The process of isothermal compression of gasses at sub-atmospheric pressures through regulated water injection in Braysson cycles

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Although the Braysson cycle constitutes the ideal limit for the Combined Cycle Power Plants, its actual implementation has not been achieved due to the difficulty in building the required isothermal compressor. The present study proposes the incorporation of regulated water injection during the final compression, which could maintain the temperature constant due to the evaporation. The analysis for the thermodynamic implications of the injection on the ideal version of the Braysson cycle indicates that the (ideal cycle) efficiency reduction will be minimal. The study provides an analysis for the water injection rate that will permit such a process and shows that the additional work needed to drive the process will not be affected significantly by the injection. In addition, it shows that the minimum temperature of the Braysson cycle will be lower than the corresponding level of the conventional (Gas–Steam turbine Combined cycle plants), something that could improve the efficiency as well. Finally it shows that the process may be expressed by a polytropic relationship of the type $p v^b = constant$, where $b = 1.06$.

1. Introduction

Bottoming cycles constitute additions to the standard (topping) ones by exploiting the high temperature enthalpy carried along by the exhausted flue gases. Such Topping—Bottoming cycle combinations that permit expansion to a temperature equal to the atmospheric one may be called “full expansion” ones. Kerrebrok [1] has shown that the incorporation of the sequence isentropic expansion—isoenthalpic compression as a Bottoming cycle constitutes the ideal limit for all such combined cycles. Over the last 10 or so years the incorporation of the isenthalpic compression—cooling process into a thermodynamic cycle was evaluated by a number of relevant studies. Goto et al. [2] have examined the ideal versions of the Diesel and Brayton cycles incorporating such bottoming additions to create the corresponding “full” expansion ones. They showed that the combined ideal cycle thermal efficiencies could exceed the 80% level for the case of the perfect air, when their simple (conventional) versions are limited around the 60% level (for practical pressure ratios). Recent reports call the single stage version of the full expansion cycle “the Braysson cycle”. Second law and finite rate analysis of this cycle have been provided by Zheng et al. [3] and Zhou et al. [4], respectively.

Unfortunately, it is hard to achieve isothermal compression through conventional heat exchangers, since it requires a very close co-ordination between the heat and work transfer operations. Actual implementation so far has been extremely limited. The cryogenic engine built by Ordonez [5] is probably the best-known case where a tubular heat exchanger was inserted inside a reciprocating engine cylinder for the purposes of implementing an isothermal expansion process. No wonder that the actual results were rather disappointing. Frost et al. [6] have proposed a combined cycle that is equivalent to a full expansion Brayton one. The topping one was a typical Brayton cycle, while the bottoming (termed the “Ericsson Cycle” by the authors) receives heat from the topping cycle (Gas Turbine or Diesel Engine) through a counter current, constant pressure heat exchanger. The rest of the cycle is completed by an isentropic expansion (up to the atmospheric temperature)—isothermal compression—cooling sequence. The last process involves a rotating heat pipe. They predicted a real plant total efficiency above the 50% level.

Water evaporation has been considered for quite some time as a heat sink mechanism for isothermal (or nearly so) compression processes, for Reciprocating Engine and Compressors as well as Gas Turbine based plants. The earliest extensive tests employing Water Injection in reciprocating compressors appears to have been conducted in the Soviet Union, starting around 1955. Gal’perin et al. [7] provide the relevant biographical data about these efforts for the period before 1967. In the same paper they developed the
thermodynamic analysis for the modifications imposed by the
water injection on the polytropic exponent of a general (not
isothermal) compression process. Kabakov and Sichbera [8]
and Dobrokhvorov et al. [9] published similar analytical and experi-
mental studies. More recently Zhao et al. [10] examined the
“isothermal” compression for a scroll compressor (forming part of
an automotive fuel cell system) incorporating injection of water
and compared the generated experimental data against an (rather
complex) analytical model for the entire process. Similarly, a more
detailed study of the water injection rate inside a reciprocating air
compressor has been carried out by the iso-engine concept group
(e.g. Coney et al. [11]), by incorporating modern CFD codes and
advanced optical experimental techniques. Of course, all these
studies refer to compressions above the atmospheric conditions.

Far greater attention has been paid on the injection of water in
the compression process of Gas Turbine plants. The recent review by
Claus [12] covers most variations on the ideas that have been tested
so far. The application closest to the present study is the so called
“wet or foggy compression” [13]. Here the water not only cools the
incoming air just before the engine inlet, but it also carries along
“large” diameter droplets to be evaporated inside the compressor
blade channels. Unfortunately, inside a Gas Turbine the air pressure
is always well above the atmospheric one. As the pressure of the
(humid) air rises, this tends to saturate rather quickly. Under such
conditions perfectly isothermal compression is not possible for
temperatures close to the atmospheric. Zheng et al. [14] studied the
thermodynamic processes for the fogging cooling application in Gas
Turbines and concluded that for a compression ratio above 10:1, the
exit (humid) air temperature will exceed the 400 K level (starting
from the reference 288 K one) even in the limit where the (humid)
air is maintained at saturated conditions throughout the entire
compressor length. In addition, the momentum losses introduced
by the “large” diameter droplets introduce additional entropy,
so that the actual compression process may be represented by
a pv = constant relationship where n = 1.12, if the thermodynamic
properties of the atmospheric air are maintained constant to those
at T = 288 K. For the higher temperatures corresponding to the
constantly saturated mixture scenario this will be increased some-
what. Roumeliotis and Mathioudakis [15,16] have provided exper-
imental data for the single and multistage axial compressors under
foggy conditions. They concluded that the water evaporation affects
not only the total pressure losses but it modifies the performance
map of the compressor as well.

The present study positions the water injection on the opposite
side of the cycle, where the (flue) gas pressures are well below that
of the atmosphere (as a result of the need to expand the gases
sufficiently, so as to bring their temperature close to the atmo-
spheric). At those pressures the (water) steam specific internal
energy and enthalpy are essentially only functions of the tempera-
ture. This leads to an analytical expression of the (perfectly)
isothermal compression process in the well-known polynomial
formulation (pv = constant), where n is of order 1.06. This,
however, requires a regulated water mass flux. By employing this
formulation, the analysis proceeds to estimate the implications of
the water injection on the ideal Braysson cycle thermal efficiency.
Although this level of analysis does not provide accurate energy
transfer estimates, the calculated relative variation for the depen-
ded cycle parameters (efficiency, specific work, etc) against the free
ones (pressure ratio, maximum to minimum temperature ratio,
constant γ, etc) usually is fairly good. The results show that the work
consumed for the injection assisted compression is increased by a
relatively small magnitude over the corresponding process of the
ideal air. Next, the analysis establishes the parameter levels for
which such a compression may be implemented, i.e. the air
temperature, pressure and specific humidity values at the start of
the process. These levels are shown to be of the order of 330–340 K
for (isothermal) compression ratios of the order of 10:1, i.e.
comparable with the levels of the bottoming Rankine cycle steam
condensers. In the proposed Braysson plant the compressor exhaust
temperature constitutes the exhaust temperature of the topping
cycle flue gases, while in the conventional Brayton–Rankine
Combine cycle plants the corresponding temperature is of order
370 K. This implies that the proposed plant will deliver a slightly
better efficiency than the conventional plants. Of course, the great
advantage of the proposed cycle comes from the absence of any
Waste Heat Recovery Boiler. This not only will reduce the plant
construction costs but it will also permit a load acceleration capa-
bility similar to that of the single Gas Turbine (or Diesel Engine)
plants. In other words, the proposed plant will not need separate “transient load” supporting plants.

2. Some introductory aspects

2.1. The isothermal compression process

The isothermal process of compressing ideal gases implies that
T = constant, which in turn leads to pv = constant. The corre-
spending equation for the work transfer between states 1 (start)
and 2 (finish) is equal to

\[ \Delta W_{12} = MRT \ln \left( \frac{p_2}{p_1} \right) = MRT \ln \left( \frac{T_2}{T_1} \right) \]  

(1)

The parameters p, v, T refer to the pressure, specific volume and
temperature of the perfect gas, respectively, while M and R refer
to the mass and the gas constant, respectively.

2.2. The humid mixture and the saturation condition

If the humid mixture is assumed to be composed of perfect
gases, the saturation specific humidity (\( \omega \)) may be evaluated
through the application of the perfect gas law, which leads to the
following relationship

\[ \omega_S = \left( \frac{m_W}{m_{DG}} \right)_S = \frac{R_W}{R_{DG}} \frac{\varphi \rho_{SAT}(T)}{p - \varphi \rho_{SAT}(T)} \]  

(2)

Here \( \rho_{SAT}(T) \) is the saturation pressure of the water at the
temperature of the gaseous mixture (T), while \( \varphi \) is the relative
humidity. Water injection increases the specific humidity (\( \omega \)) as the
regulated compression process evolves. Limit condition is reached
when the mixture becomes saturated. Beyond this point no more
water is evaporated.

2.3. The ideal full expansion cycles

The two widely employed cycles of Otto and Brayton are con-
Sidered as references for their “full expansion” counterparts, as
illustrated in Fig. 1. Fig. 1a illustrates the Otto cycle in both the
standard version (defined by the sequence 01–02–03–04–01) and
the new, “full” expansion one, (the sequence 01–02–03–05–01).
The corresponding cycle structures for the Brayton are illustrated in
Fig. 1b. It can easily be shown that the thermal efficiency of the
ideal full expansion version for both cycles is given by

\[ \eta_{TH} = 1 - \frac{1}{\vartheta} \ln(X) \]  

\[ = 1 - \frac{f(X)}{\vartheta} \]  

(3)

where \( \vartheta = r^{\gamma-1} \) is the “compression” temperature ratio, \( \Theta = T_2/T_1 \) is
the ratio between the maximum and the minimum temperatures,
\( X = \Theta/\vartheta \), and \( f(X) = \ln(X)/(X - 1) \).
Given by corresponding values of the standard ones, which are equal and efficiencies of the full expansion cycles are always larger than the function \( h_x \)\( h \). As illustrated in Fig. 2, for atmospheric air (i.e. when \( \gamma = 1.4 \)) the function \( f(X) < 1 \) as long as \( X > 1 \). In other words, the thermal efficiencies of the full expansion cycles are always larger than the corresponding values of the standard ones, which are equal and given by

\[
\eta_{\text{OTTO}} = \eta_{\text{BRAYTON}} = 1 - \frac{1}{\delta}
\]

The ideal thermal efficiencies of the full expansion cycles may exceed the 80% levels, when the corresponding standard cycles reach efficiencies in the 60% level.

2.4. The water injection effects in the Braysson and Gas Turbine cycles

The cycle modifications imposed by the water injection in the relevant processes of the Braysson cycle and the typical Gas Turbine (Brayton) one are illustrated in Fig. 3. The basic Brayton cycle (i.e. Power Generating Gas Turbine) is defined by the thermodynamic state sequence 1\( \rightarrow \)2\( \rightarrow \)3\( \rightarrow \)4\( \rightarrow \)1. The Fog Compression process intercools the Compressor, so that the Gas Turbine cycle is now defined by the sequence 1\( \rightarrow \)2\( \rightarrow \)3\( \rightarrow \)4\( \rightarrow \)1. The Braysson cycle is defined by the processes 1\( \rightarrow \)2\( \rightarrow \)3\( \rightarrow \)4\( \rightarrow \)5\( \rightarrow \)1. The sequence 3\( \rightarrow \)4\( \rightarrow \)5 represents the expansion inside the turbine of the Gas Turbine (sequence 3\( \rightarrow \)4) as well as the additional expansion inside the “bottoming cycle” turbine (sequence 4\( \rightarrow \)5). It is apparent that the area of the Braysson cycle is much larger than that of the “intercooled” Gas Turbine.

The typical structure of a Power Plant operating on the Braysson cycle as modified by the water injection is illustrated in Fig. 4. Most of the plant components are widely employed in the conventional plants (the second turbine will not be much different from the corresponding units of a Steam Turbine plant). The new “component” is associated with the exhaust of the flue gases. The best solution is given by a conventional axial compressor assisted by water injection between its stages. This configuration allows for the entire expansion (turbine)–(isothermal) compression apparatus to be built around a common shaft, in a structure reminiscent of the low-pressure steam turbine stages. Given the very low pressures near the end of the expansion–start of the compression, axial turbomachinery provides the most appropriate (non-dimensional) “specific speed”. However, for large plants the high volumetric rates and the restricted shaft angular frequency (50–60 Hz) lead to a rather large axial velocity (a well-known problem in the low-pressure section of the Steam Turbines). This, in turn, leads into a very small diameter for the injected water droplets, if the droplet atomization–evaporation rates are to be implemented within the spacing separating two successive compressor stages. Such compression magnitudes and the rather large water mass flux imply a rather serious thermal efficiency reduction for the plant, of the order of 2–3% as discussed earlier. Of course, on top of this one should include the additional losses coming from the mixing interaction between the high-speed flue gases and the water. Until such an axial compressor is developed that would eliminate the above problem, it is proposed to repressurize the exhaust gases by the method illustrated in Fig. 4. This involves a chamber (EC) that will collect the turbine exhaust and distribute it among the numerous isothermal compressors (ISC). The structure of this section will be quite sizable but the low temperatures (of order 340 K) and pressures (of order 0.1–1 bar) involved could be managed by a construction based on cheap materials (cement for the chamber and plastics for the compressors). In addition, it avoids the solid matter concentration limits imposed for the water entering the hot section of the Gas Turbine (or Diesel Engine). The proposed plant could operate even on seawater. In the last scenario, the subsequent cooling of the exhaust (outside the compressors) by the atmospheric air (in a free convection Cooling Tower unit) could lead to a very cheap desalination plant.

![Fig. 1. The full expansion versions of the (a) Otto cycle and (b) the Brayton cycle.](image1)

![Fig. 2. The function \( f(X) \).](image2)

![Fig. 3. The modification imposed by water injection in the Braysson and the Gas Turbine cycles.](image3)
3. Isothermal compression in a closed system

As discussed above the isothermal compression process may be implemented in either an open system (i.e. axial compressor) or a closed system (positive displacement) unit. Since the second is easier to construct, the following analysis will be based on a closed thermodynamic system approach. Given that the specific internal energy of the slightly superheated-saturated water steam differs only 1–2% from the corresponding enthalpy at the sub-atmospheric pressures of interest here, it is quite straightforward to show that both thermodynamic system approaches give (almost) the same results. The closed system approach will incorporate inlet–exhaust processes in addition to the compression one. Now the regulated water injection rate (actually water evaporation one) cannot be implemented easily by employing a single nozzle injecting into a single cylinder, due to the time delay between the injection and the evaporation. In our laboratory such a compressor is under development employing a Vane Rotary concept, as illustrated in Fig. 5a. This combines the roles of both the turbine and the compressor. Water is injected at a given cavity just after the separating rib passes in front of the injection nozzle. The analysis follows one such cavity as it traverses the entire compressor arc. The humid unsaturated gaseous mixture is composed of dry (perfect) gases and water steam. The volume change inside the cavity is represented by the motion of a (fictionitious) piston (Fig. 5b). The upward motion of the piston compresses the gasses while the regulated water injection (followed by an instantaneous evaporation) maintains a constant temperature. The mass of the dry gases remains constant throughout the process, while the water mass (and the corresponding humidity) increases due to the injection.

The law of mass conservation gives the following relationship

$$dm = m_{DC}d\omega = dm_W$$

Here $m$, $m_{DC}$, and $m_W$ are, respectively, the masses of the entire mixture, the dry gasses and the water steam. The first law applied to this configuration gives

$$dU = dQ - pdV + dH_{in}$$

where $dU$ = the total internal energy change of the mixture, as the piston moves to reduce the volume by $dV$; $dQ$ = the heat entering the system through the walls or due to the friction caused by the motion of the piston; $p$ = the pressure of the mixture; and $dH_{in}$ = the enthalpy of the injected water (liquid) mass.

The internal energy for the mixture ($U$) is equal to the sum of the corresponding internal energies of the dry gasses and the evaporated steam. The internal energies for the dry gasses and the liquid water are assumed to be equal to zero at $T = 273.15$ K = 0 °C. Hence

$$U = m_{DC}c_{DC}(T - 273.15) + m_Wu_{SHS}$$

Fig. 4. The typical structure of a Braysson cycle incorporating water injection for the isothermal compression process.

Fig. 5. A typical isothermal compressor configuration and the simplified control volume. (a) The expander–compressor concept studied in our laboratory. (b) The simplified control volume geometry.
where \( \omega_{DG} \) is the (average) constant volume specific heat of the dry gases (kJ/kgK); \( T \) is the mixture temperature (K); \( \omega_{SHS} \) is the specific internal energy of the superheated (or saturated) steam contained within the humid mixture (kJ/kg); \( dH_{in} = h_W dm_W = m_{DC} c_{SHS} \omega' \); \( h_W \) is the specific enthalpy of the injected water (kJ/kg).

As illustrated in Fig. 6 which represents the corresponding region from the Mollier diagram, the specific internal energy of the superheated steam \( \omega_{SHS} \) is a function only of the temperature, since it is practically independent of the pressure at levels around or below the atmospheric one. This amounts for the isothermal lines to be (practically) parallel to the entropy axis right off the saturation line. On the other hand, at high pressures and low temperatures (i.e. the practical point of view this means that the mixture may reach the saturation condition during the entire compression process. From the combination of Eqs. (15) and (16) leads to the following polytropic relationship for the isothermal compression through regulated water injection

\[
\frac{V_1 p_1}{V_5 p_5} = \frac{m_{DC} R_{DC} \left( 1 + \frac{R_W}{R_{DG}} \omega' \right) T}{m_{DC} R_{DC} \left( 1 + \frac{R_W}{R_{DG}} \omega_5 \right) T} = \frac{\omega'_1}{\omega_5}
\]

(16)

The above equation holds for any (perfect gas mixture of) flue gases. It provides the rate of the injection that sustains the isothermal compression. By assuming that the mixture is always composed of perfect gases, it is quite straightforward to relate the humidity ratio to the corresponding pressure one, since

\[
\omega'_1 = \frac{1 - \frac{\phi p_{SAT}(T)}{p}}{1 - \frac{p_{SAT}(T)}{p}}
\]

(19)

The corresponding modified (humidity) parameter becomes

\[
\omega'_1 SAT = \omega'_1 \left( T \right) = 1 + \frac{p_{SAT}(T)}{p - p_{SAT}(T)} = \frac{1}{1 - \frac{p_{SAT}(T)}{p}}
\]

(20)

In combination with Eqs. (15) and (18), this leads to the following relationship

\[
\frac{p}{(\omega'_1)^{1+\omega'_1}} = \frac{p}{(\omega'_1)^{b_1/(b_1-1)}} = \text{const.}
\]

(21)

By referring to Fig. 3, Eqs. (20) and (21) define the limiting starting conditions (i.e. pressure \( p_5 \) and modified humidity \( \omega'_1 \)) of the compression for which the isothermal compression (through regulated water injection) will terminate (at point 1) with...
The temperature during the compression process will be increased by a saturated gaseous mixture. In other words, if \( p_5 \) is fixed as well, the modified humidity \( \omega_s' \) should not exceed a certain value. The relationship \( (\omega_s')_{\text{LIMIT}} = f(p_5, T) \) is illustrated in Fig. 8 for the case \( p_1 = 1 \) bar. Mathematically, this is given by the equation

\[
(\omega_s')_{\text{LIMIT}} = \omega_s'_{\text{SAT}} \left( \frac{p_5}{p_1} \right)^{a_{V} + 1} = \frac{1}{1 - \frac{p_{\text{SAT}(T_5)}}{p_5}} \left( \frac{p_5}{p_1} \right)^{a_{V} + 1}
\]

In practice, due to H_2SO_4 condensation considerations the temperature during the compression process will be increased by 5–10 K (for the case of a conventional Power Plant based on a Gas Turbine Topping cycle Plant). For the typical flue gas composition of the Natural Gas burning Gas Turbines (mole fractions of the order of 0.06 for the water and 10^{-3} for the SO_3), the Okkes [17] equation predicts dew point temperatures of the order of 70\( \pm \)K (i.e. 343 K), when the expansion reaches the 0.1 bar level. This implies that the low-pressure (bottoming) turbine is not expected to face any serious sulfuric acid corrosion problem. The isothermal compressor, however, must be designed appropriately. Of course, for Topping cycle machines operating on a smaller \( T_3 \) these limitations will not apply.

The efficiency evaluation of the proposed Braysson plant starts with the calculation of the “coefficient of performance” \( (\text{cop}_V) \) for the isothermal compressors in the exhaust system. This parameter compares the work consumed for the entire exhaust process (volume charging–isothermal compression–discharging to the atmosphere) by employing water injection \( (W_{V(\text{inj})}) \) over the corresponding work when the isothermal compression does not employ water cooling \( (W_{\text{DRY}}) \). The first work is given by

\[
\text{cop}_V = \frac{\Delta W_{\text{V,inj}}}{\Delta W_{\text{DRY}}} = \frac{Y - 1}{\ln(Y)}
\]

The \( \text{cop}_V \) represents the work rise due to the injection (actually is defined as the ratio of the compression work in the presence of the injection over that of the isothermal compression of the dry gasses alone). Now, the work done for the wet mixture compression is given by

\[
W_{V(\text{inj})} = p_{5}V_{5} - \left( \int \frac{V_1}{V_5} \rho_d \, dv \right) - p_{1}V_{1}
= \left( \omega_s' - \omega_1' \right) MRT_5 \left( \frac{V_1}{V_5} \right) \left( \frac{V_{5}}{V_5} \right)^{-1}
= \left( \omega_s' - \omega_1' \right) MRT_5 \left( \frac{V_1}{V_5} \right) \left( \frac{V_{5}}{V_5} \right)^{-1}
\]

In the above equation \( r = V_5/V_1 \) and \( m_{R,T} = \omega MRT \), where \( M \) represents the mass of the dry flue gases while \( R \) their gas constant. The work transfer \( (W_{\text{DRY}}) \) is calculated the same way, except for the fact that the isothermal compression of the constant composition mixture obeys the \( pv = \) constant rule. Of course the second process reaches the same exit pressure \( (p_1) \) but as a result the final volume will be smaller than \( V_1 \). It is a straightforward process to prove that the ratio between these two work transfers is equal to

\[
\text{cop}_V = \frac{W_{V(\text{inj})}}{W_{\text{DRY}}} = \frac{Y - 1}{\ln(Y)}
\]
where $Y = r^{b - 1}$. The relationship $\text{cop}_V(Y)$ is illustrated in Fig. 9. It is quite apparent that the $\text{cop}_V$ is dominated by the required compression ratio. The influence of the temperature during the compression plays a much smaller role. At a given temperature there exists a (low) limit for the starting pressure ($p_s$). Below it the humid gas mixture saturates, hence no water evaporation may take place. The water injection increases the work consumed for the exhaust process by a very small amount (les than 4%) over that of the “dry” isothermal process.

The efficiency of the entire Braysson plant may be evaluated by the equation

$$\eta_{\text{BRS}} = \frac{\text{NET WORK}}{\text{HEAT INPUT}} = \frac{W_{35} - W_{12} - W_{VS1}}{Q_{12}}$$

$$= \frac{\omega^*_V \text{MRT}_1 \left( \frac{X}{C_0} (\theta - \theta) - \text{cop}_V \ln(p_1/p_5) \right)}{\omega^*_V \text{MRT}_1 \left( \frac{X}{C_0} (\theta - \theta) \right)}$$

(26)

The parameters $\theta$ and $\phi$ were defined in Section 2.3. By employing the parameter $X = \theta/\phi$ (as in Eq. (3)), it is quite simple to transform Eq. (26) into

$$\eta_{\text{BRS}} = 1 - \frac{1}{\theta} \ln \left( \frac{X \text{cop}_V}{\theta} \right) = 1 - \frac{f_1(X)}{\theta}$$

(27)

Since

$$\frac{p_1}{p_5} = \frac{p_1 p_2 p_3}{p_2 p_3 p_4} = \left( \frac{\theta}{\phi} \right)^{\frac{1}{b_V}}$$

(28)

the pressure ratio $p_1/p_5$ becomes a function of the one of the (free) $\theta$ parameters, if the other is fixed. Usually it is $\phi$ that becomes fixed, as a result of the technological limits imposed by the cooling of the blades. For typical Gas Turbine compression ratios of the order of 30:1 (i.e. $\theta = 2.65$) the corresponding $p_5/p_1 = 0.06$ as long as $\theta = 6$. Most conventional power generating Gas Turbines operate with a high temperature around 1550 K (i.e. $\theta = 5.35$), so that a minimum pressure ($p_5$) around the 0.1 bar is the expected operational limit.

The variation of the plant (ideal) efficiency is illustrated in Fig. 10, where $\theta = 6$. In addition to the efficiency of the proposed Braysson cycle version, this figure includes the corresponding efficiencies of the conventional Braysson cycle and the Brayton cycle with the same $\theta$ value (i.e. Gas Turbine compression ratio). Clearly, the modified cycle efficiency is very close to that of the conventional Braysson one and much higher than the simple Brayston cycle.

4. Conclusions

The ideal Braysson cycle was estimated for the case of employing regulated water injection rate for the isothermal compression process of the cycle. For the expected parameter range in typical applications, the analysis lead to a polytropic relationship for the process of the form $p v^\beta$ = constant, where $\beta \approx 1.06$. The analysis predicts a very small reduction in the thermal efficiency of the cycle against that of the conventional Braysson cycle and much larger than the corresponding Brayton (gas turbine) one.

Nomenclature

- $a_V$: constant (Eq. (14))
- $b_V$: constant (Eq. (18))
- $c_p$: specific heat (at constant pressure)
- $c_V$: specific heat (at constant volume)
- $\text{cop}_V$: coefficient of performance (Eq. (23))
- $h$: enthalpy
- $h$: specific enthalpy
- $p$: pressure
- $q$: heat transfer
- $r$: compression ratio
- $R$: gas constant
- $s$: entropy
- $T$: temperature
- $U$: internal energy
- $u$: specific internal energy
- $V$: volume
- $W$: work transfer
- $X$: parameter (Eq. (5))
- $Y$: parameter (Eq. (25))

Greek

- $\gamma$: the ratio $c_p/c_V$
- $\eta$: efficiency
- $\theta$: temperature ratio in the compression (Eq. (5))
- $\phi$: temperature ratio (Eq. (5))
- $\varphi$: relative humidity (Eq. (12b))
- $\nu$: specific humidity
- $\nu'$: modified specific humidity (Eq. (12a))

Subscripts

- 1, 2 endpoints of the compression process
- DG: dry gas mixture
- fg: evaporation
- SAT: saturation
- SHS: superheated
- TH: thermal
- W: water

References


