).

In steam cyclic processes, in which the stage of evapo-

ration is carnotized, only a portion of the working medium is made to change its state of aggregation from the liquid to the gaseous state and vice versa, while the remaining portion of the working medium retains its state of aggregation as a steam or gas. One of the main problems accompanying the carnotization of this stage of evaporation now is to accomplish the recompression of that portion of working medium which remains in the gaseous state of aggregation without supplying additional energy from the outside to the system, because such supply causes the loss of some or all of the efficiency gained by the carnotization.

This problem has not been satisfactorily solved in the art. The attempt has been made to carnotize the preheating and the evaporation state of the steam cycle. But in all these known processes of the regenerating or bleeding cycle type the steam tapped from the turbine must be recompressed, and this compression is carried out in the art, for instance, by a compressor whose energy is supplied by the main turbine.

All these known methods are, however, unsatisfactory and do not accomplish a sufficient degree of carnotization of the steam cycle process. Thus, the ratio between the recompressing work and the output is relatively high and may be up to 100%, which requires in the entire known systems, for instance the Field system, very expensive and complicated machinery and tends to make this known method uneconomical. Attempts to employ less expensive pressure adjusting machinery have failed in the past.

The efficiency of the known, actual steam cyclic systems is further reduced by the fact, that the working medium is conventionally introduced into the turbine at the highest temperature occurring in the cycle. Therefore, for the conversion of this heat corresponding to the highest temperature of the cycle (T_1) into kinetic energy, a high pressure turbine must be provided. It is, however, also well known that the initial, high pressure turbine is less efficient than the subsequently arranged medium and low pressure turbines. The higher the entrance pressure, the more pronounced is this relative inefficiency of the high-pressure turbine. Consequently, due to the necessity of using the high pressure turbine for the aforesaid conversion, the efficiency of the known systems is unsatisfactory.

The above-stated objects are achieved and the efficiency of carnotized cyclic systems is substantially improved for the above-mentioned known systems by the method and system of the present invention which comprises the steps and means, respectively, for the carnotization of a cyclic process composed of a primary cycle and at least one secondary cycle of a working medium, preferably steam, which secondary cycle contains steps of expansion and compression and is superimproved on the primary cycle in such a manner that, firstly, the heat given off from the secondary cycle does not only preheat and evaporate, but also additionally superheats the condensate of the working medium in the primary cycle; that, secondly, the combined pre-heating, evaporating and superheating effects of the heat given off by the secondary cycle are caused to occur at pressures in the primary cycle, which are always higher than the corresponding maximum pressure in the secondary cycle, and that, thirdly, the lowest pressure occurring in the secondary cycle is always higher than the lowest pressure occurring in the primary cycle.

It is an important feature of the invention that the steam or other vapor generated from a condensate of the working medium in the primary cycle is further superheated, for instance, by means of the heat of waste or flue gases, and is then expanded to decrease its pressure to the level of the highest pressure of the secondary cycle, whereby the energy released is at least sufficient to do the compression work required in the secondary cycle.

It is another feature of my invention, that the amounts of steam or other vapor in the primary and secondary cycles are expanded and/or compressed separately and are then brought together, whereupon they are jointly further superheated, for instance in the aforesaid manner, and are then expanded, in mixture with each other, to do mechanical work, whereafter the working medium is again separated into portions passing through the primary and secondary cycle, respectively.

According to yet another feature of my invention, the regenerative pre-heating of the cyclic process to be carnotized is supplied at least partially by the already expanded working medium in the secondary cycle, part of which expanded medium is divided from the secondary cycle prior to the compression stage in the latter; the output of which divided part of the expanded medium is additionally used for the aforesaid compression in the secondary cycle.

The method and system according to my invention are further characterized by the step of controlling the output of the cyclic process by changing the output of the cyclic process by changing the pressures in both the primary and secondary cycles at constant temperatures, while maintaining the ratio of pressure in the primary cycle to pressure in the secondary cycle at a given load substantially constant.

The system according to my invention comprises a primary cycle for the flow of the working medium in which cycle there is provided a machine for expanding the working medium to thereby obtain mechanical work. This machine may be of any conventional type, such as for instance a piston steam engine, but is preferably a steam turbine. The primary cycle may further contain the conventional condensing and pumping means for circulating the working medium as well as pre-heating means and means for carnotizing the pre-heating stage of the primary cycle. Finally, the primary cycle passes through the system according to the invention for pre-heating, evaporating and superheating the working medium on the pressure side of the primary cycle feed pump, i.e. substantially at maximum pressure and before the working medium in the primary cycle enters the expanding, compressing and combining means at the juncture of the primary and secondary cycles. The system according to the invention further comprises the necessary heat exchange means between the secondary cycle and the primary cycle for pre-heating, evaporating and superheating the latter with the aid of heat derived from the secondary cycle. The system according to the invention comprises further additional superheating means for superheating the primary cycle working medium after its superheating by heat from the secondary cycle, and superheating the combined working media from the primary and secondary cycle after their combination in the aforesaid expanding, compressing and combining means forming the juncture of the primary and secondary cycles.

These latter means may be a combined turbo compressor, or, preferably, a pressure equalizing jet nozzle of the kind described in my copending patent application Serial No. 652,187 filed of even date.

The invention will be better understood by the following detailed description of the accompanying drawings, wherein

Figure 1 is a diagram illustrating the functional interrelation of temperature, entropy and pressure in the carnotized cyclic process of the invention at subcritical pressures;

Figure 2 is a diagram illustrating the functional interrelation of temperature, entropy and pressure in the carnotized cyclic process of the invention at supercritical pressures;

Figure 3 shows in a schematic view one embodiment of the system according to the invention applied to a conventional turbine arrangement, in which embodiment the expanding, compressing and combining means at the

juncture of the primary and secondary cycle consist of a turbine compressor;

Figure 4 illustrates schematically another embodiment of the system according to the invention similar to that shown in Figure 3;

Figure 5 illustrates schematically a preferred embodiment of the system according to the invention, in which the expanding, compressing and combining means at the juncture of the primary and secondary cycle consist of a pressure equalizing jet nozzle; and

Figure 6 illustrates yet another embodiment of the system according to the invention similar to that shown in Figure 5.

Referring now to the drawings more in detail and turning first to Figure 1, T represents the temperature and s the entropy axis. The first steam cycle (I) which is to be carnotized according to the method of the present invention, is indicated by the curve $a—d—e—g—h—i—l—n—a$.

If an intermediary super-heating step is to be applied during the flow of the working medium through the working turbine T, such super-heating must take place outside the combined flow and exclusively in the primary cycle flow of the working medium through the low pressure part of the work turbine means. In this case the primary cycle is indicated by curve $a—d—e—g—h—i—l—m—n'—a$.

If the preheating stage $a—o'$ of this primary cycle is carnotized by a bleeding and regenerative preheating cycle (B and RP in Figure 3), the process is then defined by: $a'—o'—d—e—g—h—i—l—m—n$. Without the above mentioned final intermediate superheating the process is defined by $a'—o'—d—e—g—h—i—l—n'$. In addition, the secondary steam cycle is defined by $o—p—i—k$. This cycle is confined to the area of super-heated working medium above the saturation limit curve $a—y—g$, indicated by a heavy line and, accordingly, no change in the state of aggregation of the working medium occurs during this secondary cycle. Along the lines $h—i$ and $i—k$ all values and changes occur simultaneously in the primary and secondary cycles, and both cycles can therefore be conducted together in the same path.

The heat released by the secondary steam cycle is used in order to effect the pre-heating of the working medium in the primary cycle from o' till d, the evaporation, at a pressure p_1, from d to e, and the super-heating from e to f. For this purpose, the amount of steam G_I passing through the primary steam cycle and the amount of steam G_{II} in the secondary steam cycle must have a ratio $G_{II}:G_I$, equal to the ratio of entropy differences (which correspond to the amount of steam) $(s_4—s_2):(s_8—s_6)$.

When changing correspondingly the scale of entropy by dividing $(s_4—s_2)$ by $(s_8—s_6)$, but maintaining the same temperature scale, the secondary cycle $o—p—i—k$ is changed to $o'—p'—i'—k'$. As can now be easily seen from Figure 1, heat is transferred from the latter cycle to the primary cycle to effect the pre-heating of the working medium up to the temperature of evaporation at the pressure p_1' along curve piece $o'—d$. The flow of the working medium in the first cycle is countercurrently to that in the second cycle during this exchange of heat along curve $o'—d$. The transfer of the amount of heat characterized in Figure 1 by the area $3—d—k'—4$ from the secondary cycle to the primary cycle results in an increase of entropy $\Delta s=s_5—s_4$, because the amount of heat of evaporation and super-heating represented by area $3—d—e—f—5$ is equal to the amount of heat represented by area $3—d—k'—4$. From f to g and from l to m the heat for super-heating the working medium in the primary cycle is supplied through a special super-heating device SH, whereas the super-heating of the working media in both the primary and the secondary cycle from p' to i' and from h to i can be effected in a common super-heater, because the steam pressures of both media are identical and these two stages of both cycles are conducted together along the same path.

The primary cycle has its highest pressure at p_1 and its lowest pressure at p_0, at which latter pressure the working medium, for instance steam, condenses. The secondary cycle has a pressure range between p_2 and p_3. The expansion i' to k' and the compression o' to p' have the pressure ratio p_2/p_3.

As has been stated hereinbefore in accordance with the invention, the pressure p_1 of the primary cycle, at which the combined pre-heating, evaporating and superheating effects of the heat released from the secondary cycle along $o'—d$ are realized in the primary cycle, is always greater than the maximum pressure p_2 of the secondary cycle. Furthermore, the lowest pressure p_3 of the secondary cycle is always higher than the lowest pressure p_0 of the primary cycle.

Whereas Figure 1 illustrates the carnotization according to the invention for subcritical pressures, Figure 2 illustrates the same for pressures which exceed the critical pressure. The primary cycle is defined by $a'—m'—d—e—f—g—i—k—l$. The secondary cycle $m—n—g—h$ is changed to $m'—n'—g'—h'$ corresponding to the ratio of the circulating amounts of steam

$$G_{II}:G_I=(s_4—s_2):(s_8—s_6)$$

with a corresponding change of the entropy scale. Between h' and m' of the secondary cycle, the amount of heat expressed by area $2—m'—h'—4$ is countercurrently transferred to the primary cycle, wherein this heat appears as the amount represented by area $2—m'—d—5$. The increase in entropy is again designated by Δs. The actual amount of Δs is smaller in Figure 2 than in the example illustrated by Figure 1. This means that the efficiency of heat transfer from the secondary cycle to the primary cycle is increased, i.e. more favorable, when increasing the pressure p_1 so as to operate in the supercritical instead of the subcritical pressure range. By supercritical and subcritical pressures I mean pressures above and below, respectively, the pressure of the working medium at the critical point.

The heat supply from the outside, for instance from fuel or waste gases, is taken in by the secondary cycle along curve $n'—g'$, and by the primary cycle along $d—e$, $f—g$ and $i—k$. Again, during the stages $n'—g'$ and $f—g$, the working media in the primary and secondary cycles can be conducted together along the same path, since their pressures are identical. The heat is dissipated during the stage of condensation from l to a'. The primary cycle has a pressure range from p_1 to p_0, and the secondary cycle has a pressure range from p_2 to p_3. Consequently, both the expansion $g'—h'$ and the compression $m'—n'$ have again a pressure ratio of $p_2:p_3$.

If the final intermediate super-heating is dispensed with, the last stage of the cycle illustrated in Figure 1 will be $i—n'$ (expansion) and $n'—a'$ (condensation). The last stage of the cycle illustrated in Figure 2 is from $g—l'$ (expansion) to $l'—a'$ (condensation).

According to the invention, the pressure p_1 is so chosen that the expansion down to the pressure p_2 produces an amount of energy which is sufficient to cover the compression of the working medium of the secondary cycle, for instance steam which has given off its heat, which steam is again compressed from pressure p_3 up to pressure p_2. The recompressed working medium thereby attains again the pressure p_2 which is the entrance pressure at the turbine.

In Figures 3 to 6 a number of embodiments of the system according to the invention are illustrated, by way of example.

The system shown schematically in Figure 3 of the drawings comprises a conventional turbine arrangement in which T designates the main work performing turbine. T_n indicates the low pressure stage or stages of the main turbine. From the low pressure stage T_n of the

main turbine, a line **10** conveys exhausted working medium of the primary cycle at minimum pressure p_0 through condenser **11** to a feed pump **12** and from there at maximum pressure p_1 through the regenerative multi-stage pre-heater RP to the carnotization system CS according to the invention. The chambers **13** of the regenerative pre-heater RP receive, as heating agent, steam through bleeding lines **14** from the low pressure stage T_n of the main turbine.

The carnotization system according to the invention comprises in successive arrangement a pre-heating device **15**, an evaporating device **16** and a super-heating device **17**. Through a line **18** the working medium passes from the regenerative pre-heater RP through devices **15**, **16** and **17**, while through a line **19**, the working medium of the secondary cycle is conducted from an outlet at the end of the main turbine T where the latter opens into the low pressure stage T_n thereof, through devices **17**, **16** and **15** countercurrently to the flow of working medium of the primary cycle through these devices and onward to the compressor side **20** of a turbo compressor TC.

From the super-heating device **17** the working medium of the primary cycle is conducted through a line **21** via an external super-heater **22** to the expansion side **23** of the aforesaid turbo compressor TC. In the turbo compressor TC, pressure of the working medium is decreased by expansion from pressure p_1 to a medium pressure p_m, while on the compressor side **20** of turbo compressor TC, the working medium of the secondary cycle is compressed so as to raise its pressure from p_3 to p_n.

This medium pressure p_m which is common to the working media of both cycles at their juncture in outlet **24** of the turbo compressor TC, is equal to the maximum pressure p_2 of the secondary cycle.

From outlet **24** a line **25** conducts the combined working media of both cycles at the common pressure $p_m=p_2$ to another external super-heater **26** and from there to the inlet of main turbine T.

It is therefore a characteristic feature of the method according to my invention that the working medium is fed to the main work performing turbine T at a medium pressure p_m which is below a maximum pressure p_1 of the primary cycle.

External super-heaters **23** and **26** are heated by means of a waste gas or hot fuel gas line **27**.

It is a further advantageous feature of the method according to my invention that, as will be easily understood from the above description of the system shown in Figure 3, the working medium is at maximum pressure p_1 outside the work performing turbine means, and that only that portion of the working medium passing through the primary cycle must be brought to this maximum pressure.

In contrast thereto, all known methods require that the entire working medium be brought to maximum pressure and introduced in this state into the work performing turbine.

A further important advantage of the method according to my invention resides in the fact that, in the secondary cycle, the portion of the working medium derived from the main turbine T is maintained at a substantially constant pressure p_3, and that only this portion of the working medium passing through the secondary cycle must be brought from this pressure p_3 to a medium pressure p_m by means of an equalizing device such as a turbo compressor or a pressure equalizing jet nozzle. This fact distinguishes the method and system common to my invention from those known in the art, in which a part of the working medium of the primary cycle is evaporated in the secondary cycle, thereby requiring compressor means of a much larger value.

The embodiment of my invention shown in Figure 4 is similar to that shown in Figure 3. Like reference numerals in this figure designate like parts.

Furthermore, the regenerative pre-heating device RP is supplied with heating agent through bleeding lines **14** from the low pressure stage T_n of main turbine T, but also comprises pre-heating stages **30** which receive their heating agent in the form of a portion of the working medium passing through line **19** of the secondary cycle, which portion is divided from the secondary cycle prior to the entrance to the turbo compressor TC. This divided portion is introduced into a pre-heater turbine T_p in which it is expanded and at intermediary stages of which bleeding lines **31** lead to pre-heating chambers **30**. The work gained by the expansion of the divided portion of the working medium in turbine T_p can be used for operating the compressor side **20** of turbo compressor TC. Consequently, the turbine side **23** of the turbo compressor can be built smaller.

The condensed working medium acting as heating agent in the regenerative pre-heater RP is conducted through a line **32** to a condensate pump **33** which re-introduces the divided portion of the working medium from the secondary cycle into the primary cycle prior to the regenerative pre-heater RP.

This has the advantage that less working medium has to be compressed in the compressor side **20** of the turbo compressor TC, so that the required compressor output is smaller.

Moreover, while in the bleeding lines **14** from the left side of the low pressure stage T_n in Figure 3, the working medium conducted to the regenerative pre-heater RP is still substantially super-heated which represents a waste of energy, in the embodiment shown in Figure 4, these lines **14** on the left side are replaced by the pre-heater turbine T_p which is supplied with a working medium whose heat content has already been largely exchanged in devices **17**, **16** and **15** so that the working medium is approximately just above the saturation limit. Thus, the pre-heater turbine is loaded substantially with saturated steam which is more favorable for the regenerative pre-heating leading only to a minimum increase of entropy, and generally raises the degree of efficiency of the entire process. Consequently, the system as illustrated in Figure 4 does not require special cooling devices which are necessary in the conventional regenerative pre-heating methods.

Yet another embodiment of the invention is illustrated in Figure 5 of the drawings. Also in this figure like reference numerals designate the same parts as in the preceding Figures 3 and 4. However, the pressure equalizing device in this embodiment is constituted by a pressure equalizing jet nozzle J of the type described in my co-pending patent application Serial No. 652,187, filed April 11, 1957.

In this jet nozzle, the pressure p_1 of the working medium of the primary cycle and the pressure p_3 of the working medium of the secondary cycle are equalized to attain a common medium pressure p_m equal to the maximum pressure p_2 of the secondary cycle.

When a turbo compressor is used as the pressure equalizing device in the system according to my invention, it is permissible and of advantage to use supercritical pressures as the maximum pressure p_1 of the working medium in the primary cycle between the feed pump **12** and the entrance to the turbine compressor TC. Since this turbine compressor is an independent unit, it can be laid out for very high speeds in the order of 18,000–20,000 r.p.m. and more. It is often desirable to operate with large volume turbines, and in this case a large volume turbo compressor can be operated at a higher degree of efficiency if fed with a working medium of supercritical pressure.

On the other hand, the degree of efficiency of the pressure equalizing device and therewith of the entire process is further increased, in particular at higher outputs in the order of 50,000–100,000 kilowatts and more,

if the above-mentioned equalizing jet nozzle is used instead of the turbo compressor.

Finally, Figure 6 illustrates an embodiment of the system according to the invention similar to that shown in Figure 5 in which there is further provided an intermediary super-heating device **40** which is disposed in the low pressure stage T_n of the work performing turbine, i.e. subsequent to and outside of the main turbine T through which the working media of both the primary and the secondary cycle flow. This intermediary pre-heating device thus serves only for superheating the working medium in the primary cycle passing through the low pressure stage prior to being bled through the lines **14**. Furthermore, and also exclusively in the primary cycle there is provided an evaporator **41** which is heated externally by heating means **42** and is connected in parallel to devices **15**, **16** and **17** in the primary cycle, by means of a by-pass line **43** leading from the discharge end of the regenerative pre-heater RP to a point intermediate the discharge end of super-heating device **17** and the entrance to super-heating device **22**.

This evaporator **41** is of particular advantage for starting operation of the entire system. Also it may serve for balancing fluctuations in the load of the process.

It is a further important advantage of the system according to my invention that no control devices are required between the exit of the pressure equalizing device and the entrance to the work performing turbine. Pressure losses are thereby avoided.

The medium pressure turbine T_m used in the method of the invention just described produces the same amount of work as the combined high pressure and medium pressure turbines used in the methods known in the art.

The method according to the invention just described is suitable for both very high and very low pressure ranges, i.e. for high pressures of about $p_1=300$ atmospheres corresponding to $p_m=120$ atmospheres; and, on the other hand $p_1=10$ atmospheres and $p_m=4$ to 5 atmospheres or even less. It is also suitable for steam engines of small capacity but relatively high efficiency and of a simple and inexpensive construction, as for example for the propulsion of vessels etc.

It is to be understood that the examples shown in Figures 3 to 6 are but a few of many arrangements, which can be based on the method of the invention.

By employing the method of the invention, these cyclic processes yield a much higher degree of efficiency than the known processes, such as, for instance, the Clausius-Rankine steam cycle process. Although the highest pressure is the same as in the known processes, the main turbine is operated at a lower pressure and a greater entrance volume of the steam. This results in a higher efficiency of the turbine. At the same time, the entire construction of the steam cycle system of the invention is less expensive, because the expensive high pressure turbine is replaced by the far more economical medium pressure turbine.

The relatively low entrance pressure at the work performing turbine permits an easier control of high temperatures, for instance in the order of 600 to 650° C., and makes it possible to better exploit the advantages offered by these high temperatures. Due to the carnotization of the new method, every increase in temperature applied in the system according to the invention shows the same favorable influence on the degree of efficiency as, for instance, the gas turbine.

In the known processes, an increase in temperature, when all other conditions remain unchanged, leads to a smaller increase in the degree of efficiency. The new method leads to a better degree of efficiency because, at given maximum temperatures of the primary cycle, the main work performing machine can be operated at a considerably lower pressure, and, on the other hand with a substantially larger loading volume of the working medium than the conventional machines.

The overall construction of the system becomes less expensive since the high pressure turbine can be replaced, in the preferred embodiment of the system according to my invention, by the much cheaper pressure equalizing jet nozzle, while the increase of the loading capacity of the main turbine in no way consumes the savings achieved by the use of the jet nozzle.

The low pressure stage of the turbine requires fewer bleeding lines, thus permitting a simpler construction and more favorable conditions of flow in the turbine stage. Toward the discharge end of the low pressure stage a few bleeding lines are preferably retained to provide for withdrawal of condensed water in this part of the turbine.

Finally, the system according to the invention can be controlled most favorably by changing the maximum pressure p_1 in the primary cycle. This can be achieved by changing the pressure of the feed pump **12**. In order to maintain the degree of efficiency of the secondary cycle constant, the pressure ratio $p_2:p_3$ is preferably not changed, even though the pressure in the primary cycle and together therewith the amount of working medium passing through a given cross sectional area during the time unit is changed.

If pressure in the primary cycle is changed, the rate of super-heating in external super-heaters **22** and **26** must be correspondingly adjusted. This is achieved, for instance, by control means **50** provided in the heat supply line **27** for these super-heaters.

Once a desired maximum pressure p_1 in the primary cycle has been attained by these measures just mentioned, it is very important that all cycles of the system operate with full cross sectional areas of their conduits, i.e. without any throttling effects.

If the load in the entire process is, for instance, to be decreased from full load to half load, this can be achieved by a corresponding change in the speed of the feed pump without any throttling. Consequently, the change in the degree of efficiency proportioned to the load shows no abrupt, but only a very gradual decrease as would be the case if control would be effected by throttling.

The method according to my invention is generally suitable for all outputs, pressures and temperatures, and even small arrangements of, for instance, 5,000 kilowatts and less can be operated with a relatively high degree of efficiency of above 30% and more, which has hitherto been possible only when using gas turbines. Other turbine arrangements of known type and the same output only permit to achieve efficiency degrees of 22 to 26%.

On the other hand, if very high outputs of 50,000 to 100,000 kilowatts and more are desired, all known methods with the best carnotization thus far achieved and considering all losses from the combustion of the fuel, to current delivering terminals of the generator, do not attain a degree of efficiency above 30 to 40%.

In contrast thereto the method and system according to my invention permit to achieve, if all conditions are similar, a degree of efficiency of 44 to 45% while saving considerably machinery and material compared with the known methods.

It will be understood that this invention is susceptible to modification in order to adapt it to different usages and conditions, and, accordingly, it is desired to comprehend such modifications within this invention as may fall within the scope of the appended claims.

What I claim is:

1. A power plant operated by a working medium in a thermodynamic cycle process and comprising a work performing machine having a high pressure stage entrance for the admission of a heated working medium in a vapor state, a condenser for the working medium exhausted from said work performing machine, a regenerative pre-

heating device for preheating the condensate from said condenser, means for heating said preheated condensate to the vapor state, means for bleeding said working medium in a vapor state at an intermediate pressure stage of said work performing machine and circulating said bled working medium through a portion of said heating means for partially heating said preheated condensate, and a pressure equalizer device connected to said entrance of said work performing machine for increasing the pressure of the bled medium after circulation through a portion of the heating means and for decreasing the pressure of the vapor state of the heated working medium to a common medium pressure at which both working media are admitted concurrently into said entrance of said work performing machine.

2. A power plant, as claimed in claim 1, wherein said portion of the heating means comprises heat exchanger means traversed countercurrently by the preheated condensate and by the bled working medium, said heat exchanger means transmitting heat from the bled working medium to the preheated condensate and constituting means for successively preheating, evaporating and superheating the preheated condensate prior to the introduction of said preheated condensate into said pressure equalizer device.

3. A power plant, as claimed in claim 2, and further comprising superheater means for superheating the preheated condensate after said preheated condensate emerges from said heat exchanger means and before entering said pressure equalizer device.

4. A power plant, as claimed in claim 2, with said pressure equalizer device comprising a turbo compressor having an expansion side and a compression side, first conduit means connecting said superheater of said heat exchanger means to said expansion side, and second conduit means for connecting that portion of the heat exchanger means through which said bled working medium moves to the compression side of said turbo compressor.

5. A power plant, as claimed in claim 4, and further comprising a preheating turbine having an inlet and a plurality of bleeding outlets, said inlet being connected to the path of said bled working medium intermediate said heat exchanger means and said pressure equalizer device, said bleeding outlets being connected to said regenerative preheating device so as to conduct a portion of the bled working medium prior to its introduction into said pressure equalizing device, into said regenerative preheating device as a heating agent.

6. A power plant, as claimed in claim 5, wherein said preheating turbine is coupled with said turbo compressor so as to drive the compression side of the said turbo compressor jointly with the expansion side thereof.

7. A power plant, as claimed in claim 2, and further comprising super-heater means for the combined working media of the bled working medium and the preheated condensate as said combined working media flows from said pressure equalizer device to the high pressure entrance of said work performing machine.

8. A power plant, as claimed in claim 2, and further comprising an evaporator connected in parallel to said heat exchanger means intermediate said regenerative preheater device and said pressure equalizing device for the flow of the pre-heated condensate therethrough.

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