Modified Steam-Turbine Rankine Cycle Without Rejection of the Cycle Condensation Heat, Driven by a Wet-Vapor-Region Thermocompressor (Ejector)

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Abstract - **The disclosed concept relates to a novel modified and simplified steam-turbine Rankine cycle without rejection of the cycle waste heat of condensation, which is driven by a thermocompressor (ejector) operating in the wet-vapor region. The thus modified steam-turbine Rankine cycle can theoretically achieve the maximum possible thermal efficiency (~100%). The wet-vapor mixture circulating within the thermocompressor is being separated in a dedicated cylindrical separation tank, so that the saturated water is pumped to a water heater where it receives the cycle heat input, while the saturated vapor is expanded in a backpressure steam turbine producing useful mechanical work and is then recirculated back to the thermocompressor, where it is being re-pressurized by means of the primary fluid (pumped and heated saturated water). The concept can be applied to steam-turbine-cycle power-plants fueled by: coal or solid/liquid/gaseous fuel, waste heat, nuclear fuel (used by boiling water reactors, pressurized water reactors, pressurized heavy-water reactors, gas-cooled reactors, molten salt reactors or liquid-metal-cooled fast reactors) or renewable energy sources (Solar energy, biomass, geothermal). The concept can also be applied as the "bottoming" steam-turbine-cycle part of a combined gas-turbine/steam-turbine cycle power plant.**

INTRODUCTION

It is a well-known fact that the simplest and the most straightforward way of decreasing concentration of the main green-house gases (GHGs) in the atmosphere (CO_2–gas and water vapor H_2O), representing the main ingredients of flue gases resulting from combustion of fossil fuels, and also extending the use of non-renewable (fossil & nuclear) fuels, is to **increase/ improve the (cycle thermal) energy efficiency** of thermal-to-mechanical (electrical) energy conversion for any kind of fossil or nuclear fuel used.

State-of-the-art energy conversion systems (one of which is disclosed and described in the prior-art document [4]) claim higher-than-conventional cycle thermal efficiencies and also larger-than-conventional cycle specific outputs. However, the claimed thermal efficiencies are still limited by and lower than the in-theory maximum possible thermal efficiency of a thermodynamic cycle operating between a higher temperature (T_H) and a lower temperature (T_C), that is, the **Carnot-cycle efficiency**, defined by the following simple expression:

$$\eta_{th} \leq \eta_{th,Carnot} = \frac{T_H - T_C}{T_H} = 1 - \frac{T_C}{T_H}$$

In general, **thermal efficiency** of a thermodynamic cycle is defined as the ratio of the difference between the heat added to the cycle (Q_{in}) and the heat rejected from the cycle (Q_{out}) and the heat added to the cycle (Q_{in}):

$$\eta_{th} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

Obviously, when $Q_{out} = 0$, then the cycle thermal efficiency $\eta_{th} = 1 = 100\%$. This invention shows how can this ideal **maximum-thermal-efficiency** thermodynamic cycle be **practically achieved**, thus enabling a much longer use of fossil and nuclear fuels and an efficient reduction of global-warming gases (greenhouse gases) in the Earth's atmosphere.

SUMMARY OF THE INVENTION AND PREFERRED CONFIGURATIONS

The disclosed invention (patent applications [1] and [2]) proposes and describes a novel **modified and simplified** Rankine steam-turbine cycle **without rejection of the cycle waste heat**, which is driven by a **thermocompressor** (ejector) operating in the **wet-vapor region**, to the end of achieving of the **maximum possible** (~100%) thermal efficiency of the thus modified Rankine cycle. The wet-vapor mixture contained in the modified-Rankine-cycle system and circulating within the thermocompressor is separated in a **cylindrical separation tank**, so that the saturated water is pumped to a **water heater** where it receives the cycle heat input, while the saturated vapor is expanded in a **backpressure steam turbine** producing useful mechanical work and is then recirculated back to the thermocompressor, where it is being **re-pressurized** by the primary ejector fluid (pumped and heated saturated water). Since the backpressure-steam-turbine's power output largely exceeds the saturated-water-pump's power input and there is **no cycle waste heat rejection**, the theoretical maximum thermal efficiency of the thus modified Rankine cycle is **close to 100%**.

The proposed modified Rankine-cycle power-plant **without rejection of the cycle waste heat** can be arrangeed in the 2 (two) following distinctive power-plant configurations:

(1) using a **non-superheated** (saturated) backpressure steam turbine, with the cycle heat input limited only to the water heater (**Fig. 1**); and

(2) using a **superheated** backpressure steam turbine, where the cycle heat input is applied also to the saturated steam separated in the separation tank, in addition to the water heater (**Fig. 3**).

FIG. 1

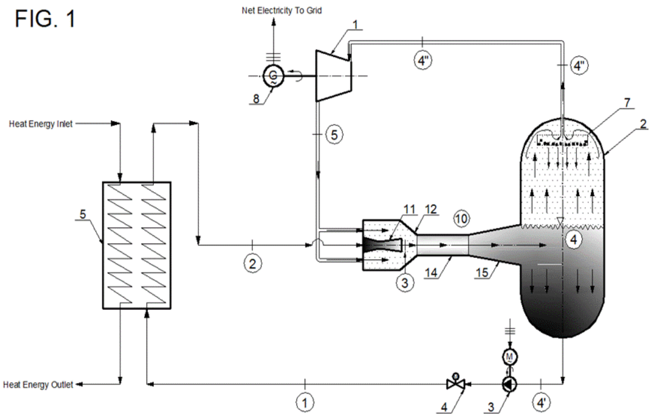

Figure 1. Flow diagram of **indirectly-heated** modified Rankine-cycle power-plant using a **non-superheated (saturated) backpressure steam turbine**

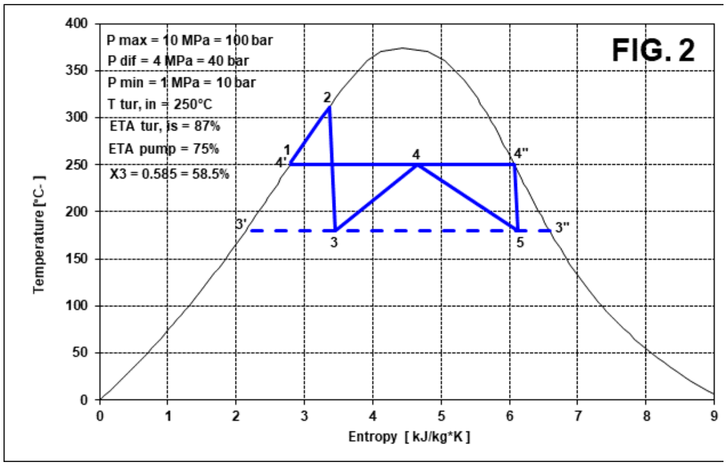

Figure 2. T-s diagram of **indirectly-heated** modified Rankine-cycle power-plant using a **non-superheated (saturated) backpressure steam turbine**

Fig. 2 depicts a **temperature/specific-entropy (T-s) diagram** corresponding to the modified Rankine-cycle power-plant without rejection of the cycle waste heat, whose **flow diagram** is depicted in the above **Fig. 1**, wherein the following symbols are used to designate the involved thermodynamic states and processes:

State 1 – pumped primary ejector fluid (saturated water) prior to heating in the liquid/water heater (**5**)

State 2 – heated primary ejector fluid (saturated water) prior to acceleration in the nozzle (**11**) of the wet-vapor thermocompressor (**10**)

State 3 – heated primary ejector fluid (saturated water) after acceleration in the nozzle (**11**) of the wet-vapor thermocompressor (**10**)

State 4 – wet-vapor mixture at the exit of the diffuser (**15**) of the wet-vapor thermocompressor (**10**)

State 4' – saturated liquid (water) at static pressure at the exit of the diffuser (**15**) of the wet-vapor thermocompressor (**10**), and at the suction of the condensate pump (**3**)

State 4" – saturated vapor (dry steam) at static pressure at the exit of the diffuser (**15**) of the wet-vapor thermocompressor (**10**), and at the inlet of the backpressure steam turbine (**1**)

State 5 – secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**)

Process 4'-1 – Adiabatic pumping of the primary ejector fluid (saturated water) from the diffuser outlet static pressure to the maximum cycle static pressure

Process 1-2 – Isobaric heating of the pumped primary ejector fluid (saturated water) to the maximum cycle temperature in the liquid/water heater (**5**)

Process 2-3 – Adiabatic acceleration of the heated primary ejector fluid (saturated water) to the minimum static pressure in the nozzle (**11**)

Process 4"-5 – Adiabatic expansion of the secondary ejector fluid (saturated vapor) in the backpressure steam turbine (**1**) to the minimum static pressure in the nozzle (**11**)

Processes 3-4 and 5-4 – Adiabatic compression of the primary ejector fluid (low-quality wet vapor) and the secondary ejector fluid (exhausted high-quality wet vapor), respectively, in the diffuser (**15**)

FIG. 3

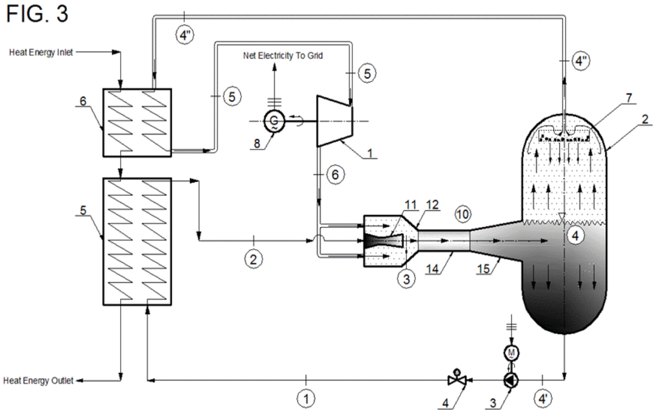

Figure 3. Flow diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant using a **superheated backpressure steam turbine**

Similarly, the above **Fig. 3** depicts a **flow diagram** of the **alternative** modified Rankine-cycle power-plant without rejection of the cycle waste heat. The main difference of **Fig. 3** relative to **Fig. 1** is use of an **additional heat exchanger/ superheater** (**6**) for isobaric heat addition to the saturated vapor (gas fraction) separated in the cylindrical separation tank (**2**), to the end **superheating** of the saturated vapor and thus enabling the backpressure steam turbine (**1**) to operate with **superheated steam** at its **inlet**, resulting in an **increased** steam-turbine **specific work** for the same expansion pressure ratio. Consequently, **Fig. 4** depicts a **temperature/specific-entropy (T-s) diagram** corresponding to the **alternative** modified Rankine-cycle power-plant depicted in **Fig. 3**. The following symbols are used to designate additionally involved/altered thermodynamic states and processes:

State 5 – heated primary ejector fluid (superheated vapor/steam) prior to adiabatic expansion in the backpressure steam turbine (**1**)

State 6 – secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**)

Process 4”-5 – Isobaric heating of the primary ejector fluid (saturated vapor) to a maximum chosen steam temperature in the additional heat exchanger/superheater (**6**)

Process 5-6 – Adiabatic expansion of the secondary ejector fluid (superheated vapor/steam) in the backpressure steam turbine (**1**) to the minimum static pressure in the nozzle (**11**)

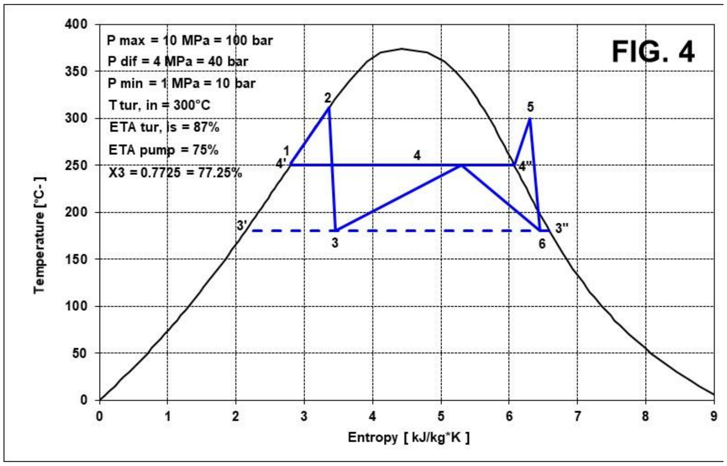

Figure 4. T-s diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant using a **superheated backpressure steam turbine**

It is also important to further emphasize that the **expansion pressure ratio** (EPR) of the said backpressure steam turbine (**1**) **must be proportional** to the **pressure recovery ratio** of the wet-vapor thermocompressor (**10**) obtainable at an **ejector entrainment ratio** (ratio of mass flow rates of suction and driving fluid) that is defined by the vapor quality at the ejector’s diffuser outlet. The EPR of the backpressure steam turbine (**1**), and thus the ejector entrainment ratio, can be chosen to be a **moderate one**, say from **2 : 1** to **4 : 1**, while a **lower-than-typical** maximum temperature of the cycle heat addition can be used. In relation to this, the invention highlights an option to add a **steam compressor** (**18**) to **any configuration** of the modified Rankine steam-turbine cycle without rejection of the cycle waste heat, which is used for **precompression** of the secondary ejector fluid (separated saturated steam/vapor) **prior to its expansion** in the said backpressure steam turbine (**1**), thus **artificially increasing** the thermocompressor pressure recovery ratio.

FIG. 18

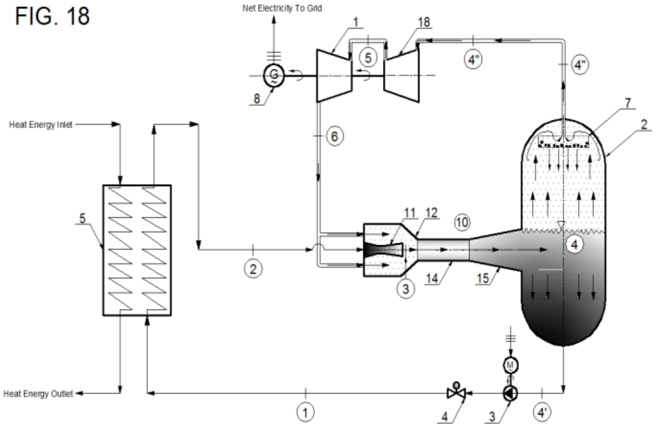

Figure 18. Flow diagram of **indirectly-heated** modified Rankine-cycle power-plant of **Fig. 1** using an **additional steam compressor** for **precompression** of the **separated steam/vapor prior to** its expansion in the **backpressure steam turbine**

FIG. 19

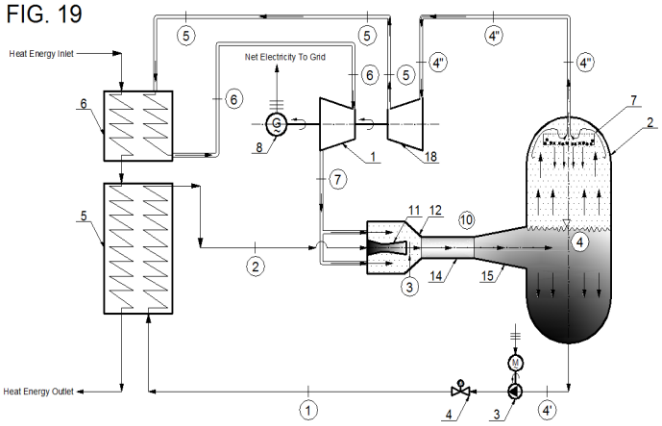

Figure 19. Flow diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant of **Fig. 3** using an **additional steam compressor** for **precompression** of the **separated steam/vapor prior to** its expansion in the **backpressure steam turbine**

Further, it is possible to perform **steam/water separation** in the **cylindrical separation tank** (**2**) of the proposed modified Rankine-cycle power-plant driven by a wet-vapor-region thermocompressor in several different ways, of which the 3 (three) following ways/means are briefly explained herewith:

(1) a **dry-pipe steam separator** (located typically within the steam drum of a steam boiler) (**Fig. 1** & **Fig. 3**), having of a lot of holes at the top and two holes at the bottom half, whereas the turbulently moving steam-water mixture is directed through the top half holes of the dry pipe and forced to separate between water and steam, whereby the separated steam will flow to the steam turbine and the separated water will drop through bottom holes;

(2) a **baffle-plate steam separator** (**Fig. 14** & **Fig. 15**), having the cylindrical separation tank fitted with typically two (2) to three (3) **baffle plates**, which serve to change the direction of the incoming steam flow when the steam strikes the baffle plates, prompting heavier water particles contained in the steam-water mixture to fall down to the bottom of the separation tank, while

the separated steam is freed from water particles and passed to the steam turbine; and

(3) a **centrifugal/cyclone steam separator** typically used in large-scale boilers, having the cylindrical separation tank fitted with at least one cyclone, which utilizes **centrifugal force** to separate water and steam from the steam-water mixture, whereby the steam-water mixture is forced to move around the cyclone and make the rotation; typically, the more turbulent flow forces the mixture to separate more easily.

FIG. 14

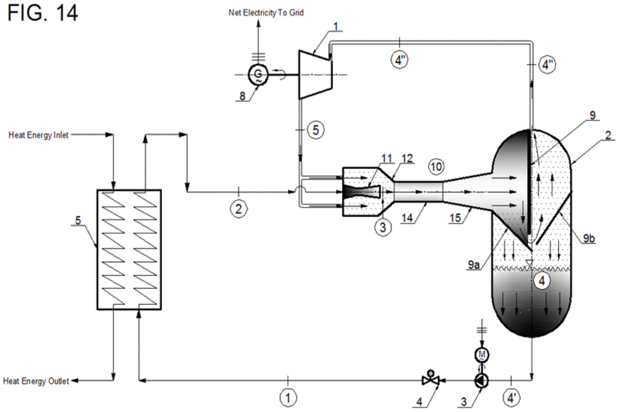

Figure 14. Flow diagram of **indirectly-heated** modified Rankine-cycle power-plant of **Fig. 1** using a **baffle-plate steam separator** (**2**)

FIG. 15

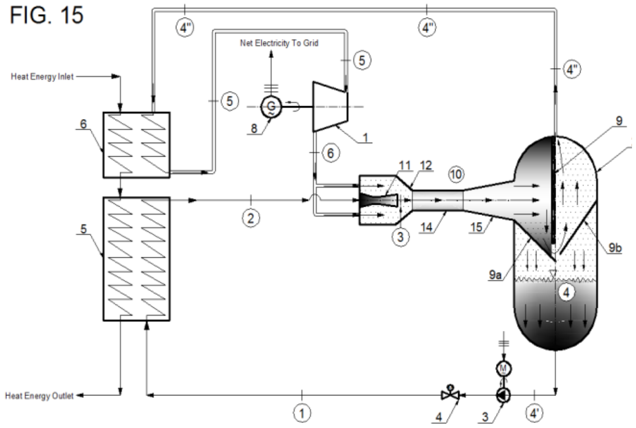

Figure 15. Flow diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant of **Fig. 3** using a **baffle-plate steam separator** (**2**)

Regulation of the cycle output/load of the proposed modified Rankine-cycle power-plant without rejection of the cycle waste heat in working regimes other than the nominal working regime can be performed using one of the two (2) following methods:

(1) **qualitative regulation**, that is, regulation of the cycle output/load by alteration of the **steam-turbine inlet temperature** via the cycle heat input, which, although a quite simple regulation method, can result in a probable existence of a **non-stationary normal shock wave** (**not necessarily** a **weak one**) somewhere in the mixing-chamber throat of a **supersonic wet-vapor** mixing thermocompressor/ejector for any change of the cycle load and hence change of the ejector working regime, especially in closed-loop configs of Rankine-cycle power-plants, which could potentially result in a **substantial** reduction of the recoverable pressure rise in the thermocompressor/ejector; or

FIG. 16

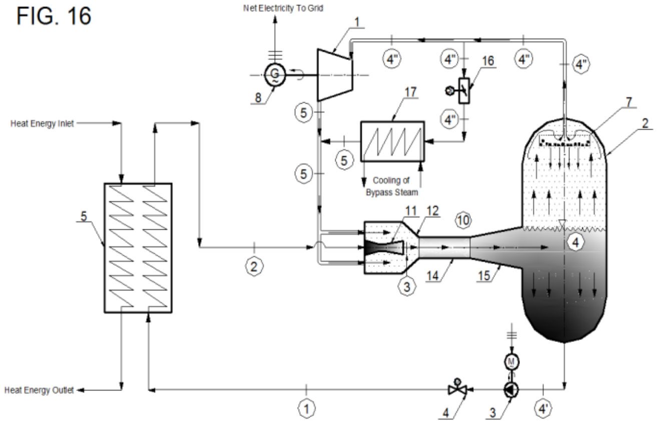

Figure 16. Flow diagram of **indirectly-heated** modified Rankine-cycle power-plant of **Fig. 1** using **quantitative regulation** of the cycle output via a **steam-turbine bypass** (**16**)

FIG. 17

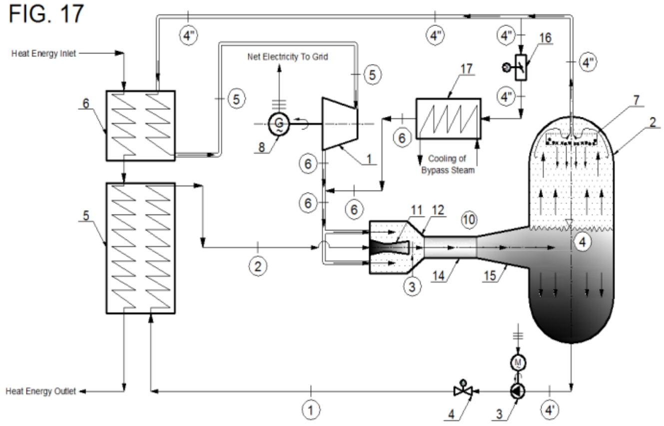

Figure 17. Flow diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant of **Fig. 3** using **quantitative regulation** of the cycle output via a **steam-turbine bypass** (**16**)

(2) **quantitative regulation**, that is, regulation of the cycle output/load by alteration of the **steam-turbine mass flow rate** using **bypassing** (**16**) of the steam turbine and a subsequent **external cooling** of the corresponding portion of the steam-turbine **bypass** mass flow rate **regime** (using an **external water or air cooler**, **17**) up to the steam-turbine outlet temperature existing in the **nominal cycle working** (**Fig. 16** & **Fig. 17**), which, coupled with an appropriate thermocompressor design, should ensure that eventually-occurring **normal shock wave** is preferably located in the mixing-chamber throat of the **supersonic wet-vapor** mixing ejector and that it is a **weak one**, occurring in the vicinity of the unity Mach number (1.0), and also a **stationary one** at a continually maintained **steady-state** ejector working regime, and hence a potential reduction of the recoverable pressure rise in the thermocompressor/ejector would likely be **minor**.

The proposed modified Rankine steam-turbine cycle driven by a wet-vapor-region thermocompressor and **without cycle waste heat rejection** can ideally be applied in the conventional steam-turbine-cycle thermal power-plants **fueled by externally-fired coal** (**Fig. 5**) and/or in **indirectly-heated** (via the heater **5**) steam-turbine-cycle power-plant configurations powered by **nuclear fuel** and using any of the commercially used **thermal-neutron** nuclear reactors: light-water moderated **boiling water reactor (BWR)** (**Fig. 7 & Fig. 11**) and **pressurized water reactor (PWR)** (**Fig. 8 & Fig. 9**), heavy-water moderated **pressurized heavy-water reactor (PHWR)**, graphite-moderated **molten salt reactor (MSR)** and graphite-moderated **gas-cooled reactor (GCR)**, as well as commercially used **fast-neutron** nuclear reactors, such as **liquid-metal-cooled fast reactor (LMFR)**.

FIG. 5

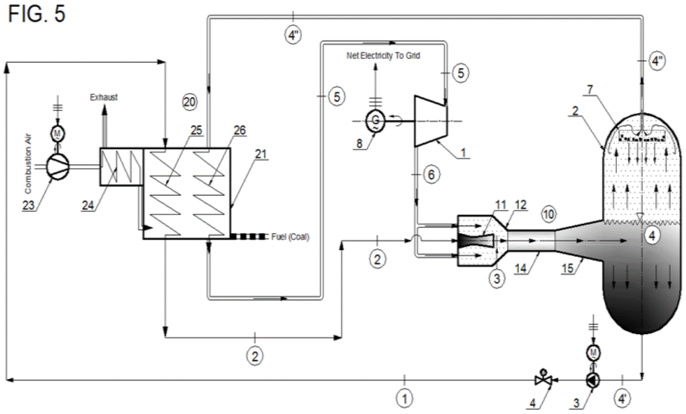

Figure 5. Flow diagram of **externally-coal-fired alternative indirectly-heated** modified Rankine-cycle power-plant using a **superheated backpressure steam turbine**

FIG. 7

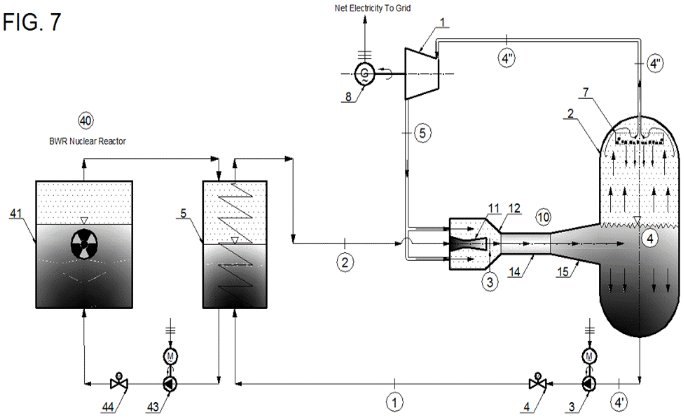

Figure 7. Flow diagram of **indirectly-heated** modified Rankine-cycle power-plant using a **non-superheated (saturated) backpressure steam turbine** powered by a **boiling-water nuclear reactor (BWR)** (**40**)

The proposed modified Rankine steam-turbine cycle **without cycle waste heat rejection** can also ideally be applied in either **directly-heated** or **indirectly-heated** steam-turbine-cycle power-plant configs **fueled/powered** by **renewable energy sources**, such as: **Solar energy**, **biomass** and **geothermal energy**.

FIG. 8

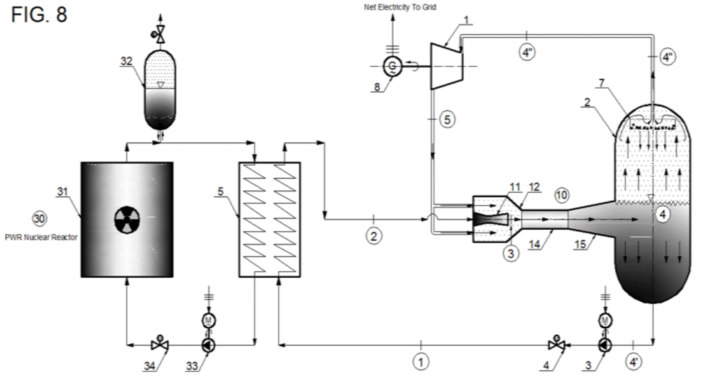

Figure 8. Flow diagram of **indirectly-heated** modified Rankine-cycle power-plant using a **non-superheated (saturated) backpressure steam turbine** powered by a **pressurized-water nuclear reactor (PWR)** (**30**)

FIG. 9

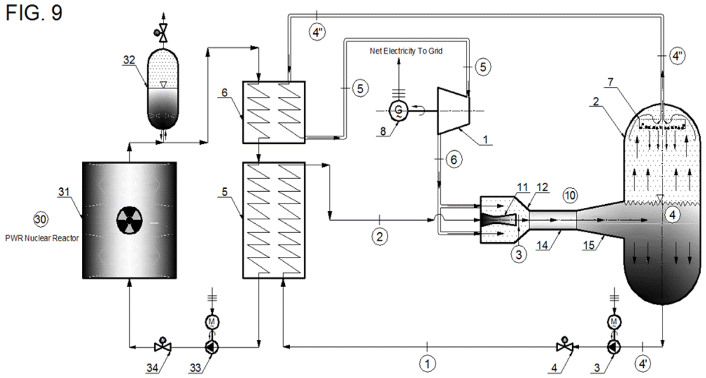

Figure 9. Flow diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant using a **superheated backpressure steam turbine** powered by a **pressurized-water nuclear reactor (PWR)** (**30**)

Alternatively, **PWR**-based (and even **PHWR**-based) nuclear power plants may opt and attempt to use the **directly-heated** liquid reactor coolant as a primary fluid of the thermo-compressor, thus omitting use of the intermediary water heater **5**.

FIG. 11

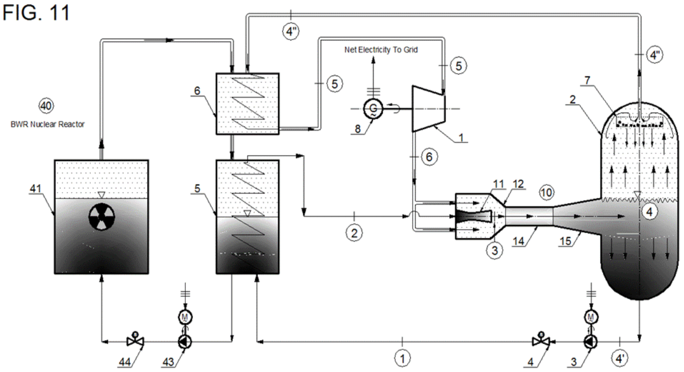

Figure 11. Flow diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant using a **superheated backpressure steam turbine** powered by a **boiling-water nuclear reactor (BWR)** (**40**)

In addition, the proposed modified Rankine steam-turbine cycle driven by a wet-vapor-region thermocompressor and **without cycle waste heat rejection** can also be very suitably applied as an **indirectly-heated bottoming** steam-turbine-cycle power-plant of a **natural-gas-fired combined gas-turbine/steam-turbine cycle (NGCC)**.

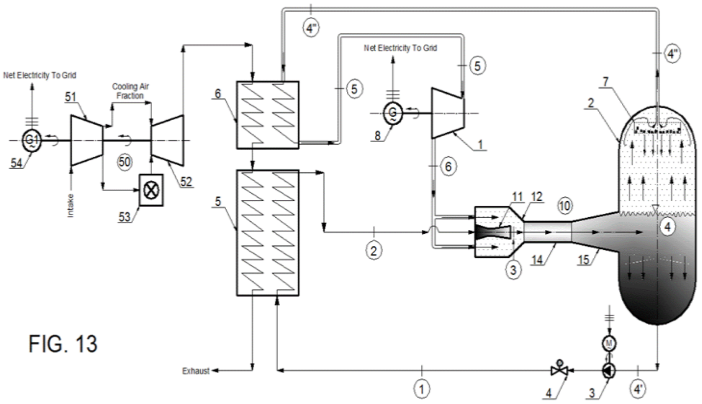

FIG. 13

Figure 13. Flow diagram of **alternative indirectly-heated** modified Rankine-cycle power-plant using a **superheated backpressure steam turbine** powered by waste heat of a **natural-gas-fired combined gas-turbine/steam-turbine cycle (NGCC)** (**50**)

In addition, the **boiler** or the **integral water/steam heater** (**20**) in the **externally-fired** (by coal, solid/liquid waste fuel or biomass) configuration of the proposed modified Rankine steam-turbine cycle power-plant (depicted in **Fig. 5**) is **simpler** (and therefore likely less expensive) than the boiler of a conventional steam-turbine power-plant, since it does not include either an **evaporator** nor a **steam reheater**. It incorporates the liquid/water heater **5** and the superheater **6** in the form of **multi-tube bundles 25** and **26**, respectively, and in addition also contains a **furnace refractory 21**, a **forced-draft fan 23** for combustion-air circulation, and a **regenerative combustion-air preheater 24**.

Finally, the fact that the proposed modified Rankine steam-turbine cycle driven by a wet-vapor-region thermocompressor is **without cycle waste heat rejection** should be emphasized, because it also means that it either does not include the **condensation process**. Consequently, the proposed modified Rankine steam-turbine cycle configurations contain **neither the condenser system nor the feedwater regenerative heater system**, which **considerably reduces the capital cost** of the proposed modified Rankine-cycle power-plant.

The proposed energy-conversion concept was derived based on the Serbian patent [4] by the same author (Branko Stanković). It bears some configurational similarity with the advanced vapor-ejector-based refrigeration system disclosed in the US patent publication [3], however, not the functional similarity, because the herewith proposed concept is from the power-generation/energy-conversion field. Besides, the prior-art source [3] uses vapor/steam as the primary ejector fluid, while the herewith proposed energy concept uses heated water as the primary ejector fluid, while the vapor/steam is the secondary ejector/thermocompressor fluid.

APPLIED MATHEMATICAL MODEL

Applied mathematical model uses the following simple system of basic fluid-mechanic & thermodynamic equations: **conservation of energy equation**, expressions for **saturated water and saturated vapor** (quality) **mass fractions** in an equilibrium wet-vapor mixture, **wet-vapor enthalpy** expression, and expression for **primary ejector fluid velocity at nozzle outlet**, as follows:

$$\dot{m}_{prim} \cdot \left(h_3 + \frac{v_3^2}{2} \right) + \dot{m}_{sec} \cdot \left(h_5 + \frac{v_5^2}{2} \right) = \left(\dot{m}_{prim} + \dot{m}_{sec} \right) \cdot \left(h_4 + \frac{v_4^2}{2} \right)$$

$$\dot{m}_{prim} = \dot{m}_{H2O} \cdot (1 - x_4) \;\; \& \;\; \dot{m}_{sec} = \dot{m}_{H2O} \cdot x_4 \rightarrow \left(\dot{m}_{prim} + \dot{m}_{sec} \right) = \dot{m}_{H2O}$$

$$h_4 = h_4' + x_4 \cdot (h_4'' - h_4') \qquad \& \qquad v_3 = \sqrt{2 \cdot (h_2 - h_3)}$$

where: **m_{prim}** and **m_{sec}** [kg/s] are mass flow rates of the **primary ejector fluid**, or jet/motion fluid (pumped and heated saturated liquid/water in this case) and the **secondary ejector fluid**, or suction (injected) fluid (exhausted wet vapor in this case), **m_{H2O}** [kg/s] is total mass flow rate of the wet-vapor mixture, that is, the sum of mass flow rates of the primary and the secondary ejector fluid, **h_2** [kJ/kg] and **h_3** [kJ/kg] are enthalpies of the jet/motion fluid prior to and after acceleration in the nozzle (**11**) of the wet-vapor thermocompressor (**10**), respectively, **h_4** [kJ/kg] is enthalpy of the wet-vapor mixture at the exit of the diffuser (**15**) of the wet-vapor thermocompressor (**10**), **h_4'** [kJ/kg] and **h_4"** [kJ/kg] are saturated liquid (water) and saturated vapor (dry steam) enthalpies, respectively, at static pressure at the exit of the diffuser (**15**) of the wet-vapor thermocompressor (**10**), **x_4** [**-**] is quality of the wet-vapor mixture at the exit of the diffuser (**15**) of the wet-vapor thermocompressor (**10**), **v_3** [m/s] is velocity of the jet/motion fluid after acceleration in the nozzle (**11**) of the wet-vapor thermocompressor (**10**), **v_4** [m/s] is velocity of the wet-vapor mixture at the exit of the diffuser (**15**) of the wet-vapor thermocompressor (**10**), and **v_5** [m/s] is velocity of the suction (injected) fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**).

Combining and rearranging the above equations, the **vapor quality** at the exit of the wet-vapor-region thermocompressor diffuser, **x_4**, can be expressed as follows:

$$\rightarrow \quad (1 - x_4) \cdot h_2 + x_4 \cdot \left(h_5 + \frac{v_5^2}{2} \right) = \left[h_4' + x_4 \cdot (h_4'' - h_4') + \frac{v_4^2}{2} \right] \quad \rightarrow$$

$$\rightarrow \quad h_2 - h_4' - \frac{v_4^2}{2} = x_4 \cdot \left(h_4'' - h_4' + h_2 - h_5 - \frac{v_5^2}{2} \right) \quad \rightarrow$$

$$\boxed{\rightarrow \quad x_4 = \frac{\left(h_2 - h_4' - \frac{v_4^2}{2}\right)}{\left(h_4'' - h_4' + h_2 - h_5 - \frac{v_5^2}{2}\right)}}$$

Thermal efficiency $\eta_{i,turb}$ of the modified Rankine cycle without cycle-heat rejection and driven by a wet-vapor-region thermocompressor is then defined according to the following expression:

$$\boxed{\eta_{cycle} = \frac{x_4 \cdot (h_4'' - h_5) - (1 - x_4) \cdot (h_1 - h_4')}{(1 - x_4) \cdot (h_2 - h_1)}}$$

where: h_1 [kJ/kg] is enthalpy of the pumped primary ejector fluid (saturated water) prior to heating in the liquid/water heater (**5**).

The above explained mathematical model has been based on the general assumption of **uniformity of static pressure** across the mixing-tube/chamber inlet: (**p_3 / p_5 = 1.0**), where p_5 [kPa] is static pressure of the secondary ejector fluid (exhausted wet vapor) after adiabatic expansion in the backpressure steam turbine (**1**), while p_3 [kPa] is static pressure of the primary ejector fluid (saturated water) after adiabatic acceleration in the nozzle (**11**).

An **exemplary case #1** has been chosen, which applies to the configuration of indirectly-heated **modified Rankine-cycle power-plant** using a **non-superheated (saturated)** backpressure steam turbine (depicted in **Fig. 1**). The following general assumptions have been adopted: the backpressure-steam-turbine isentropic efficiency of $\eta_{i,turb}$ = **87%**, overall efficiency of the condensate pump of η_{pump} = **75%**, the maximum cycle static pressure of $p_1 = p_2$ = **10 MPa** (100 bar or 1,450 psi), the minimum static pressure at the outlet of the nozzle (**11**) of $p_3 = p_5$ = **1 MPa** (10 bar or 145 psi), the designed static pressure at the outlet of the diffuser (**15**) of p_4 = **4 MPa** (40 bar or 580 psi), the velocity of the wet-vapor mixture at the exit of the diffuser (**15**) of v_4 = 200 m/s, and the velocity of the secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**) of v_5 = 100 m/s. Neglecting pressure drop in the liquid/water heater (**5**), the calculation shows that the cycle thermal efficiency of $\eta_{cycle,Ia}$ = **~87.47%** of the proposed modified Rankine-cycle power-plant configuration depicted at **Fig. 1** is achievable, at the **vapor quality** at the exit of the wet-vapor-region thermocompressor diffuser of $x_{4,Ia}$ = **0.557**. However, the cycle thermal efficiency of the proposed modified Rankine-cycle power-plant configuration depicted at **Fig. 1** can be even higher, almost **close to 100%**, when the velocity of the wet-vapor mixture at the exit of the diffuser (**15**) becomes **equal** to the velocity of the secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**), that is, when $v_4 = v_5$ = 100 m/s: $\eta_{cycle,Ib}$ = **~98.39%** is achievable, at the **vapor quality** at the exit of the wet-vapor-region thermocompressor diffuser of $x_{4,Ib}$ = **~0.5848**.

Similarly, an **exemplary case #2** has been chosen, which applies to the configuration of indirectly-heated **modified Rankine-cycle power-plant** using a **superheated** backpressure steam turbine (depicted in **Fig. 3**). The following **additional/altered** general assumptions have been adopted: the maximum chosen steam temperature in the additional heat exchanger/superheater (**6**) of T_5 = **300°C** (573 K or 572°F) at the designed static pressure at the outlet of the diffuser (**15**) of p_4 = **4 MPa** (40 bar or 580 psi), velocity of the wet-vapor mixture at the exit of the diffuser (**15**) of v_4 = 200 m/s, and velocity of the secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**) of v_6 = 100 m/s. Neglecting pressure drops in both the liquid/ water heater (**5**) and the additional heat exchanger/superheater (**6**), the calculation shows that the cycle thermal efficiency of $\eta_{cycle,IIa}$ = **91.64%** of the proposed modified Rankine-cycle power-plant configuration depicted at **Fig. 3** is achievable, at the **vapor quality** at the exit of the wet-vapor-region thermocompressor diffuser of $x_{4,IIa}$ = ~**0.736**. However, the cycle thermal efficiency of the proposed modified Rankine-cycle power-plant configuration depicted at **Fig. 3** can be even higher, ideally **close to 100%**, when the velocity of the wet-vapor mixture at the exit of the diffuser (**15**) becomes **equal** to the velocity of the secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**), that is, when $v_4 = v_5$ = 100 m/s: $\eta_{cycle,IIb}$ = **99.4%** is achievable, at the **vapor quality** at the exit of the wet-vapor-region thermocompressor diffuser of $x_{4,IIb}$ = ~**0.773**.

In the above calculations of **exemplary cases #1** and **#2** it has been assumed that the pressure recovery ratio of the said thermocompressor (**10**) is **4 : 1**, which may seem an overestimation. However, similar calculation results and cycle efficiencies would have been obtained even for a much lower assumed thermocompressor pressure recovery ratio of **2 : 1**. For such or even lower thermocompressor pressure recovery ratios, it is recommendable and feasible to use the said **optional steam compressor** (**18**), which artificially increases the pressure recovery ratio of the said thermocompressor (**10**), thus allowing the said backpressure steam turbine (**1**) driving the said steam compressor (**18**) to still achieve a positive net surplus work.

CONCLUSION

A novel **modified and simplified** Rankine steam-turbine cycle **without rejection of the cycle waste heat** has been proposed, which is driven by a **thermocompressor** (ejector) operating in the **wet-vapor region**, to the end of achieving of the **maximum possible** (~100%) thermal efficiency of the thus modified Rankine cycle. The wet-vapor mixture contained in the modified-Rankine-cycle system and circulating within the thermocompressor is separated in a **cylindrical separation tank**, so that the saturated water is pumped to a **water heater** where it receives the cycle heat input, while the saturated vapor is expanded in a **backpressure steam turbine** producing useful mechanical work and is then recirculated back to the thermocompressor, where it is being **re-pressurized** by the primary ejector fluid (pumped and heated saturated water). Since the backpressure-steam-turbine's power output largely exceeds the saturated-water-pump's power input and there is **no cycle heat rejection**, the theoretical maximum thermal efficiency of the thus modified Rankine cycle is **close to 100%**.

The result of the above calculation for **exemplary case**

#1 shows that **nearly 100%-cycle-thermal-efficiency** can be obtained using the basic configuration (**Fig. 1**) of the proposed modified Rankine cycle without rejection of the cycle waste heat & driven by the wet-vapor-region thermocompressor, **provided** the velocity of the wet-vapor mixture at the exit of the diffuser (**15**) is **equal** to the velocity of the secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**), that is, when $\mathbf{v}_4 = \mathbf{v}_5$.

Similarly to conclusion for **exemplary case #1**, the above calculation result for the **exemplary case #2** shows that **almost exactly 100%-cycle-thermal-efficiency** can be obtained even more easily using the altered configuration (**Fig. 3**) of the proposed modified Rankine cycle cycle without rejection of the cycle waste heat & driven by the wet-vapor-region thermocompressor, **provided** the velocity of the wet-vapor mixture at the exit of the diffuser (**15**) is **equal** to the velocity of the secondary ejector fluid (exhausted wet vapor) at the exit of the backpressure steam turbine (**1**), that is, when $\mathbf{v}_4 = \mathbf{v}_5$.

The proposed concept of the modified Rankine steam-turbine cycle can ideally be applied in steam-turbine-cycle power-plant configurations **externally-fired** by **any kind of fuel** (fossil or nuclear), any type of **waste heat** or a suitable type of **renewable energy sources** (geothermal, Solar or biomass), using either **direct heating** or **indirect heating** of the working gas. Finally, the proposed modified-Rankine steam-turbine cycle is **without cycle waste heat rejection**, which also means that it does not include the **condensation process**, and, consequently the proposed power-plant contains **neither the condenser system nor the regenerative feedwater heater system**, which **considerably reduces capital cost** of the proposed modified Rankine-cycle power-plant.

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