Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45259

# INTERCOOLED-RECUPERATED GAS-TURBINE-CYCLE ENGINE COUPLED WITH PNEUMATIC MOTOR WITH QUASI-ISOTHERMAL HEAT ADDITION

**Branko Stankovic**
**Ex-Mc Master University**
**110-2091 Hurontario Street**
**Mississauga, ON, Canada**
**+1-647-618-2114**
**stankovb@yahoo.com**

## ABSTRACT

**A hybrid energy system has been proposed, consisting of an intercooled-recuperated gas-turbine-cycle engine coupled with a pneumatic motor (compressed-air engine), using reciprocating linear motion of a piston or a rotary vane motor, with quasi-isothermal heat addition process. Both gas-turbine (GT) engine and pneumatic motor have their own and separate combustion processes, occurring serially one after another, owing to the fact that in the GT engine exhaust gas there is still enough oxygen needed for combustion of fuel (liquid or gaseous) in an internal combustion engine. The quasi-isothermal nature of the heat addition process within the pneumatic motor is a result of averaging of the two thermodynamic processes, simultaneously interconnected within a cylinder-piston motor: isobaric heat addition and adiabatic gas expansion. The final expansion pressure at the end of the pneumatic-motor quasi-isothermal heat addition/expansion process is considered to correspond to the initial specific volume of inlet ambient air.**

**Three (3) possible configurations of such a hybrid energy system were analyzed, differing only in the sequence of equipment connecting in the direction of air/ working-gas flow (GT-cycle combustor, GT, pneumatic motor with combustor and recuperator). The results showed that the most efficient cycle configuration is No. 2, in which quasi-isothermal heat addition / gas expansion in the pneumatic motor occurs right after the GT-cycle heat addition in the associated GT-cycle combustor, then the partly expanded combustion gas first cools down in the GT-cycle recuperator, prior to its final expansion in the GT and exhaust to atmosphere. Estimated overall cycle thermal efficiency for the system configuration #2 ranges from ~62% for a maximum GT/pneumatic motor inlet temperature of 1500 K (1227°C or 2240°F) to ~66% for a maximum GT/pneumatic motor inlet temperature of 1700 K (1427°C or 2600°F), assuming a purely isothermal heat addition/expansion process in the pneumatic-motor cylinder. This is likely due to the fact that there is no necessity to cool the low-temperature GT of this configuration. Overall cycle thermal efficiency increases with the ambient temperature decrease for any cycle configuration.**

## INTRODUCTION

It is well known that increasing thermal efficiency (efficiency of thermal energy conversion into mechanical/ electrical energy) of thermodynamic cycle used for a power-plant reduces environmental pollution and climate changes phenomena ("global warming effect") caused by GHG-s, such as carbon-dioxide ($CO_2$) and water vapor ($H_2O$), as direct products of fuel combustion in fossil-fuels power-plants.

It is also very well known fact that natural-gas fired combined gas/steam turbine power plants (GTCC-s) are among those with highest cycle thermal efficiency. The best cycle thermal efficiency of GTCC-s is currently about 60% (based on lower heating value (LHV) of fuel, ISO standard conditions), for those GT-s using closed-loop steam cooling (internal convection cooling) of rotors and vanes/blades. Significance of the type of GT rotors/vanes/blades cooling for cycle thermal efficiency has been described and explained in detail in references [5], and [6]. The open-loop cooling techniques incorporate: internal convection, film cooling and transpiration cooling. Ref [6] states that: open-loop transpiration steam cooling, open-loop steam internal convection cooling, transpiration air cooling, film steam cooling, film air cooling and internal convection air cooling have been found to yield lower values of plant efficiency, in decreasing order, as compared to closed loop steam cooling.

Ref. [7] (as well as many other sources) recognizes that GT cycle thermal efficiency could benefit from introduction of a recuperative heat exchanger (recuperator or regenerator), compressor intercooler or their combination (intercooled-recuperated GT cycle).

On the other hand, it is well known that the conventional internal combustion engine (ICE), whether of the compression ignition or Diesel type, or of the spark-ignition or Otto type, has a disadvantage of being of low thermal efficiency. A principal source of the ICE-s inefficiency is its low volumetric efficiency, related to high energy losses contained in large volumes of the ICE-s exhaust gases, resulting in wasteful dissipation of the energy (pressure, temperature) contained in them. Plenty of technical solutions have been proposed and/or employed using the ICE-s exhaust gases as an energy device, with only modest (or not at all) improvements in the ICE-s thermal efficiency.

For example, the ICE-s exhaust gases have been used to drive a turbocharger, a form of supercharger, a gas compressor that is used for forced air induction of an ICE, which increases the density of air entering the ICE to create more power, thus improving the ICE-s volumetric efficiency. A turbocharger has the compressor powered by a gas-turbine, which is driven by the ICE-s exhaust gases rather than direct mechanical drive (as with supercharger). The turbocharger (or supercharger in general) is typically followed by a compressed-air intercooler which contributes to the improvement of the ICE-s volumetric efficiency by increasing the intake air density through an isobaric cooling process. By lowering of the intake air temperature the intercooler also eliminates the danger of pre-detonation (knock) of the fuel-air mixture prior to timed spark ignition in a spark-ignition ICE. However, while turbocharger (with intercooler) has increased the power output of the ICE, no significant increase (or not at all) of the ICE-s thermal efficiency has been achieved.

Thermal efficiency improvement of a turbocharged ICE has been considered by many researchers and has been proposed by ref. [1] and [2], among many others. This topic is also the main subject of this paper.

Ref. [2] disclosed/proposed a compound positive-displacement internal-combustion engine having a gas turbine driven by the ICE-s exhaust gases, a recuperator for transferring the waste heat contained in the fully-expanded exhaust gases to the fresh air incoming into the ICE, and a two-stage intercooled compressor for compressing and evacuating of the cooled-down exhaust gases to the atmosphere, driven by the said gas turbine. Thus, the proposed energy system does not present a turbocharged ICE, but a compound (a hybrid energy system) between a recuperated ICE and an intercooled GT engine. Ref. [2] does not explicitly quantify the improvement of the ICE-s thermal efficiency, but states that: "the improved internal combustion compound engine is simple, rugged and highly efficient, has a wide range of applicability and is of great versatility and adaptability".

Ref. [1] disclosed/proposed an "Engine Turbo Compound System", that is a gas engine (ICE) – GT combined cycle, comprising: a GT engine consisting of a compressor and a GT, with a gas engine (ICE) installed between them, coupled with a bottoming steam-cycle. Thus, essentially the concept represents a turbocharged ICE with supplementary-fired GT, optionally integrated with a bottoming steam-cycle. The source claims high efficiency of the compound energy system resulting from use of the ICE, which represents a "super-high pressure and super-high temperature GT with no cooling air". Thus, for the ICE-s maximum pressure of 160 bar (16 MPa or 2320 psi) and the maximum temperature of 1700°C (1973 K or 3092°F), and for the GT inlet temperature of 1200°C (1473 K or 2192°F) and inlet pressure of 14 bar (1.4 MPa or 203 psi), the claimed cycle thermal efficiency is 45% in the GT-ICE simple-cycle mode of operation and 59% in the GT-ICE combined-cycle mode of operation, based on the lower heating value (LHV) of fuel.

Ref. [3] disclosed an improved fluid power drive system, developing pressurized fluid form a partly-expanded GT-engine (compressor + GT) exhaust gas, supplied for use in operating pneumatic motors or similar fluid responsive devices. In the case that some or all of the pneumatic motors are non-operational for whatever reason, a normally idle GT has been provided, which is able to run on the excess pressurized fluid not required by some or all of the pneumatic motors, released by a variable pressure relief mechanism.

The motivation for this work is continued research on possible thermal efficiency improvement of hybrid intercooled GT-ICE based energy systems, especially when a pneumatic motor (compressed-air driven engine) is used instead of the ICE (engine with inherently included compression process). Thus, the work is primarily focused on hybrid energy systems combining an intercooled GT and a pneumatic motor with a separate heat addition, independent from the GT combustion process.

## DESCRIPTION OF G.T. ENGINE-PNEUMATIC MOTOR HYBRID ENERGY SYSTEM

This section of the paper outlines a brief description of the proposed hybrid energy system, consisting of an intercooled-recuperated gas-turbine-cycle engine coupled with a pneumatic motor, using reciprocating linear (axial or radial) motion of a piston or a rotary vane motor or a turbine motor, with quasi-isothermal heat addition process. Both GT engine and pneumatic motor have their own and separate combustion processes, occurring serially one after another, owing to the fact that in the GT engine exhaust gas there is still enough oxygen needed for combustion of fuel (liquid or gaseous) in a pneumatic motor or an ICE. The quasi-isothermal nature of the heat addition process within the pneumatic motor is a result of averaging of the two thermodynamic processes, simultaneously interconnected within a cylinder-piston motor: the isobaric heat addition and the adiabatic gas expansion. The final expansion pressure at the end of the pneumatic-motor quasi-isothermal heat addition/expansion process is considered to correspond to the initial specific volume of inlet ambient air. Since in the pneumatic motor there really is not any internal compression, there is also no danger of pre-detonation of the fuel-air mixture, as typically is the case with a spark-ignition ICE with inherent internal compression.

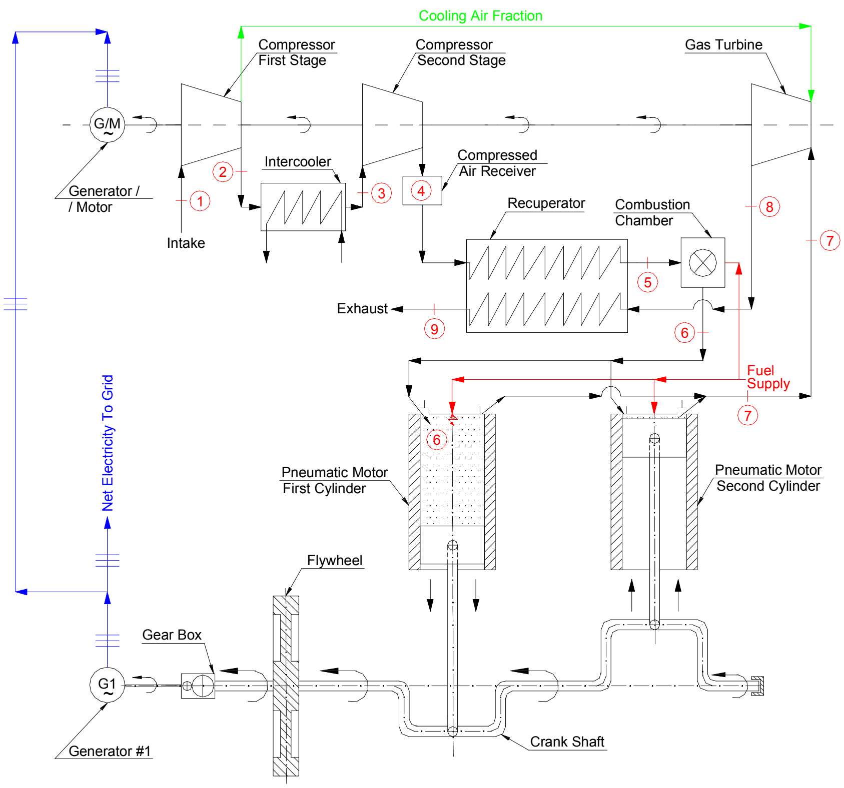


**FIG. 1** – Flow Diagram of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition – **Configuration #1**

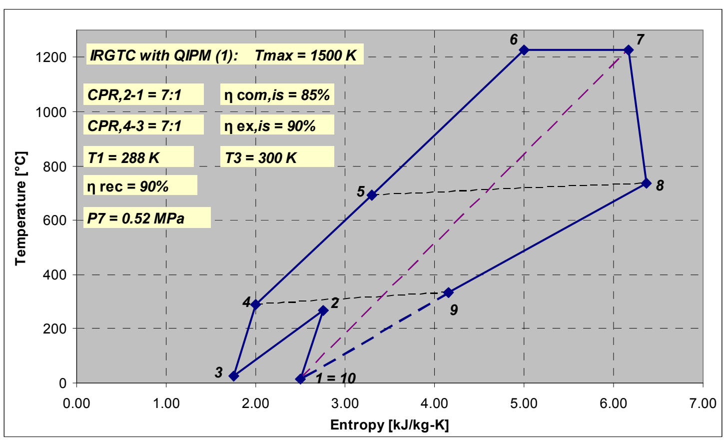


**FIG. 2** – Typical Thermodynamic Temperature-Entropy Diagram of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition – **Configuration #1**

The associated flow diagrams are depicted in figure FIG. 1 thru FIG. 3. Fuel (natural gas or fuel oil) supply lines are depicted by red lines, GT cooling-air flow is depicted by green lines, while electricity lines are colored blue.

**Figure 1** depicts a flow diagram of the **Configuration #1** of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition, consisting of the following interconnected equipment and processes depicted in the associated thermodynamic temperature-entropy diagram in **Figure 2**:

(a) a first-stage of air compressor for first-stage adiabatic compression of ambient air (adiabatic process **1-2**);

(b) a compressor intercooler for cooling down to near ambient temperature of the air partly compressed in the first-stage compressor (isobaric process **2-3**);

(c) a second-stage air compressor for second-stage adiabatic compression of the intercooled partly compressed air (adiabatic process **3-4**);

(d) a compressed-air receiver for equalizing eventual air flow surges due to coupling a GT engine and a piston engine (pneumatic motor);

(e) a recuperator/heat exchanger for internal heat-exchange between the compressed air and the fully expanded combustion gas (isobaric processes **4-5** and **8-9**);

(f) a combustion chamber for isobaric heat addition to the compressed air to a maximum cycle temperature (isobaric process **5-6**);

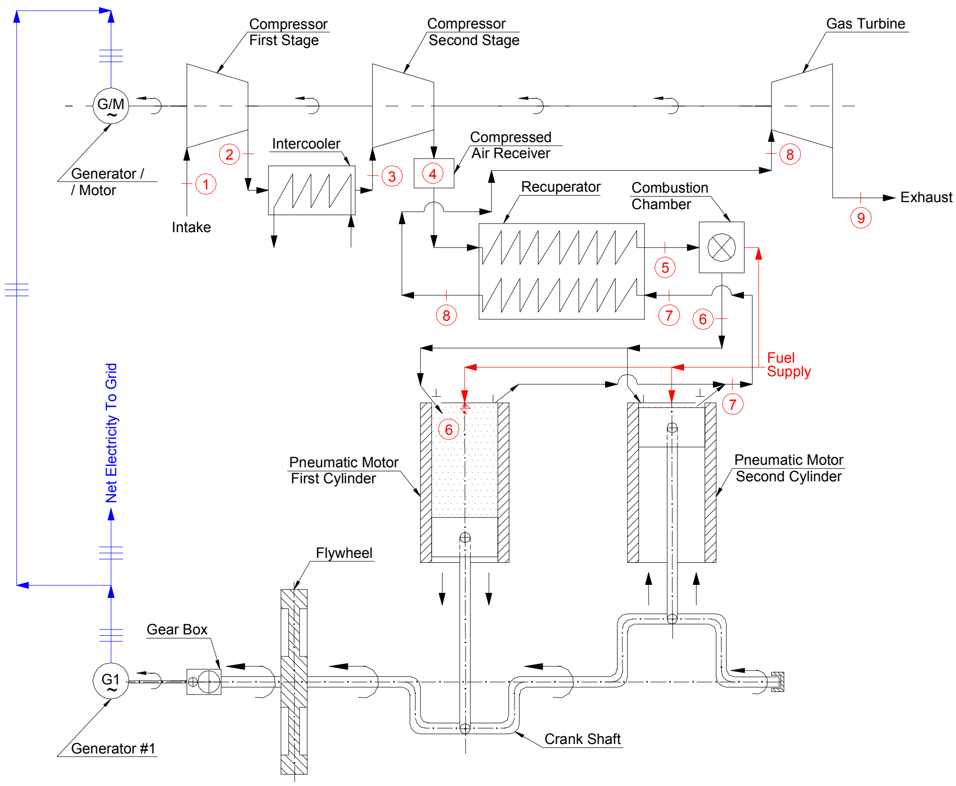


**FIG. 3** – Flow Diagram of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition – **Configuration #2**

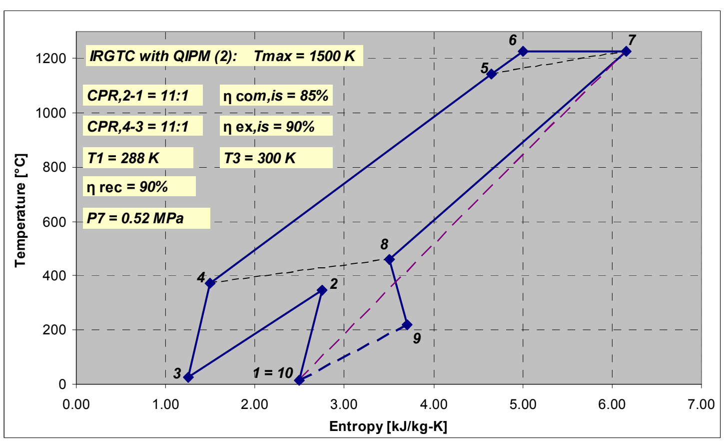


**FIG. 4** – Typical Thermodynamic Temperature-Entropy Diagram of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition – **Configuration #2**

(g) a pneumatic motor (compressed-air engine), typically using reciprocating linear (axial or radial) motion of a piston in a cylinder, for partial adiabatic expansion of the combustion gas; although two pneumatic-motor cylinders are shown, four or more cylinders may be needed;

(h) a quasi-isothermal combustor for isobaric heat addition to the combustion gas simultaneously expanding in the pneumatic-motor cylinder, virtually maintaining the constant maximum cycle temperature during the combustion-gas expansion in the pneumatic motor (isothermal process **6-7**);

(i) a gas turbine for full adiabatic expansion to a near atmospheric pressure of the combustion gas partly expanded in the pneumatic motor (adiabatic process **7-8**);

(j) a crank shaft of the pneumatic motor for conversion of a reciprocating linear motion of the piston into a rotational motion;

(k) preferably, a flywheel of the pneumatic motor for maintaining of the crank-shaft rotational speed using its inertial forces (moment of inertia), thus equalizing a potentially fluctuating torque of the pneumatic motor during startup/ operation/transients;

(l) a gear box for transmission of relatively slow rotational speed of the crank shaft into the rotational speed needed for a rotor of an electric generator; and

(m) electric generator(s) for electricity generation.

The working-gas sequence of flow in the **Configuration #1** is as follows: quasi-isothermal heat addition / gas expansion

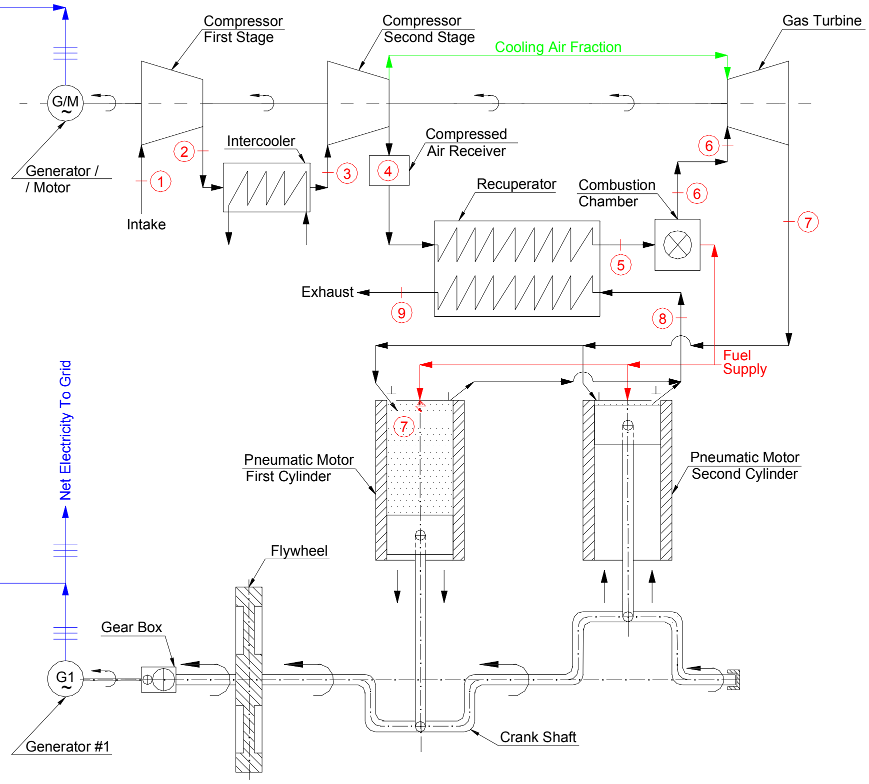


**FIG. 5** – Flow Diagram of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition – **Configuration #3**

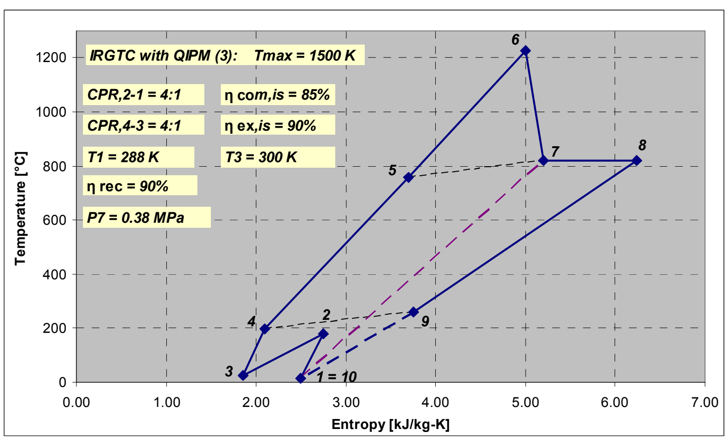


**FIG. 6** – Typical Thermodynamic Temperature-Entropy Diagram of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition – **Configuration #3**

in the pneumatic motor occurs right after the GT-cycle heat addition in the associated GT-cycle combustor, then the partly expanded combustion gas continues to expand in the GT prior to its exhaust to the GT-cycle recuperator.

As the first/second compressor stage CPR increases, the overall compressor work surpasses the GT output, which is installed on the same shaft in order to drive the compressor(s).

In this case, the electric generator ***G*** should act as an electric motor, supplied with electricity from the electric generator ***G1***, driven by the pneumatic motor.

In the **Configuration #1**: the cooling-air flow (depicted with green line) for cooling of the GT is taken from the high-pressure end of the first-stage compressor.

**Figure 3** depicts a flow diagram of the **Configuration #2** of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition, consisting of the same interconnected equipment/processes as depicted/explained in **FIG. 1** & **FIG. 2**, with the following variation in the working-gas flow sequence:

(i) quasi-isothermal heat addition / gas expansion in the pneumatic motor occurs right after the GT-cycle heat addition in the associated GT-cycle combustor, then the partly expanded combustion gas first cools down in the GT-cycle recuperator, prior to its continued expansion in the low-temperature GT and exhaust to atmosphere.

The working-gas flow sequence of the **Configuration #2** is depicted by the following processes, accordingly shown in the associated thermodynamic temperature-entropy diagram in **Figure 2**: first-stage adiabatic compression process **1-2**, isobaric intercooling process **2-3**, second-stage adiabatic compression process **3-4**, isobaric internal heat-exchange processes **4-5** and **7-8** in the recuperator, isobaric GT-combustor heat addition process **5-6**, quasi-isothermal head-addition (heat addition – adiabatic expansion) process **6-7** in the pneumatic motor, low-temperature GT adiabatic expansion process **8-9**, and isobaric heat rejection process **9-10** to atmosphere.

For this configuration, the overall compressor work for any chosen first/second compressor stage CPR surpasses the GT output, which is installed on the same shaft in order to drive the compressor(s). Thus in this case, the electric generator ***G*** always acts as an electric motor, supplied with electricity from the electric generator ***G1***, driven by the pneumatic motor.

For the **Configuration #2** there is no need for cooling of the low-temperature GT and, consequently, the cooling-air flow lines are not depicted in **FIG. 3**.

**Figure 5** depicts a flow diagram of the **Configuration #3** of an Intercooled-Recuperated GT-Cycle Engine Coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition, consisting of the same interconnected equipment/processes as depicted/explained in **FIG. 1**, with the following variation in the working-gas flow sequence:

(ii) after the GT-cycle heat addition in the associated GT-cycle combustor, the combustion gas first expands in the GT and then continues its quasi-isothermal heat addition / gas expansion in the pneumatic motor, prior to its exhaust to the GT-cycle recuperator and then to atmosphere.

The working-gas flow sequence of the **Configuration #3** is depicted by the following processes, accordingly shown in the associated thermodynamic temperature-entropy diagram in **Figure 2**: first-stage adiabatic compression process **1-2**, isobaric intercooling process **2-3**, second-stage adiabatic compression process **3-4**, isobaric internal heat-exchange processes **4-5** and **8-9** in the recuperator, isobaric GT-combustor heat addition process **5-6**, high-temperature GT adiabatic expansion process **6-7**, quasi-isothermal lower-temperature head-addition (heat addition – adiabatic expansion) process **7-8** in the pneumatic motor, and isobaric heat rejection process **9-10** to atmosphere.

For this configuration, with the first/second compressor stage CPR increase the overall compressor work surpasses the GT output, which is installed on the same shaft in order to drive the compressor(s). Thus, for this case also, the electric generator ***G*** should act as an electric motor, supplied with electricity from the electric generator ***G1***, driven by the pneumatic motor.

In the **Configuration #3**: the cooling-air flow (depicted with green line) for cooling of the GT is taken from the highest-pressure end of the second-stage compressor.

## ASSUMPTIONS AND RESULTS

This section of the paper summarizes assumptions and simplifications upon which associated thermodynamic calculations have been based and discusses obtained results.

There are two sets of assumptions used in associated thermodynamic calculations: (1) **simplified** (more ideal) set of assumptions; and (2) **realistic** (more real) set of assumptions.

In addition, the following general assumptions have been used throughout the calculations: ambient temperature of 15°C (288 K or 59°F), temperature of compressed air intercooling 27°C (300 K or ~81°F), two maximum cycle temperatures of 1500 K (1227°C or ~2241°F) and 1700 K (1427°C or ~2601°F), respectively, used in both the GT and the pneumatic-motor, a GT compression process isentropic efficiency of 85%, a GT expansion process isentropic efficiency of 90%, a recuperator effectiveness of 90%, a pressure drop of 2.5% in the intercooler, a pressure drop of 2% in the combustor and the pneumatic motor, and a pressure drop of 5% in each side of the recuperator.

An additional general assumption is that the final expansion pressure at the end of the pneumatic-motor quasi-isothermal heat addition/expansion process in all three hybrid-energy-system configurations corresponds to the initial specific volume of inlet ambient air, since the working-gas exhaust process in the pneumatic-motor cylinder is considered isochoric (constant-volume).

Thermal efficiency of the hybrid energy system has been estimated according to its definition from the First Law of Thermodynamics: 100% minus the ratio of the energy rejected to atmosphere and the total heat input.

Accordingly, thermal efficiency of the **Configurations #1** & **#2** is generally given by the following expressions, in accordance with thermodynamic states numerals shown in **FIGS. 2** & **4**, respectively:

$$\eta_1 = 1 - \frac{\left[c_P \cdot (T_2 - T_3) \cdot (1 - m_{cool}) + c_P \cdot (T_9 - T_1)\right]}{(1 - m_{cool}) \cdot \left[c_P \cdot (T_6 - T_5) + R \cdot T_{6\equiv 7} \cdot LN\left(\frac{p_6}{p_7}\right)\right]}$$

$$\eta_2 = 1 - \frac{\left[c_P \cdot (T_2 - T_3) + c_P \cdot (T_9 - T_1)\right]}{\left[c_P \cdot (T_6 - T_5) + R \cdot T_{6\equiv 7} \cdot LN\left(\frac{p_6}{p_7}\right)\right]}$$

Similarly, thermal efficiency of the **Configuration #3** is generally given by the following expression, in accordance with thermodynamic states numerals shown in **FIG. 6**:

$$\eta_3 = 1 - \frac{\left[c_P \cdot (T_2 - T_3) + c_P \cdot (T_9 - T_1)\right]}{\left[c_P \cdot (T_6 - T_5) \cdot (1 - m_{cool}) + R \cdot T_{7\equiv 8} \cdot LN\left(\frac{p_7}{p_8}\right)\right]}$$

**"*R*"** in the above expressions designates gas constant of the working gas, typically estimated as a ratio of the universal gas constant (8.314 kJ/(k-mol*K)) and the molecular mass of the working gas (28.96 kg/k-mol for air). Also, **"$C_p$"** in the above expressions designates working-gas specific heat at constant pressure in kJ/kg. It is considered constant (invariable) or variable with gas temperature, depending on assumptions. **“$m_{cool}$”** in the above expressions designates a GT cooling-air fraction, depending on adopted assumptions.

Cycle thermal efficiency of the hybrid energy system can be rewritten in the form of ratio Total-Power-Output/Total-Heat-Input, where compressor-stages and GT outlet temperatures $T_2$, $T_4$ and $T_7/T_8/T_9$, respectively, take into account inefficiencies in the compression and expansion processes, respectively. Thus, expressions for thermal efficiencies of Configurations #1, #2 and #3 become as follows:

$$\eta_1 = \frac{\left[c_P \cdot \frac{(T_7 - T_8)}{(1-m_{cool})} - c_P \cdot \left[\frac{(T_2 - T_1)}{(1-m_{cool})} + (T_4 - T_3)\right] + \frac{R \cdot T_{6\equiv 7}}{(1-m_{cool})} \cdot LN\left(\frac{p_6}{p_7}\right)\right]}{\left[c_P \cdot (T_6 - T_5) + R \cdot T_{6\equiv 7} \cdot LN\left(\frac{p_6}{p_7}\right)\right]}$$

$$\eta_2 = \frac{\left[c_P \cdot (T_8 - T_9) - c_P \cdot (T_2 - T_1 + T_4 - T_3) + R \cdot T_{6\equiv 7} \cdot LN\left(\frac{p_6}{p_7}\right)\right]}{\left[c_P \cdot (T_6 - T_5) + R \cdot T_{6\equiv 7} \cdot LN\left(\frac{p_6}{p_7}\right)\right]}$$

$$\eta_3 = \frac{\left[c_P \cdot (T_6 - T_7) - c_P \cdot (T_2 - T_1 + T_4 - T_3) + R \cdot T_{7\equiv 8} \cdot LN\left(\frac{p_7}{p_8}\right)\right]}{\left[c_P \cdot (T_6 - T_5) \cdot (1-m_{cool}) + R \cdot T_{7\equiv 8} \cdot LN\left(\frac{p_7}{p_8}\right)\right]}$$

The main deficiency of the above expressions for estimate of cycle thermal efficiency is assuming of the heat addition process, and in the same time work extraction process from the pneumatic motor (PM), as an ideal isothermal (constant-temperature) process. In reality, this process is a result of interference of numerous small-scale adiabatic expansions and isobaric heat additions (sort of a horizontal zigzag line in temperature-entropy diagram), simultaneously interconnected within a PM cylinder-piston motor. The larger the number of these small-scale processes, the closer is the process to the isothermal heat addition/work extraction process and the closer is the cycle thermal efficiency to an ideal one. Besides, pneumatic motors have inherent inefficiencies (internal leakage, internal friction), which reduce/limit power output than can be extracted from the compressed air at an operating PM expansion pressure ratio. Conveniently, **Configurations #1** & **#2** of the proposed hybrid energy system can considerably compensate any possible loss of internal PM efficiency, because of another component – GT – which is located downstream of the PM in the direction of working-gas flow and can retake and reuse whatever surplus of working-gas flow rate and/or pressure comes out from the PM.

*<u>Simplified Set of Assumptions and Associated Results</u>*

In addition to general assumptions, the simplified set of assumptions includes also the following assumptions:

(a) invariable constant-pressure specific heats of working gas;

(b) invariable ratios of specific heats at constant pressure, $C_p$, and at constant volume, $C_v$, of *1.40* for air, in both compression and expansion processes;

(c) neglecting of the GT air cooling fraction, that is, use of an uncooled GT in all three hybrid-energy-system configurations.

**Figure 7** depicts a graph of the thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage-compressor compression pressure ratio (CPR) for the maximum cycle temperature of **1500 K** (1227°C or ~2241°F), using the **simplified** set of assumptions, for all three hybrid-energy-system configurations. It is implicit that the second-stage-compressor CPR is identical to the first-stage-compressor CPR.

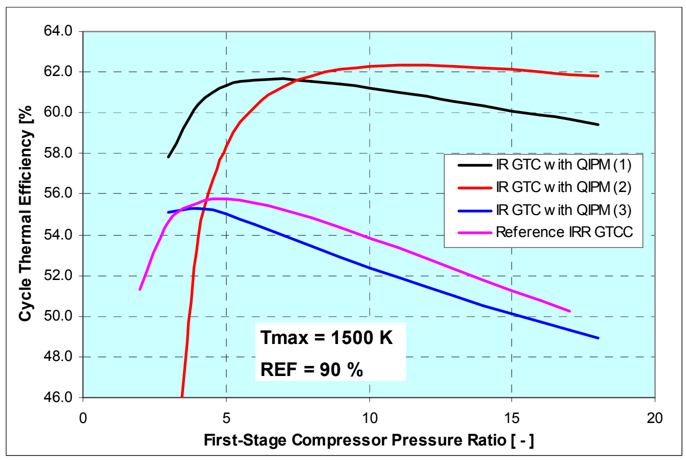


**FIG. 7** – Thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage CPR for the maximum cycle temperature of **1500 K**, using **simplified** set of assumptions

From the above graph it can be noted that, under the **simplified** assumptions, **Configurations #1** and **#2** seem to be the most thermally efficient with similar values of thermal efficiency **~62%** (LHV) for the first-stage compressor CPR of 6:1-12:1. On the other hand, **Configuration #3** seems to be the least thermally efficient, with thermal efficiency of about 55.3% (LHV) at the first-stage compressor CPR of ~4:1. For illustration purposes, the same graph depicts also thermal efficiency curve for the reference intercooled-reheat-recuperated (IRR) GT combined-cycle, reaching thermal efficiency of nearly 56% at the first-stage compressor CPR of ~5:1.

Similarly, **Figure 8** depicts a graph of the thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage-compressor CPR for the maximum cycle temperature of **1700 K** (1427°C or ~2601°F), using the simplified set of assumptions, for all three hybrid-energy-system configurations. Likewise, it is implicit that the second-stage-compressor CPR is identical to the first-stage-compressor CPR.

From the graph shown below in **FIG. 8** it can be noted that, under the simplified assumptions, **Configuration #2** seems to be the most thermally efficient, with thermal efficiency of about **66%** (LHV) for the first-stage compressor CPR of ~12:1. **Configuration #1** is similarly but slightly less thermally efficient, with thermal efficiency of about **65%** (LHV) for the first-stage compressor CPR of ~7:1. On the other hand, **Configuration #3** seems to be the least thermally efficient, with thermal efficiency of about 59% (LHV) at the first-stage compressor CPR of ~4:1. Similarly, the graph also depicts thermal efficiency curve for the reference IRR GT combined-cycle, reaching thermal efficiency of about 59% at the first-stage compressor CPR of ~6:1.

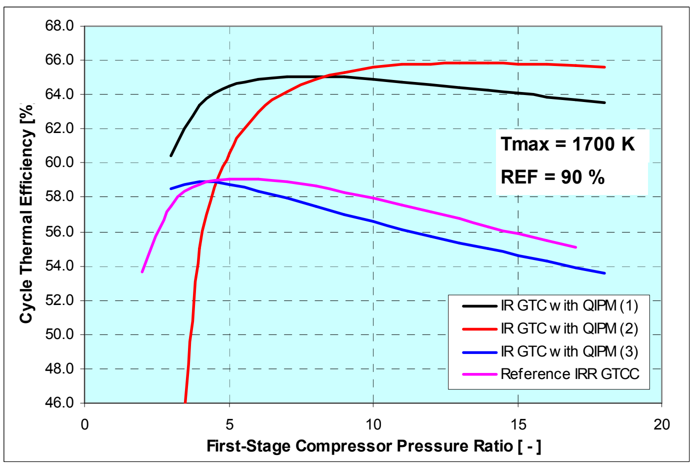


**FIG. 8** – Thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage CPR for the maximum cycle temperature of **1700 K**, using **simplified** set of assumptions

*Realistic Set of Assumptions and Associated Results*

In addition to general assumptions, the realistic set of assumptions includes the following assumptions:

(a) variable constant-pressure specific heats of working gas;

(b) quasi-variable ratios of specific heats at constant pressure, $C_p$, and at constant volume, $C_v$, for air/working gas, in both compression (*1.40*) and expansion (*1.33*) processes;

(c) accounting for a realistic GT cooling-air fraction, that is, use of a cooled GT.

Related to realistic assumption (c), ref. [4] gives a very useful estimate of cooling-air fractions for GT-s, depending on the maximum GTIT and the CPR. The crucial referent value in [4] is the cooling-air fraction of about 22% for a max. GT with a TIT of 1425°C (1698 K or 2597°F) and a CPR of 17:1. The realistic set of assumptions has used this referent value at the referent parameters and has assumed a simple linear change of the real TIT for a cooled GT, $T_{IT,cool}$, with respect to the cooling-air fraction, $m_{cool}$, the maximum TIT, $T_{IT,max}$, and the air temperature at the compressor outlet, $T_{comp,out}$, as follows:

$$T_{IT,cool} = T_{IT,\max} \cdot (1 - m_{cool}) + T_{comp,out} \cdot m_{cool}$$

The cooling-air fraction, $m_{cool}$, is given by assumed proportional function of the referent parameters, $T_{IT,max,ref} \approx$ 1700 K, $CPR_{ref}$ = 17:1, and $m_{cool,ref}$ = 0.22, as follows:

$$m_{cool} = m_{cool,ref} \cdot \frac{T_{comp,out}}{T_{comp,out,ref}} \cdot \frac{(T_{IT,\max} - 1000)}{(T_{IT,\max,ref} - 1000)}$$

Consequently, **Figure 9** depicts a graph of the thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage-compressor CPR for the maximum cycle temperature of **1500 K** (1227°C or ~2241°F), using the **realistic** set of assumptions, for all three hybrid-energy-system configurations.

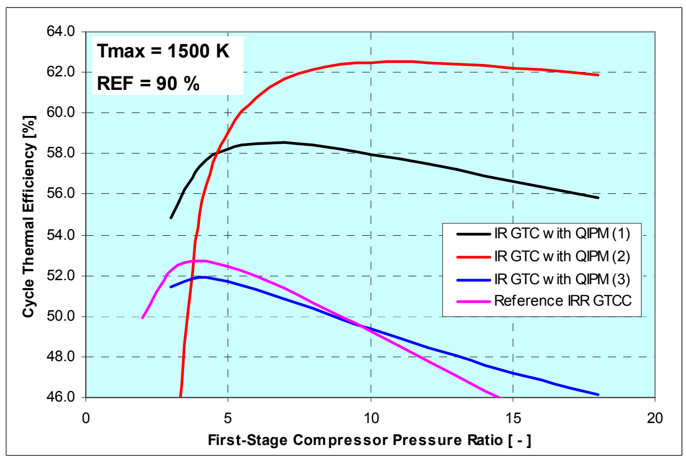


**FIG. 9** – Thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage CPR for the maximum cycle temperature of **1500 K**, using **realistic** set of assumptions

From the above graph it can be noted that, under the **realistic** assumptions, **Configuration #2** again seems to be the most thermally efficient, with thermal efficiency of about **~62.5%** (LHV) for the first-stage compressor CPR of ~11:1. On the other hand, **Configuration #1** is significantly less thermally efficient, with thermal efficiency of above ~58% (LHV) for the first-stage compressor CPR of ~7:1. As with earlier results,

**Configuration #3** seems to be the least thermally efficient, with thermal efficiency of about 52% (LHV) at the first-stage compressor CPR of ~4:1. For illustration purposes, the same graph depicts also thermal efficiency curve for the reference IRR GT combined-cycle, reaching thermal efficiency of about 52.7% at the first-stage compressor CPR of ~4:1.

The cycle net specific power output, both total and broken to power consumptions/generations from each of major components (compressor, GT, PM) is shown in **Summary Table of Results** on the last pages, versus maximum cycle temperature and the first/second-stage CPR.

Similarly, **Figure 10** depicts a graph of the thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage-compressor CPR for the maximum cycle temperature of **1700 K** (1427°C or ~2601°F), using the realistic set of assumptions, for all three hybrid-energy-system configurations.

From the graph shown below in **FIG. 10** it can be noted that, under the realistic assumptions, **Configuration #2** again seems to be the most thermally efficient, with thermal efficiency of about **66%** (LHV) for the first-stage compressor CPR of ~12.5:1. On the other hand, **Configuration #1** is significantly less thermally efficient, with thermal efficiency of about ~60.5% (LHV) for the first-stage compressor CPR of ~7.5:1. As with earlier results, **Configuration #3** seems to be the least thermally efficient, with thermal efficiency of about 53.3% (LHV) at the first-stage compressor CPR of ~4:1. Similarly, the graph also depicts thermal efficiency curve for the reference IRR GT combined-cycle, reaching thermal efficiency of about 54.6% at the first-stage compressor CPR of ~4:1.

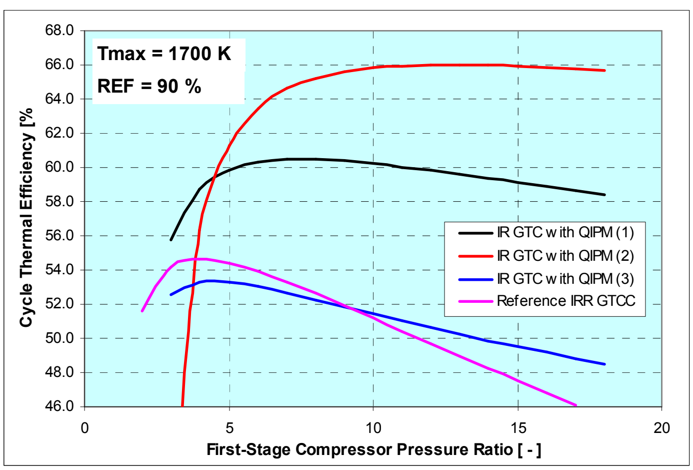


**FIG. 10** – Thermal efficiency change of an Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with quasi-isothermal heat addition with the first-stage CPR for the maximum cycle temperature of **1700 K**, using **realistic** set of assumptions

Similarly, the cycle net specific power output, both total and broken to power consumptions/generations from each of major components (compressor, GT, PM) is shown in **Summary Table of Results** (the last pages), versus maximum cycle temperature and the first/second-stage CPR.

From the graphs shown in **FIGS. 7, 8, 9** and **10**, it can be noted that the most thermally efficient configuration of the *Intercooled-Recuperated GT-Cycle Engine coupled with a Pneumatic Motor with Quasi-Isothermal Heat Addition* appears to be the **Configuration #2**, using both simplified and realistic set of assumptions. The reason and explanation for this result is most probably the absence of the cooling-air flow requirement, since there is no need for cooling of the low-temperature GT of the **Configuration #2**. Consequently, the **Configurations #1** and **#3** are significantly sensitive to the cooling-air flow requirement and their thermal efficiencies are proportionally lower when the realistic assumptions are used. Moreover, even the net specific power output of the **Configuration #2** is the largest of all three considered configurations when the realistic assumptions are used for estimation.

Overall cycle thermal efficiency increases with the ambient temperature decrease for any hybrid-energy-system configuration. For example, assuming an ambient temperature of 253 K (-20°C or -4°F), the maximum cycle temperature of 1700 K (1427°C or ~2601°F) and the realistic set of assumptions, maximum thermal efficiencies of the Configurations #1, #2 and #3 become: 62.3%, 70.5% and 54.8%, respectively.

The deficiencies/penalties associated with the use of the proposed hybrid energy system, especially its preferred **Configuration #2**, are likely the following: (1) the need for a large and an expensive recuperator, which probably has to be manufactured from special materials, like ceramics (particularly for very high maximum cycle temperatures); and (2) a large-size and potentially expensive pneumatic motor, likely to be made of a high-alloyed steel.

Typically, sizes of radial-piston pneumatic motors range up to ~35 hp (~26 kW) at speeds of ~4,500 rpm. Axial-piston PM-s have usually even smaller power outputs. Rotary-vane PM-s can operate at speeds of up to 25,000 rpm and can deliver more specific power per pound (weight) than piston PM-s. Turbine PM is the most efficient type due to absence of internal friction, which results in no need for extensive lubrication. Generally, it can be said that PM-s are best suited for small-scale applications, such as within distributed power generation systems. However, because of their constructional similarity with internal-combustion engines, especially when an additional combustion is to take place internally within them, pneumatic motors could certainly be manufactured to larger sizes, most likely resulting in an increased capital cost.

### *Brief Techno-Economic Analysis*

Considering the above technical requirements, capital cost of this hybrid energy system could exceed capital cost of a reference 58%-efficient 400-MWe GTCC plant for more than 2 times: for example ~US$1000/kW of power versus ~US$450/kW of power. Assuming: current natural-gas price of US$4 per million of BTU (November 2010), annual plant operating hours of 5000, electricity price of 5 cents/kWh and the **Configuration #2** with very high thermal efficiency of **~66%** (from the above **FIG. 10**), it is possible to roughly estimate simple return on investment periods for the reference GTCC plant and the GT-PM hybrid energy plant.

(1) Return on investment (*ROI*) period for the reference 58%-efficient 400-MWe GTCC plant is estimated at about **3.4 years**, as follows:

- 400-MWe GTCC plant capital cost:

$$CC_{plant} = \left(450 \, \frac{\$}{kW}\right) \cdot \left(400{,}000 \; kW\right) = 180 \; M\$$$

- GTCC plant annual electricity gross profit/gain:

$$G_{electric,year} = \left(5000 \, \frac{hours}{year}\right) \cdot \left(400{,}000 \; kW\right) \cdot \left(0.05 \, \frac{\$}{kWh}\right)$$

$$\Rightarrow G_{electric,year} = 100{,}000{,}000 \; \frac{\$}{year} = 100 \; \frac{M\$}{year}$$

- GTCC plant annual fuel consumption:

$$Q_{fuel,year} = \frac{\left(400{,}000 \; kJ/s\right)}{\left[0.58 \cdot \left(1.055 \, \frac{kJ}{BTU}\right)\right]} \cdot \left(5000 \, \frac{hours}{year}\right) \cdot \left(3600 \, \frac{s}{h}\right)$$

$$\Rightarrow Q_{fuel,year} = 11{,}761{,}054.6 \; \frac{MBTU}{year}$$

- GTCC plant annual fuel cost:

$$C_{fuel,year} = Q_{fuel,year} \cdot \left(4.0 \, \frac{\$}{MBTU}\right) \cong 47 \; \frac{M\$}{year}$$

- GTCC plant return on investment period:

$$ROI = \frac{CC_{plant}}{\left(C_{electric,year} - C_{fuel,year}\right)} = \frac{180}{\left(100 - 47\right)} \left[y\right] \cong 3.4 \; years$$

(2) Return on investment (*ROI*) period for the 66%-efficient **Configuration #2** of the GT-PM hybrid energy system is estimated at about **6.8 years**, as follows:

- 400-MWe GT-PM hybrid energy plant capital cost:

$$CC_{plant} = \left(1000 \, \frac{\$}{kW}\right) \cdot \left(400{,}000 \; kW\right) = 400 \; M\$$$

- GT-PM hybrid energy plant annual electricity gross profit/gain:

$$G_{electric,year} = \left(5000 \, \frac{hours}{year}\right) \cdot \left(400{,}000 \; kW\right) \cdot \left(0.05 \, \frac{\$}{kWh}\right)$$

$$\Rightarrow G_{electric,year} = 100{,}000{,}000 \; \frac{\$}{year} = 100 \; \frac{M\$}{year}$$

- GT-PM hybrid energy plant annual fuel consumption:

$$Q_{fuel,year} = \frac{\left(400{,}000 \; kJ/s\right)}{\left[0.66 \cdot \left(1.055 \, \frac{kJ}{BTU}\right)\right]} \cdot \left(5000 \, \frac{hours}{year}\right) \cdot \left(3600 \, \frac{s}{h}\right)$$

$$\Rightarrow Q_{fuel,year} = 10{,}340{,}370 \; \frac{MBTU}{year}$$

- GT-PM hybrid energy plant annual fuel cost:

$$C_{fuel,year} = Q_{fuel,year} \cdot \left(4.0 \, \frac{\$}{MBTU}\right) \cong 41.36 \; \frac{M\$}{year}$$

- GT-PM hybrid energy plant return on investment period:

$$ROI = \frac{CC_{plant}}{\left(C_{electric,year} - C_{fuel,year}\right)} = \frac{400}{\left(100 - 41.36\right)} \cong 6.8 \; years$$

## CONCLUSION

A hybrid energy system has been proposed consisting of an intercooled-recuperated GT-cycle engine coupled with a pneumatic motor (compressed-air engine) with quasi-isothermal heat addition process. The quasi-isothermal nature of the heat addition process within the pneumatic motor is a result of averaging of the two thermodynamic processes, simultaneously interconnected within a cylinder-piston motor: isobaric heat addition and adiabatic gas expansion. Three configurations of the hybrid energy system have been considered and compared: (1) **Configuration #1**: quasi-isothermal heat addition/gas expansion in the PM occurs right after the GT-cycle heat addition in the associated GT-cycle combustor, then the partly expanded combustion gas continues to expand in the GT prior to its exhaust to the GT-cycle recuperator; (2) **Configuration #2**: quasi-isothermal heat addition/gas expansion in the PM occurs right after the GT-cycle heat addition in the associated GT-cycle combustor, then the partly expanded combustion gas first cools down in the GT-cycle recuperator, prior to its continued expansion in the GT and exhaust to atmosphere, and (3) **Configuration #3**: after the GT-cycle heat addition in the associated GT-cycle combustor, the combustion gas first expands in the GT and then continues its quasi-isothermal heat addition / gas expansion in the PM, prior to exhaust to the GT-cycle recuperator and then to atmosphere. For thermal efficiency comparison purposes, two sets of assumptions have been considered: (a) **simplified** set of assumptions (invariable specific heats/ratios of specific heats of working gas, an uncooled GT); and (b) **realistic** set of assumptions (variable or quasi-variable specific heats/ratios of specific heats of working gas, a cooled GT with a realistic GT cooling-air fraction).

The analysis has indicated that the **Configuration (2)** is the most efficient one, having an estimated overall cycle thermal efficiency of ~**62-66%** for the maximum GT/PM inlet temperature of **1500-1700 K** for both set of assumptions, simplified and realistic. The superiority of the **Configuration #2** compared to other two configurations results probably from the use of a low-temperature GT which does not need cooling, as opposed to two other **Configurations (1)** and **(3)**, which comprise typical high-temperature GT-s requiring comparatively large cooling-air fractions.

The deficiencies/penalties associated with the use of the proposed hybrid energy system, especially its preferred **Configuration #2**, are the following: (1) the need for a large and an expensive recuperator, which would have to be manufactured from special materials, like ceramics (particularly for very high maximum cycle temperatures); and (2) a large-size and potentially expensive pneumatic motor, made of high-alloyed steel. Nevertheless, the favored **Configuration #2** of the hybrid system could yield a reasonable return on investment period, owing to its very high thermal efficiency and also to a high specific power output (max. from ~1127 kJ/kg to ~1325 kJ/kg) and can thus be readily recommended.

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**SUMMARY TABLE OF RESULTS – Total cycle net specific power output and its breakdown to power consumptions/generations from each of major components**

| **Hybrid Energy System Configuration #1** | **Total Cycle Net Specific Power Output [kJ/kg]** | | Compressor Specific Power Input [kJ/kg] | | Gas-Turbine Specific Power Output [kJ/kg] | | Pneumatic-Motor Specific Power Output [kJ/kg] | | **Cycle Thermal Efficiency [%]** | |
|---|---|---|---|---|---|---|---|---|---|---|
| First-Stage CPR [-] | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K |
| 3 : 1 | **418.266** | **468.014** | 244.287 | 239.511 | 473.138 | 551.884 | 189.415 | 155.641 | **54.855** | **55.791** |
| 4 : 1 | **560.570** | **628.825** | 320.455 | 313.554 | 471.364 | 548.359 | 409.660 | 394.020 | **57.368** | **58.755** |
| 5 : 1 | **663.661** | **744.658** | 383.441 | 374.545 | 470.072 | 545.698 | 577.030 | 573.505 | **58.234** | **59.880** |
| 6 : 1 | **742.834** | **833.108** | 437.560 | 426.771 | 469.091 | 543.599 | 711.303 | 716.280 | **58.511** | **60.334** |
| 7 : 1 | **806.027** | **903.297** | 485.250 | 472.654 | 468.328 | 541.897 | 822.949 | 834.055 | **58.526** | **60.481** |
| 8 : 1 | **857.864** | **960.534** | 528.035 | 513.706 | 467.726 | 540.490 | 918.173 | 933.750 | **58.408** | **60.468** |
| 9 : 1 | **901.263** | **1008.162** | 566.939 | 550.941 | 467.247 | 539.310 | 1000.955 | 1019.793 | **58.218** | **60.364** |
| 10 : 1 | **938.175** | **1048.418** | 602.686 | 585.072 | 466.866 | 538.311 | 1073.995 | 1095.180 | **57.988** | **60.207** |
| 11 : 1 | **969.965** | **1082.864** | 635.807 | 616.627 | 466.564 | 537.459 | 1139.208 | 1162.031 | **57.735** | **60.017** |
| 12 : 1 | **997.624** | **1112.630** | 666.704 | 646.002 | 466.327 | 536.730 | 1198.001 | 1221.902 | **57.470** | **59.807** |
| 13 : 1 | **1021.890** | **1138.561** | 695.692 | 673.507 | 466.144 | 536.104 | 1251.437 | 1275.965 | **57.199** | **59.584** |
| 14 : 1 | **1043.328** | **1161.301** | 723.020 | 699.388 | 466.007 | 535.565 | 1300.340 | 1325.124 | **56.926** | **59.355** |
| 15 : 1 | **1062.379** | **1181.352** | 748.889 | 723.844 | 465.911 | 535.102 | 1345.357 | 1370.094 | **56.653** | **59.121** |
| 16 : 1 | **1079.393** | **1199.113** | 773.465 | 747.037 | 465.848 | 534.705 | 1387.011 | 1411.445 | **56.383** | **58.885** |
| 17 : 1 | **1094.653** | **1214.904** | 796.887 | 769.103 | 465.815 | 534.365 | 1425.724 | 1449.642 | **56.115** | **58.650** |
| 18 : 1 | **1108.387** | **1228.986** | 819.270 | 790.157 | 465.809 | 534.077 | 1461.848 | 1485.066 | **55.851** | **58.415** |

| **Hybrid Energy System Configuration #2** | **Total Cycle Net Specific Power Output [kJ/kg]** | | Compressor Specific Power Input [kJ/kg] | | Gas-Turbine Specific Power Output [kJ/kg] | | Pneumatic-Motor Specific Power Output [kJ/kg] | | **Cycle Thermal Efficiency [%]** | |
|---|---|---|---|---|---|---|---|---|---|---|
| First-Stage CPR [-] | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K |
| 3 : 1 | **130.142** | **115.792** | 256.228 | 256.228 | 203.009 | 223.545 | 183.361 | 148.474 | **40.985** | **37.909** |
| 4 : 1 | **307.953** | **327.382** | 337.707 | 337.707 | 217.086 | 238.490 | 428.573 | 426.598 | **54.948** | **56.311** |
| 5 : 1 | **441.777** | **487.451** | 405.683 | 405.683 | 228.831 | 250.958 | 618.629 | 642.176 | **59.015** | **61.305** |
| 6 : 1 | **548.281** | **615.430** | 464.533 | 464.533 | 238.998 | 261.753 | 773.815 | 818.210 | **60.778** | **63.474** |
| 7 : 1 | **636.228** | **721.557** | 516.738 | 516.738 | 248.018 | 271.328 | 904.949 | 966.967 | **61.664** | **64.602** |
| 8 : 1 | **710.786** | **811.879** | 563.857 | 563.857 | 256.159 | 279.971 | 1018.484 | 1095.765 | **62.134** | **65.239** |
| 9 : 1 | **775.248** | **890.261** | 606.937 | 606.937 | 263.602 | 287.872 | 1118.582 | 1209.325 | **62.377** | **65.612** |
| 10 : 1 | **831.840** | **959.317** | 646.721 | 646.721 | 270.475 | 295.170 | 1208.086 | 1310.868 | **62.487** | **65.827** |
| 11 : 1 | **882.137** | **1020.898** | 683.757 | 683.757 | 276.874 | 301.963 | 1289.020 | 1402.692 | **62.514** | **65.942** |
| 12 : 1 | **927.289** | **1076.360** | 718.460 | 718.460 | 282.870 | 308.328 | 1362.879 | 1486.492 | **62.487** | **65.992** |
| 13 : 1 | **968.164** | **1126.726** | 751.154 | 751.154 | 288.519 | 314.325 | 1430.800 | 1563.556 | **62.423** | **65.998** |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 14 : 1 | **1005.431** | **1172.786** | 782.099 | 782.099 | 293.865 | 320.001 | 1493.664 | 1634.884 | **62.334** | **65.971** |
| 15 : 1 | **1039.615** | **1215.162** | 811.502 | 811.502 | 298.945 | 325.394 | 1552.172 | 1701.270 | **62.228** | **65.923** |
| 16 : 1 | **1071.137** | **1254.353** | 839.537 | 839.537 | 303.789 | 330.536 | 1606.885 | 1763.354 | **62.111** | **65.858** |
| 17 : 1 | **1100.341** | **1290.764** | 866.346 | 866.346 | 308.420 | 335.453 | 1658.266 | 1821.657 | **61.985** | **65.781** |
| 18 : 1 | **1127.506** | **1324.730** | 892.052 | 892.052 | 312.862 | 340.168 | 1706.697 | 1876.613 | **61.853** | **65.695** |
| | | | | | | | | | | |
| **Hybrid Energy System Configuration #3** | **Total Cycle Net Specific Power Output [kJ/kg]** | | Compressor Specific Power Input [kJ/kg] | | Gas-Turbine Specific Power Output [kJ/kg] | | Pneumatic-Motor Specific Power Output [kJ/kg] | | **Cycle Thermal Efficiency [%]** | |
| First-Stage CPR [-] | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K | 1500 K | 1700 K |
| 3 : 1 | **389.169** | **447.918** | 256.228 | 256.228 | 386.426 | 445.654 | 258.971 | 258.492 | **51.419** | **52.567** |
| 4 : 1 | **480.934** | **551.439** | 337.707 | 337.707 | 384.157 | 441.305 | 434.483 | 447.841 | **51.904** | **53.317** |
| 5 : 1 | **546.633** | **625.576** | 405.683 | 405.683 | 382.505 | 438.026 | 569.811 | 593.233 | **51.720** | **53.313** |
| 6 : 1 | **596.619** | **682.014** | 464.533 | 464.533 | 381.253 | 435.444 | 679.899 | 711.103 | **51.321** | **53.048** |
| 7 : 1 | **636.217** | **726.761** | 516.738 | 516.738 | 380.279 | 433.352 | 772.676 | 810.147 | **50.846** | **52.680** |
| 8 : 1 | **668.503** | **763.286** | 563.857 | 563.857 | 379.511 | 431.623 | 852.849 | 895.519 | **50.351** | **52.272** |
| 9 : 1 | **695.405** | **793.760** | 606.937 | 606.937 | 378.901 | 430.175 | 923.441 | 970.522 | **49.857** | **51.853** |
| 10 : 1 | **718.203** | **819.627** | 646.721 | 646.721 | 378.415 | 428.950 | 986.509 | 1037.398 | **49.375** | **51.435** |
| 11 : 1 | **737.786** | **841.888** | 683.757 | 683.757 | 378.029 | 427.906 | 1043.514 | 1097.739 | **48.908** | **51.026** |
| 12 : 1 | **754.797** | **861.264** | 718.460 | 718.460 | 377.727 | 427.012 | 1095.529 | 1152.712 | **48.459** | **50.628** |
| 13 : 1 | **769.710** | **878.292** | 751.154 | 751.154 | 377.494 | 426.244 | 1143.370 | 1203.202 | **48.028** | **50.244** |
| 14 : 1 | **782.889** | **893.378** | 782.099 | 782.099 | 377.320 | 425.584 | 1187.667 | 1249.892 | **47.613** | **49.872** |
| 15 : 1 | **794.613** | **906.836** | 811.502 | 811.502 | 377.197 | 425.018 | 1228.918 | 1293.321 | **47.216** | **49.514** |
| 16 : 1 | **805.104** | **918.918** | 839.537 | 839.537 | 377.117 | 424.531 | 1267.524 | 1333.923 | **46.833** | **49.169** |
| 17 : 1 | **814.541** | **929.821** | 866.346 | 866.346 | 377.075 | 424.116 | 1303.812 | 1372.051 | **46.466** | **48.836** |
| 18 : 1 | **823.067** | **939.708** | 892.052 | 892.052 | 377.067 | 423.763 | 1338.052 | 1407.997 | **46.112** | **48.515** |