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(54) **METHOD FOR IMPROVING THE COMBINATION BETWEEN UN GAS TURBINE AND A STEAM CYCLE WITH AN ANOTHER NON FOSSILE SOURCE OF PRIMARY ENERGY.**

(57) Method for improving the combination between a gas turbine and a steam cycle with an another non fossile source of primary energy, consisting in recycling part of the residual heat of exhaust gases from the gas turbine (5) in order to reheat at intermediate pressure the main flow of the steam cycle. In order to complete the recycling, the remainder of the residual heat is used to heat liquids and, usually, steamings at maximum pressure of the cycle or at the characteristic intermediate reheating pressure. The steam cycle may be a water vapour cycle or a steam cycle which is a mixture of water and a less volatile fluid which vaporizes at the characteristic intermediate reheating pressure.

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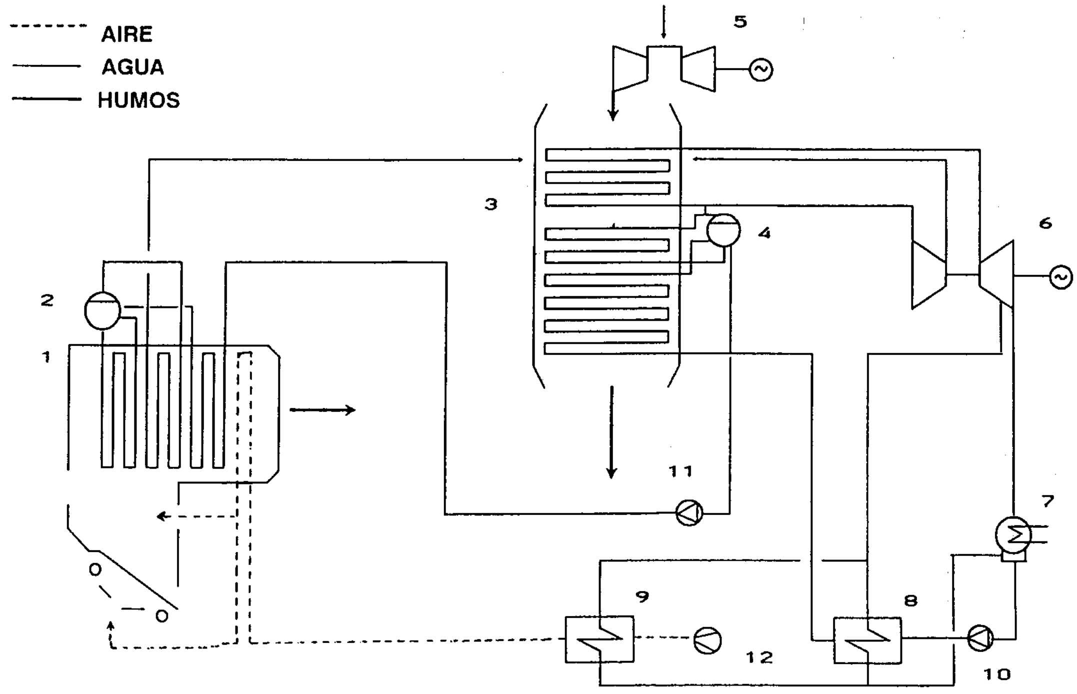

FIG. 1

The invention relates to a process for improving the combination between a gas turbine and a steam cycle that uses an additional non-fossil fuel as primary energy, applicable to electric power plants or electricity and thermal energy cogeneration plants employing at least two primary energy sources: an appropriated fuel for the gas turbine and an additional non-fossil fuel (refuse, geothermal, biomass, solar energy, etc.) for the steam cycle.

One of the basic characteristics of the simple steam cycle is determined by the strong tendency of superheated steam to approach the saturation line and enter the wet-steam condition during the expansion in turbine. This phenomenon considerably limits the techno-economical optimization of steam cycle's working parameters in each application, because, for a given temperature of steam superheating, a maximum working pressure that leads to a maximum wetness admissible in turbine exhaust can be determined. This is to limit the blade erosion problems which in turn mean maintenance costs and deterioration of efficiency with time. Both the limitation of the maximum working pressure and the presence of wetness in the expansion are efficiency-reducing factors of a steam cycle.

In order to improve the steam cycle efficiency, it is a usual practice in large power plants and, in general, provided that no techno-economical conditions advises against it, to carry out an intermediate reheating of steam, which allows the final wetness of expansion to be reduced and/or the maximum working pressure to be raised, all of which redounds in a notable improvement of efficiency and working conditions of the last turbine stages.

However, when it is intended with steam cycle to employ non-fossil primary energy sources, such as all types of refuse, biomass, geothermal energy, solar energy, etc., it often happens that the intermediate reheating of steam using this primary energy is not techno-economically feasible or that its advantages are very much reduced.

The invention proposed herein, applicable to all those cases in which the practice of intermediate reheating using the primary non-fossil energy source is not techno-economically feasible, gives a limited cost effectiveness or presents appreciable technical risks, consists in applying an intermediate reheating of the steam cycle by means of the waste heat of gas turbine exhaust gases in their high-temperature range.

With this system the following advantages are attained over the previous art:

- Reduction of the wetness in the steam turbine, with the subsequent efficiency increase in design conditions and diminution of blade deterioration, the latter additionally resulting in a greater conservation of design efficiency and reduction of maintenance costs.
- Optimum use of exergy available in gas turbine exhaust gases, resulting in a much higher global conversion efficiency of the combined cycle.
- Possibility of a much freer selection of maximum working parameters of the steam cycle, owing to the elimination or reduction of wetness problem in the turbine. This allows in many cases an increase of the maximum steam pressure and a reduction of exergy losses through heat transmission to the working fluid of the steam cycle, thereby increasing the efficiency.

On the other hand, the proposed schematic offers a number of advantages which, in general, considerably improve the conditions under which the reheating could be carried out using non-fossil primary energy, provided that this would be feasible, as here the steam reheating at relatively low pressure and temperature may be carried out near the steam turbine, in a typical heat recovery boiler, with clean gases, allowing the option of using finned tubes in the gas area and notably improving the thermal transfer, with or without intermediary fluid. Additionaly, the regulation is much simpler as the steam cycle can operate even when the gas turbine is out of service with an admissible temporary increase in the wetness in turbine and, if necessary, regulation of the steam turbine load. With all this, the techno-economical balance is much more favorable as compared with that of existing solutions. Naturally, the characteristic use of high-temperature waste heat of gas turbine exhaust gases for the intermediate steam reheating should be complemented with other partial uses of said waste heat in the low-temperature area of the same, as is usual (heating of condensates and feedwater vaporization, etc.) in such a manner that the recovery is exergetically optimized.

In line with this, three possible basic schematics of recovery, logically all with the characteristic steam reheating at intermediate pressure in the high-temperature zone of the gas turbine exhaust gases, are proposed. Any of these schematics allows the plant operation without consuming natural gas, as any conventional plant, using some the other non-fossil energy source.

The first schematic is the simplest and corresponds to a layout in which the rest of the low-temperature energy contained in exhaust gases is used exclusively for heating condensate and feedwater up to the inlet temperature to economizer of non-fossil fuel boiler. This schematic requires that the total energy contained in the gas turbine exhaust gases, in their cooling until their release to atmosphere, coincides with that required for the aforementioned heatings and for the characteristic reheating.

This schematic conditions and limits the gas turbine output for a given thermal output of the non-fossil fuel boiler, because of which it may be in general more convenient to use part of the gas turbine exhaust gases, after the steam reheating and before the feedwater heating, for carrying out at the maximum cycle pressure a vaporization of a part of water flow, which may be superheated either in the recovery boiler or together with the rest of the steam generated in the non-fossil boiler.

Finally, if, due to working conditions in the non-fossil boiler, the steam to be reheated leaves the high-pressure turbine in near-saturating conditions at this pressure, it is also possible to resort to a schematic in which steam is generated in the recovery boiler at the reheating pressure and is mixed with the rest of the steam to be reheated.

An important variant of this schematic consists in not limiting the operation to the intermediate reheating of steam, as described, but in generating a reheated mixture of vapors, of water and another substance having a higher boiling point than water, mixed in liquid phase with the vapor to be reheated in such a manner that, in this case, the intermediate reheating process is a combined process of vaporization-reheating wherein the less volatile substance vaporizes at variable temperature at the same time as the vapor mixture, gradually becoming richer in the less volatile substance, is reheated until all the less volatile substance has been vaporized, from which point the vapor mixture with the final composition continues being reheated, until reaching the maximum temperature of the intermediate reheating.

In the mixing process prior to the reheating described, a part of water can condense, thereby delivering energy to the less volatile substance which vaporizes in part until reaching conditions of thermodynamic equilibrium, in such a manner that in this case, before the aforementioned combined process of non-isothermal vaporization plus reheating begins, an isothermal vaporization of water and the other substance would take place. It is likewise feasible to simultaneously mix with the vapor to be reheated the less volatile substance and additional water, which would also vaporize isothermally with the less volatile substance, before the combined process of non-isothermal vaporization and reheating described begins.

The reheated vapor mixture thereby generated, once it has been expanded in the low-pressure turbine to the minimum cycle pressure, is still at a sufficiently high temperature as to deliver heat for various uses in the cycle itself (condensate heating, vaporizations at low pressure, etc.), or external uses (water heating for domestic or industrial purposes, air heating, etc.) at variable temperature. During this heat yielding process, there takes place the non-isothermal condensation of the less volatile substance which can thus be separated almost totally from the steam. Once it has been pumped and heated by means of energy yielded by the main vapor flow at turbine exhaust and/or waste heat of turbine exhaust gases, it is mixed again with the vapor to proceed to the intermediate reheating described. In this way, the steam with a very small proportion (about 5%) of the less volatile substance (function of the thermodynamic equilibrium for the conditions of minimum cycle pressure and temperature reached during the aforementioned non-isothermal condensation) passes to the final condenser where it is completely condensed. This allows an operation from said point virtually as a conventional steam cycle until it is again mixed with the less volatile substance in liquid phase.

This variant presents additional advantages over the use of pure steam, as follows:

- It increases the average specific heat of absorption of the vapor reheating, thus improving the thermal transfer.
- It allows a better adaptation of the form and slope of heat yield and absorption curves, especially when it is desired to increase the size of the gas turbine for a defined non-fossil primary energy source, reducing the exergy losses, with the advantage of having an additional parameter available to be freely defined within a certain range, such as the vapor mixture composition.
- It allows the utilization of the energy yielded in the condensation of the less volatile fluid at variable temperature in the vapor mixture for regenerative heating of condensates, generation of additional vapor at different pressure (for it to be admitted in the low-pressure turbine or expanded in another turbine body) and, eventually,a part of this energy may be used in cogeneration systems, all of which using energy of the cycle itself, with optimum practical exergy losses and without the need to resort to typical turbine extractions for these purposes.

Finally, it must be noted as another important advantage of the proposed invention that, by using at least two different types of primary energy, the efficiency obtained from these is higher than that achievable by the currently most advanced technology using them in the same proportions (independent or jointly) with any of the existing solutions, with the importance this means both economically and from the viewpoint of reduction in environmental impact involved in the electric power generation from these primary energy sources.

As examples of application of the process proposed herein, the case of a municipal solid waste (MSW) incineration plant with 1000 t/day capacity has been selected. The first example shows the application with a pure steam cycle and the second, the system working with a mixture of water and a thermal fluid which is a eutectic mixture of biphenyl and biphenyl oxide and will be called hereinafter TF.

As explained above, example 1 (figure 1) is the application of the process to a MSW incinerating plant working with a pure steam cycle.

The parameters of the steam cycle are typical values for this type of plants, except the maximum pressure in the furnace-boiler (1) which in this case is not limited by the final wetness in the expansion and, therefore, has been raised to an optimum value of 105 bar abs.

In furnace-boiler (1), feedwater is heated until saturation, vaporized and superheated at the maximum process pressure.

For the purpose of adjusting the available capacity of heat absorption in the recovery heat exchanger (3) to the heat yielding capacity of the exhaust gases of the gas turbine (5) chosen for this case, a vaporization of water is carried out at the pressure at which the proposed intermediate steam reheating is carried out. This slightly increases the exergy losses in the heat recovery but allows the optimization of the facility from the economical viewpoint.

The additional saturated steam obtained is mixed with that exhausted from the high-pressure body of steam turbine (6), which in this case is at a very similar temperature, and is reheated by the energy corresponding to the heat yield of higher temperature of gas turbine (5) exhaust gases before it is admitted to the low-pressure body of turbine (6).

The combustion gases of gas turbine (5), after the aforementioned reheating and vaporization, are used for heating the liquid from the temperature of preheating (carried out by means of an extraction from low-pressure steam turbine (6)) to the inlet temperature to incineration furnace-boiler (1).

The primary heating of condensate as already described and that of the combustion air are carried out by means of an extraction of steam in low-pressure turbine (6).

Example 2 (figure 2) shows an application of the process to a MSW incinerating plant working with a vapor mixture of water and TF.

This option has all the advantages of the vapor reheating using the exhaust waste heat of a gas turbine (16) plus those offered by the use of a vapor mixture of thermal fluid (TF) and water.

The schematic includes the partial heating of feedwater until saturation at the maximum pressure, the vaporization at this pressure and the superheating of the vapor to the temperature of 400 °C in furnace-boiler (13), in exactly the same way and under identical conditions as in example 1, except the small percentage of TF (4%) that accompanies the water, without hardly any practical incidence, which therefore has been neglected for the sake of simplicity of calculation, and the vapor is considered pure steam.

The heat recovery from the exhaust gases of turbine (16) is carried out in the following way:

- The highest-temperature area of the gases is recovered by heating the TF-Water vapor mixture in the working proportions, at the intermediate pressure of reheating, until the temperature conditions of admission to low-pressure turbine (17).
- The intermediate-temperature area of the gases is used for vaporizing the TF to the saturating conditions at the intermediate pressure of reheating. The vaporization at variable temperature of TF notably increases the heat transmission coefficients, making the implementation of this recovery more economic.
- The coldest area of gases is used for heating the condensates (TF and water) from the temperature at which they have been preheated by the waste heat of non-isothermal TF condenser (18) of the cycle.

The waste heat corresponding to the non-isothermal condensation of TF is used with very little exergy loss for the primary heating of combustion air and the primary heating of condensates (TF plus water). The energy of higher thermal level of this recovery is used for vaporizing a part of water at low pressure which is introduced into the expanding flow in turbine (19) through a partial admission.

Tables 1 and 2 show the basic results of the thermal balances for application examples 1 and 2, respectively. Figures 1 and 2 are the basic thermal schematics of the plant for application examples 1 and 2, respectively, which include the following elements:

Figure 1:

1. Furnace-boiler
2. High-pressure drum
3. Recovery heat exchanger
4. Intermediate-pressure drum
5. Gas turbine
6. Steam turbine

7. Condenser
8. Water heater
9. Air heater
10. Condensate pump
11. Feedpump
12. Combustion-air fan

Figure 2:

13. Furnace-boiler
14. High-pressure drum
15. Recovery heat exchanger
16. Gas turbine
17. Vapor turbine
18. Thermal fluid condenser
19. Vapor admission drum
20. Final condenser
21. Air heater
22. Water pump
23. Thermal fluid pump
24. Combustion-air fan

Figure 3 shows a temperature - heat exchange diagram in the furnace-boiler, valid for both application examples, wherein the following transformations are represented:

Figure 3:

25. Cooling of combustion gases in the furnace-boiler
26. Secondary heating of combustion air
27. Heating of the feed to the drum
28. Superheating of vapor
29. Vaporization in the high-pressure drum

Figures 4 and 5 show the temperature -heat exchange diagrams in the recovery boiler of the gas-turbine exhaust gases, for examples 1 and 2, respectively, wherein the following transformations are represented:

Figure 4:

30. Cooling of gas-turbine exhaust gases
31. Heating of condensate
32. Vaporization in the intermediate-pressure drum
33. Reheating of vapor at intermediate pressure

Figure 5:

34. Cooling of gas-turbine exhaust gases
35. Heating of water and thermal fluid (they are heated separately in parallel, although the transformation has been represented as a whole)
36. Isothermal vaporization of water and thermal fluid
37. Mixed process of vapor reheating with vaporization at variable temperature of thermal fluid
38. Reheating of vapor mixture

TABLE 1

- INCINERATING CAPACITY:	41,667	kg/h
- AVERAGE LHV OF REFUSE:	7,535 kJ/kg (1,800 kcal/kg)	
- THERMAL POWER OF REFUSE:	87,210	kWt
- COMBUSTION-AIR FLOW:	232,573	kg/h
- COMBUSTION-GAS FLOW:	258,415	kg/h
- MAXIMUM TEMPERATURE IN FURNACE:	1,100	°C
- ECONOMIZER OUTLET TEMPERATURE:	240	°C
- THERMAL POWER ABSORBED BY FLUID IN FURNACE-BOILER:	72,356	kWt
- AIR-HEATER OUTLET TEMPERATURE:	186	°C
- AIR-HEATER INLET AIR TEMPERATURE:	80	°C
- AIR-HEATER OUTLET AIR TEMPERATURE	150	°C
- THERMAL POWER YIELDED TO AIR IN AIR-HEATER:	4,543	kWt
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- ELECTRIC OUTPUT OF GAS TURBINE:	54,339	kWe
- THERMAL POWER OF GAS CONSUMED IN GAS TURBINE:	134,836	kWt
- GAS-TURBINE EXHAUST GAS FLOW:	589,107	kh/h
- GAS-TURB. EXHAUST GAS TEMPERATURE:	445	°C
- RECOVERY BOILER OUTLET GAS TEMP.:	95	°C

- POWER ABSORBED BY FLUID IN RECOVERY BOILER:	61,137	kWt

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- FURNACE-BOILER ECONOMIZER FEEDWATER TEMPERATURE:	212	°C
- VAPOR PRESSURE IN FURNACE-BOILER (HIGH PRESSURE):	105	bar abs
- HIGH-PRESSURE VAPOR FLOW GENERATED IN FURNACE-BOILER.	120,204	kg/h
- TEMPERATURE OF H-P VAPOR SUPERHEATED IN FURNACE-BOILER:	400	°C
- H-P TURBINE EXHAUST PRESSURE:	20.0	bar abs
- H-P TURBINE ISENTROPIC EFFICIENCY:	80	%
- H-P TURBINE EXHAUST TEMPERATURE:	213	°C
- POWER YIELDED BY FLUID IN H-P TURBINE:	9,860	kW
- L-P TURBINE ADMISSION VAPOR PRESSURE:	19.0	bar abs
- TEMPERATURE OF L-P VAPOR REHEATING IN RECOVERY BOILER:	420	°C
- VAPOR EXTRACTION PRESSURE FOR WATER AND AIR HEATERS:	0.60	bar abs
- EXTRACTION VAPOR FLOW:	16,610	kg/h
- L-P TURBINE EXHAUST PRESSURE:	0.07	bar abs
- L-P TURBINE ISENTROPIC EFFICIENCY:	85.0	%
- WETNESS IN EXHAUST:	7	%
- POWER YIELDED BY FLUID IN L-P TURBINE:	36,396	kW
- TOTAL POWER YIELDED BY FLUID IN TURBINE:	46,256	kW
- POWER ABSORBED BY FLUID IN PUMPING:	572	kW
- MECHANO-ELECTRICAL EFFICIENCY OF TURBOALTERNATOR.	97.5	%
- TURBOALTERNATOR ELECTRICAL OUTPUT:	45,100	kWe
- ELECTRIC POWER CONSUMED IN PUMPING:	635	kWe
- NET ELECTRICAL OUTPUT OF STEAM CYCLE:	44,465	kWe

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- POWER GENERATED ON ALTERNATOR TERMINALS:	98,804	kWe
- NET ELECTRIC POWER GENERATED:	98,169	kWe
- TOTAL THERMAL POWER CONSUMED (MSW + GAS):	22,046	kWt
- ELECTRICAL EFFICIENCY ON TERMINALS:	44.5	%
- NET ELECTRICAL EFFICIENCY OF GENERATION:	44.2	%

TABLE 2

- INCINERATING CAPACITY:	41,667	kg/h
- AVERAGE LHV OF REFUSE:	7,535 kJ/kg (1,800 kcal/kg)	
- THERMAL POWER OF REFUSE:	87,210	kWt
- COMBUSTION-AIR FLOW:	232,573	kg/h
- COMBUSTION-GAS FLOW:	258,415	kg/h
- MAXIMUM TEMPERATURE IN FURNACE:	1,100	°C
- ECONOMIZER OUTLET GAS TEMPERATURE:	240	°C
- THERMAL POWER ABSORBED BY FLUID IN FURNACE-BOILER:	72,356	kWt
- AIR-HEATER OUTLET TEMPERATURE:	186	°C
- AIR-HEATER INLET AIR TEMPERATURE:	80	°C
- AIR-HEATER OUTLET AIR TEMPERATURE:	150	°C
- THERMAL POWER YIELDED TO AIR IN AIR-HATER:	4,543	kWt

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- ELECTRIC OUTPUT OF GAS TURBINE:	70,373	kWe
- THERMAL POWER OF GAS CONSUMED IN GAS TURBINE:	174,622	kWt
- GAS-TURBINE EXHAUST GAS FLOW:	762,938	kh/h
- GAS-TURBINE EXHAUST GAS TEMPERATURE:	445	°C
- RECOVERY BOILER OUTLET GAS TEMPERATURE:	95	°C
- POWER ABSORBED BY FLUID IN		

RECOVERY BOILER:	79,177	kWt

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- FURNACE-BOILER ECONOMIZER FEEDWATER TEMPERATURE:	212	°C
- VAPOR PRESSURE IN FURNACE-BOILER (HIGH PRESSURE):	105	bar abs
- HIGH-PRESSURE VAPOR FLOW GENERATED IN FURNACE-BOILER.	120,204	kg/h
- TEMPERATURE OF H-P VAPOR SUPERHEATED IN FURNACE-BOILER:	400	°C
- H-P TURBINE EXHAUST PRESSURE:	20.0	bar abs
- H-P TURBINE ISENTROPIC EFFICIENCY:	80	%
- H-P TURBINE EXHAUST TEMPERATURE:	213	°C
- POWER YIELDED BY FLUID IN H-P TURBINE:	9,860	kW
- TF INJECTION RATE INTO RECOVERY BOILER:	180,306	kg/h
- TEMPERATURE OF TF INJECTED:	200	°C
- REHEATED L-P VAPOR MIXTURE FLOW:	300,510	kg/h
- FINAL TEMPERATURE OF VAPOR REHEATING:	400	°C
- L-P TURBINE ADMISSION VAPOR PRESSURE:	19.0	bar abs
- SATURATED STEAM ADMISSION PRESSURE TO L-P TURBINE:	0.6	bar abs
- SATURATED STEAM ADMISSION RATE:	19,800	kg/h
- L-P TURBINE EXHAUST PRESSURE:	0.07	bar abs
- L-P TURBINE ISENTROPIC EFFICIENCY:	85.3	%
- L-P TURBINE EXHAUST TEMPERATURE:	107	°C
- WETNESS IN EXHAUST:	1	%
- POWER YIELDED BY FLUID IN L-P TURBINE:	44,650	kW
- TOTAL POWER YIELDED BY FLUID IN TURBINE:	54,510	kW
- POWER ABSORBED BY FLUID IN PUMPING:	719	kW
- WASTE HEAT DELIVERED BY EXTRACTION FOR HEATING OF CONDENSATE AND COMB. AIR TO 80°C AND GENERATION OF ADMISSION VAPOR AT 0.6 bar abs:	26,292	kWt

- MECHANO-ELECTRICAL EFFICIENCY OF TURBOALTERNATOR.	97.5	%
- TURBOALTERNATOR ELECTRICAL OUTPUT:	53,147	kWe
- ELECTRIC POWER CONSUMED IN PUMPING:	799	kWe
- NET ELECTRICAL OUTPUT OF STEAM CYCLE:	52,348	kWe

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- POWER GENERATED ON ALTERNATOR TERMINALS:	122,721	kWe
- NET ELECTRIC POWER GENERATED:	121,922	kWe
- TOTAL THERMAL POWER CONSUMED (MSW + GAS):	261,832	kWt
- ELECTRICAL EFFICIENCY ON TERMINALS:	46.9	%
- NET ELECTRICAL EFFICIENCY OF GENERATION:	46.6	%

Since the nature of the present invention and a mode of its implementation have been sufficiently described, it is to be noted that, as a whole and in composing parts, changes in mode, materials and layout may be introduced provided that they do not substantially alter the characteristics of the invention claimed in the following pages.

Claims

1. A process for improving the combination between a gas turbine and a vapor cycle that uses an additional non-fossil primary energy source, in which the high-temperature area of the waste heat of the exhaust gases from the gas turbine is used for generating superheated vapor at the maximum pressure of the vapor cycle and for carrying out a reheating at intermediate pressure of the main flow of the vapor cycle.

2. A process, as per claim 1, in which the vapor cycle is a typical pure steam cycle.

3. A process, as per claim 2, in which the rest of the energy contained in the gas-turbine exhaust gases is used for heating at least part of the main flow of condensate and feedwater of the steam cycle.

4. A process for improving the combination between a gas turbine and a vapor cycle that uses a different non-fossil primary energy source, in which the high-temperature area of the waste heat of the gas-turbine exhaust gases is used for vaporizing part of the vapor cycle working fluid at an intermediate pressure and for carrying out a reheating of the vapor cycle main flow at the same intermediate pressure.

5. A process, as per claim 4, in which the vapor cycle is a typical pure vapor cycle and the saturated steam generated' at the intermediate pressure, by means of part of the heat of the gas-turbine exhaust gases, is mixed with the main steam flow at the same pressure before proceeding to the reheating at intermediate pressure.

6. A process, as per claim 4, in which a less volatile substance than water,that is mixed with the main flow rich in water at the intermediate pressure, is vaporized at intermediate pressure before the intermediate reheating, in such a manner that there takes place, in this case, a simultaneous mixed process of vaporization-reheating wherein a vaporization of the less volatile substance occurs at the same time as the reheating of the vapor mixture, until the maximum temperature of the reheating at intermediate pressure.

7. A process, as per claim 6, in which the vapor mixture generated is expanded from the pressure of the intermediate reheating to a lower pressure and the less volatile substance is separated from the vapor mixture at said lower pressure by its non-isothermal condensation, by means of the energy yielded in this transformation to the very vapor cycle, in an exclusive way or it being complemented with use for other purposes extrinsic to the vapor cycle.

8. A process, as per claim 6, in which the less volatile substance is not a pure substance but a mixture of substances having very similar vapor tensions for each temperature, in such a manner that, for the process purpose, it virtually behaves as one pure substance.

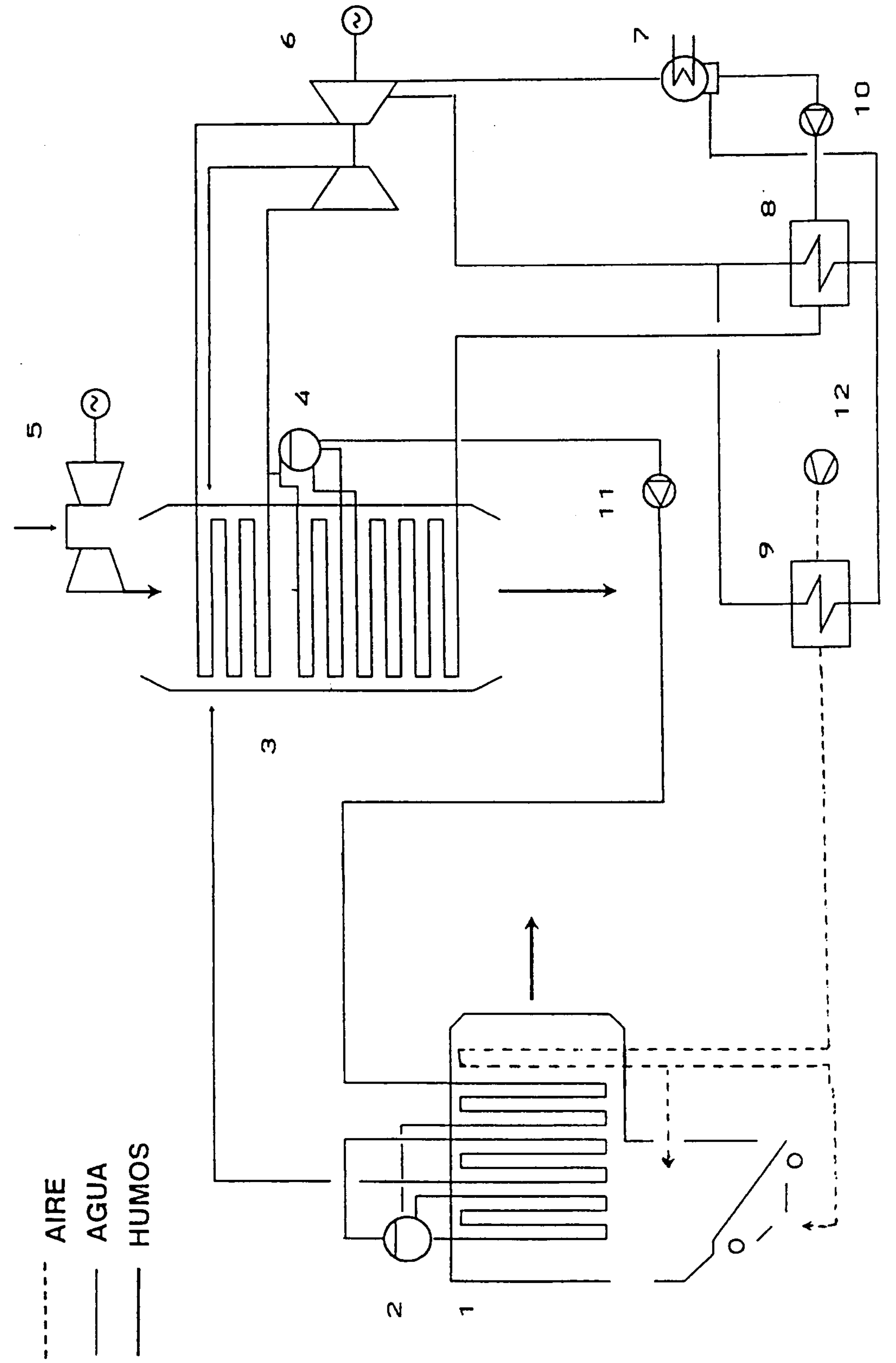

FIG. 1

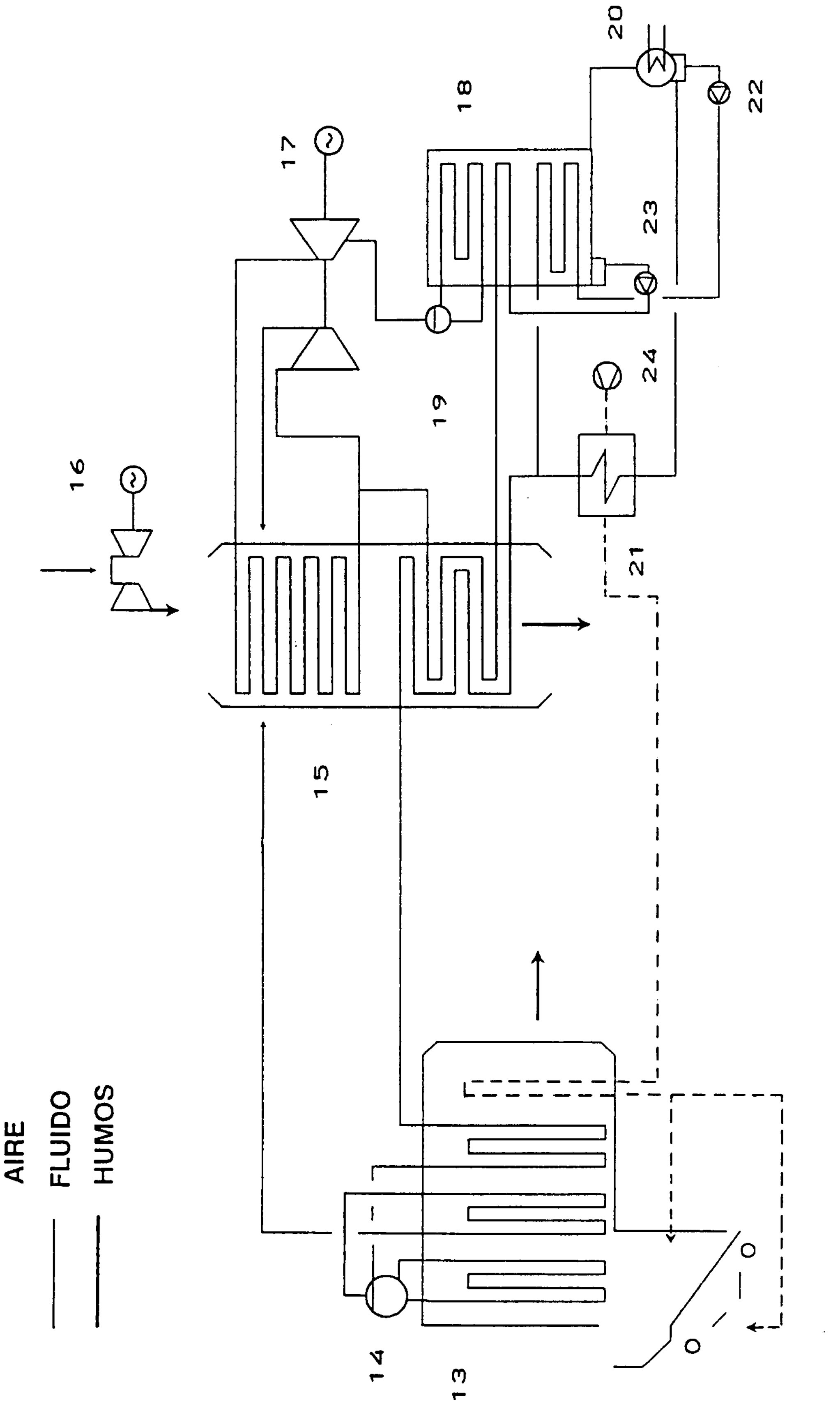

FIG. 2

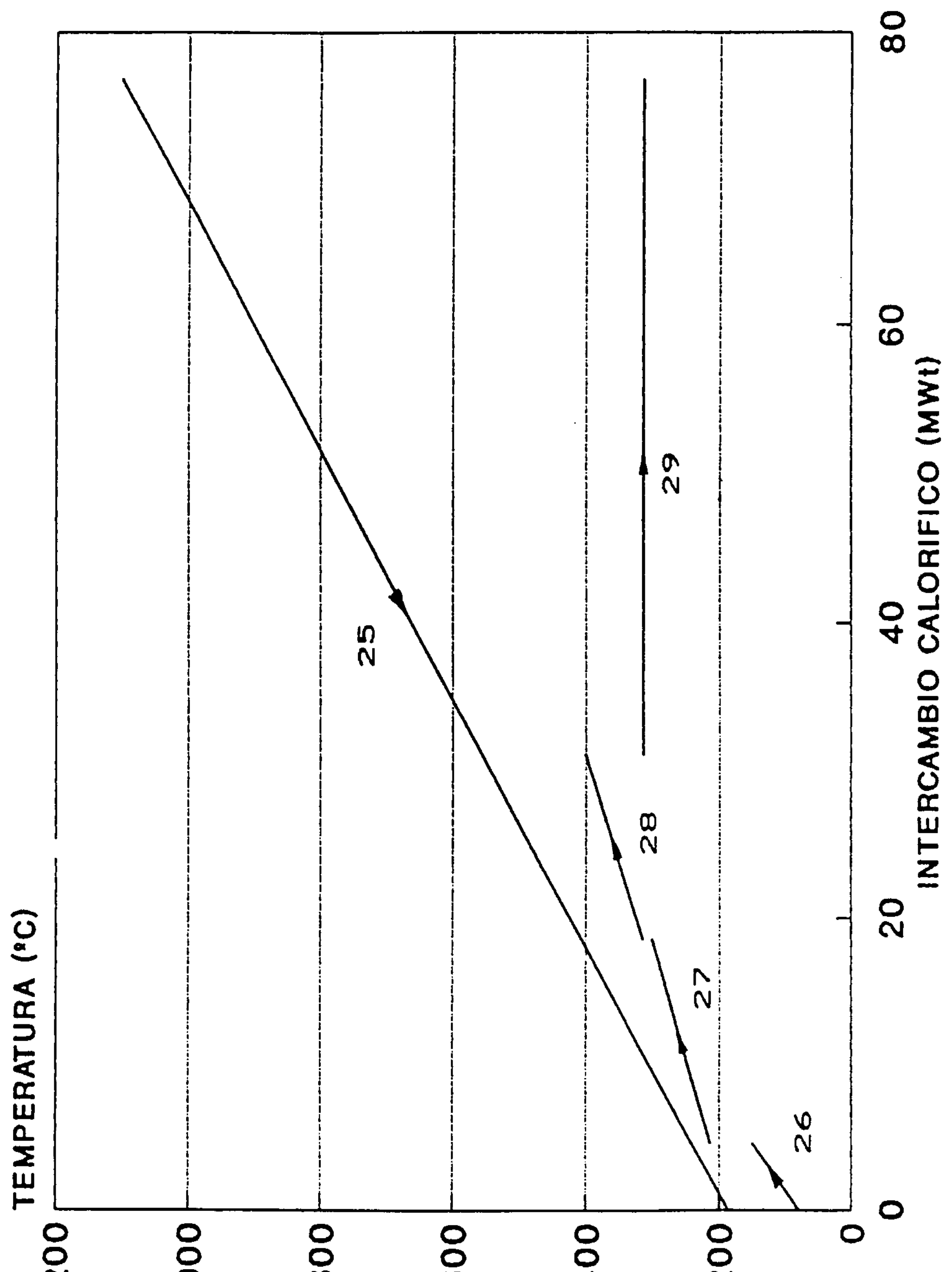
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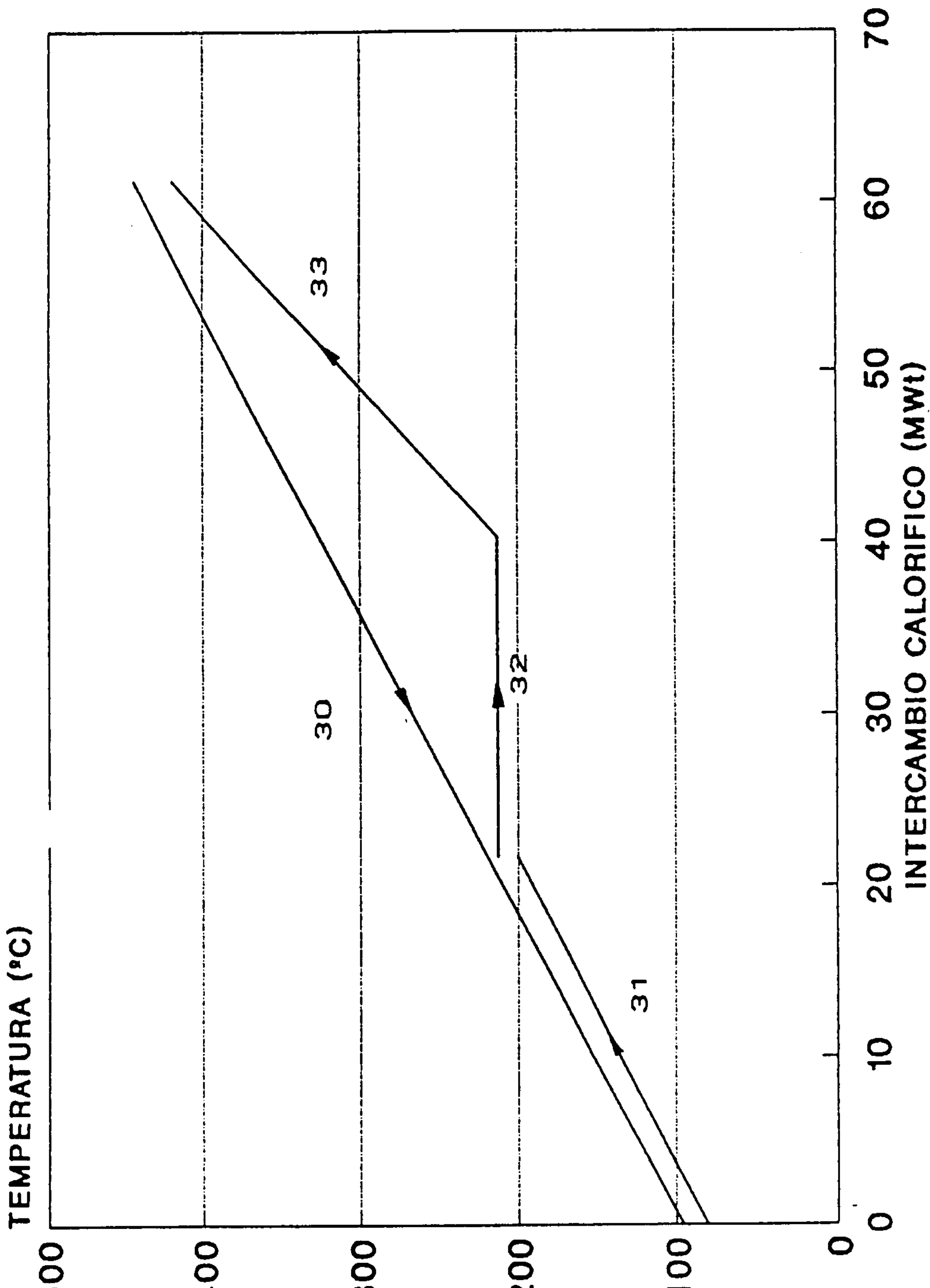
FIG. 3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INTERCAMBIO CALORIFICO (MWt)
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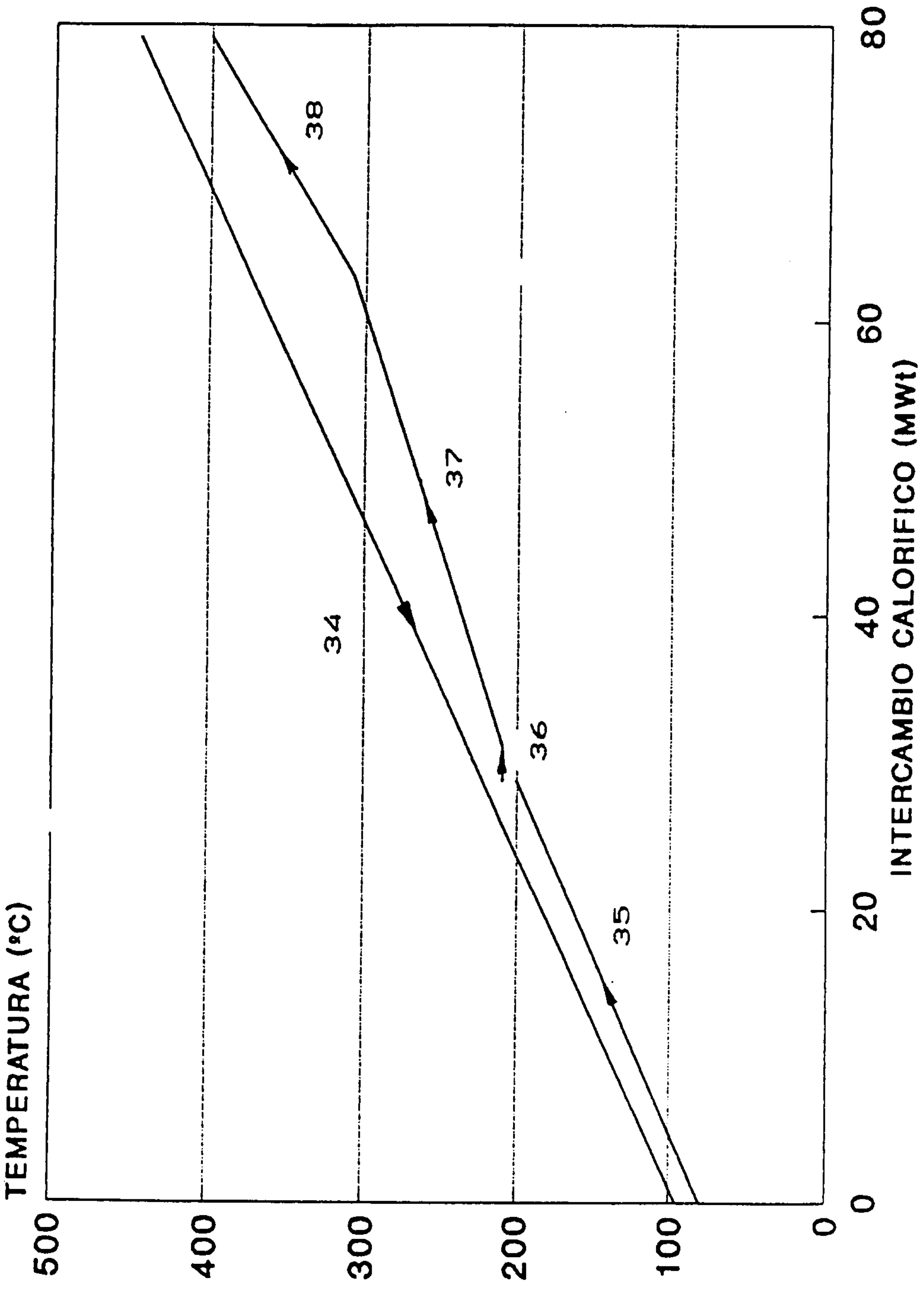
FIG. 4


TEMPERATURA (°C)
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INTERCAMBIO CALORIFICO (MWt)
34
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36
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FIG. 5

INTERNATIONAL SEARCH REPORT

International application No.
PCT/ES 94/00017

A. CLASSIFICATION OF SUBJECT MATTER
IPC 5 F01K23/10

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 5 F01K

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	EP,A,0 325 083 (SENER INGENIERIA Y SISTEMAS S.A.) 26 July 1989 see column 1, line 8 - line 14 see column 2, line 9 - column 3, line 7 ---	1-5
Y	EP,A,0 299 555 (PROMETHEUS ENERGY SYSTEMS B.V.) 18 January 1989 see column 1, line 15 - line 25 see column 1, line 46 - column 3, line 8 see column 3, line 38 - line 50 see column 4, line 20 - line 47 see column 5, line 12 - line 36 ---	1,2
Y	WO,A,92 21860 (SAARBERGWERKE AG) 10 December 1992 see abstract; figure --- -/--	3

[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

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"A" document defining the general state of the art which is not considered to be of particular relevance

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
25 May 1994	15.06.94

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+ 31-70) 340-2040, Tx. 31 651 epo nl, Fax (+ 31-70) 340-3016	Alconchel y Ungria,J

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	GEC ALSTHOM TECHNICAL REVIEW no. 4 , January 1991 , PARIS FR pages 15 - 26 XP223870 PAREN ET AL. 'Combined Cycle Plants Three-Pressure Reheat Vega 109F' see figure 5 ---	4,5
A	EP,A,0 286 565 (CARNOT S.A.) 12 October 1988 see the whole document ---	6-8
A	DE,A,41 01 064 (ENERGIE UND UMWELTTECHNIK GMBH) 23 July 1992 see figure 1 ---	1
A	DE,A,41 03 228 (ENERGIE UND UMWELTTECHNIK GMBH) 6 August 1992 see figures -----	1