Air turbine – an interesting solution for straw energy conversion into electricity

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Abstract. Straw is a non-hazardous by-product of crop plants processing. Currently, it represent one of the most important biomass resource. The huge quantities of straw annually produced generate big problems in what concerns their disposal. The traditional field burning is no longer accepted, so another disposal solutions must be found and recycling is the most attractive. The paper refers to such a solution consisting in the conversion of the straw energy potential into electricity in a power plant based on an air turbine. This power system it is in fact an external combustion engine, derived from a gas turbine engine and operating with air as working fluid instead of combustion gases. In order to make possible the use of straw as fuel, the conventional combustion chamber is substituted by a hot air generator. Schematic of this power system and the results of its energetic analysis are presented in the paper. There are analysed the main performance indicators, namely thermal efficiency, output power, fuel consumption and specific fuel consumption. The results of the study indicate the analysed power system as an interesting solution for straw recycling.

1. Introduction

As a result of the production of crop plants (such as barley, rice, wheat or rape), straw represents today one of the most important biomass resource. According to the directive 2008/98/EC, straw is a non-hazardous agricultural material which do not harms the environment or endangers human health. Even so, straw disposal remains a big problem since huge quantities of straw are annually produced [1], [2]. As long the traditional and cheapest disposal solution, consisting in field burning, is no longer acceptable, recycling of the straw is the challenge in the current approach. As consequence, there are developed several studies regarding straw energy recovery, by pyrolysis [3], gasification [4], hydrolysis, fermentation [5] or direct combustion (fixed or fluidized bed combustion [6]), for heat and/or power (co)generation [7], [8]. If only heat is required the most convenient current solution is a hot water boiler [9], but this is not a typical case. Usually, there is no heat demand and electric power is required; the most common solution in this case is a steam power plant [10].

In order to answer to the straw problem, power plants covering an output range of 1 to 5 MW are required in Romania in the current context. A facile change of plant location, with minimum effort, beside independency by the water sources would be great features of the plant because offer possibility of its optimum placement relative to the straw provider. A steam power plant does not match these requirements, so it is not the best option in this case; a very interesting solution it is a hot air turbine power plant (HATPP) [11]. According to this idea, an analysis of a HATPP from performance point of view is performed in the present paper.
2. Description of the Hot Air Turbine Power Plant configuration
As can be seen in figure 1, the analysed HATPP, operating in opened cycle, is a gas turbine engine (GTE) converted to operate with straw (solid fuel) instead of kerosene or natural gas, as usual configurations. Obviously, in this case the combustion gases cannot directly drive a turbine, so a compressed air heater (CAH) must be used instead of the conventional combustion chamber. Thus, compressed hot air is the working fluid instead of the combustion gases, which means the engine becomes an external combustion one.

There are considered two turbines in the scheme: one drives the compressor (the High Pressure Turbine) and the second drives the Electrical Generator via a Reduction Gear. Configuration of CAH, consisting in two heat exchangers (a parallel-flow heat exchanger, HE1, and a counter-flow heat exchanger, HE2) was decided by two considerations:

- Maximum HPT Inlet Temperature (TIT), since the higher TIT the higher is the efficiency of the turbine engine;
- Minimum CAH flue gas temperature (t_{fg}), since the lower t_{fg} the higher is the CAH efficiency.

3. Energetic analysis
In order to analyze performances of HATPP a dedicated code named AIRTURB1 was developed. Calculations were performed by considering the following chemical composition for fuel (straw), expressed in mass percent [12]: C = 45%; H = 5.58%; O = 38.75%; N = 0.54%; S = 0.09%; ash – A = 0.04%; water – W = 10%. Power output, thermal efficiency, fuel consumption and specific fuel consumption are the main calculated parameters since they were assumed as the most significant performance indicators.

All calculations were made assuming the following constant values:

- Ambient parameters (pressure and temperature): \( p_a = 1.013 \text{ bar} ; t_a = 15°C \)
- Air mass flow rate: \( \dot{m}_a = 6.5 \text{ kg/s} \)
- Compression efficiency: \( \eta_{ic} = 85\% \)
- HPT inlet temperature: \( \text{TIT} = 900°C \)
- Temperature difference on CAH exit (figure 1): \( \Delta t = t_{fg} - t_{ina} = 30 \text{ deg} \)
- Turbines mechanical efficiencies: \( \eta_m = 99.5\% \)
• Turbines isentropic efficiencies: \( \eta_{it} = 85\% \)
• Compressor inlet pressure drop coefficient: \( \varphi_{ad} = 0.99 \)
• CAH pressure drop coefficient for compressed air: \( \nu_{K} = 0.95 \)
• CAH heat loss by incomplete combustion: \( q_3 = 2\% \)
• CAH heat loss owing to unburned carbon: \( q_4 = 3\% \)
• CAH heat loss owing to convection and radiation: \( q_5 = 2.65\% \)
• Excess air coefficient at the exit of CAH: \( \lambda_{CAH} = 1.2 \)

Knowing the chemical composition, Lower Heating Value of straw can be calculated with formula (Mendeleev method)

\[
LHV = 4.1868 \cdot \left[81C + 246H - 26(O - S) - 6W \right] \quad [kJ / kg]
\]

and results \( LHV = 16548 \text{ kJ/kg} \).

Entire output power of HATPP, generated by LPT (HPT doesn’t generates net power, just drives the compressor) is expressed as

\[
P_{LPT} = 10^2 \cdot \eta_{m} \cdot \dot{m}_{a} \cdot (i_{LPT_{in}} - i_{LPT_{out}}) \quad [kW],
\]

where \( i_{LPT_{in}} \) and \( i_{LPT_{out}} \) are the enthalpies of air on LPT inlet and LPT exhaust, respectively, in kJ/kg. The enthalpies of air are calculated by considering linear dependence of the specific heat at constant pressure by absolute temperature [13];

\[
i = c_p T = \left(0.9702 - 9.299 \cdot 10^{-5} T \right) T \quad [kJ / kg].
\]

The heat input of HATPP is, in fact, the heat input of CAH, which is calculated with formula

\[
Q_{in} = 100 \cdot \dot{m}_{a} \cdot (i_{out_{a}} - i_{ina}) \cdot \eta_{CAH}^{-1} \quad [kW],
\]

where:
- \( i_{out_{a}}, i_{ina} \) - CAH air inlet / outlet enthalpy [kJ/kg];
- \( i_{TII} \) - HPT inlet enthalpy [kJ/kg];
- \( \eta_{CAH} \) - CAH efficiency [%].

As can be seen in figure 1, \( T_{IT} = t_{out_{a}} \). Consequently, \( i_{TII} = i_{out_{a}} \) and formula (4) becomes

\[
Q_{in} = 100 \cdot \dot{m}_{a} \cdot (i_{TII} - i_{ina}) \cdot \eta_{CAH}^{-1} \quad [kW].
\]

CAH efficiency is expressed as

\[
\eta_{CAH} = 100 - (q_2 + q_3 + q_4 + q_5 + q_6) \quad [%],
\]

where \( q_2 \) and \( q_6 \) are the heat loss through stack gas and the heat loss through the sensible heat of ash / slag, respectively. They are calculated with formulas

\[
q_2 = \left(100 - q_4\right) \cdot \left(I_{fl} - \lambda_{CAH} \cdot I_{inha}\right) \cdot LHV^{-1} \quad [%],
\]

\[
q_6 = \left(1 - a_f\right) \cdot A \cdot I_{ash} \cdot LHV^{-1} \quad [%],
\]

where:
- \( I_{fl}, I_{inha} \) - enthalpies of flue gas, ambient air and ash, respectively [kJ/kg fuel];
- \( a_f \) - fly ash fraction.

Enthalpies of flue gas, ambient air and ash are calculated by considering five-degree polynomial temperature dependence of the specific heat [12].

Once \( Q_{in} \) and \( \eta_{CAH} \) known, thermal efficiency, fuel consumption and specific fuel consumption of HATPP can be estimated with formulas

\[
\eta_t = 100 \cdot P_{LPT} \cdot Q_{in}^{-1} \quad [%],
\]
4. Results of the analysis

The higher TIT, the higher efficiency of GTE and the higher the capital cost are. Currently, the advanced materials for high temperature applications operate up to 1000°C and even beyond [14]. As mentioned above, the value of TIT is assumed 900°C, which complies with these materials.

It is known that for a given value of TIT it is an optimum compression ratio ($\pi_c$) for maximum efficiency. As fig. 2a shows, HATPP optimum compression ratio is $\pi_c = 12$, when $\eta_{\text{HATPP}}$ is maximum, namely 24.7%. Obviously, minimum SFC$_{\text{HATPP}}$ (see figure 2c), namely 0.88 kg/kWh, comes together with maximum $\eta_{\text{HATPP}}$, reached in optimum conditions. As figure 2d shows, output power is $P_{\text{LPT}} = 1338$ kW while fuel consumption is $F_{\text{CHATPP}} = 1177$ kg/h in these conditions. Maximum output power, of 1375 kW, could be achieved if $\pi_c = 8$ but $\eta_{\text{HATPP}}$ is lower in this case, namely 23.7%.

In figure 2a it is also represented the variation curve $\eta_{\text{GTE}} = f(\pi_c)$, where $\eta_{\text{GTE}}$ is the efficiency of a conventional GTE operating in similar conditions, with the same TIT. A much higher efficiency level stands out in this case on the entire range of $\pi_c$; the maximum value $\eta_{\text{GTE}} = 33.8\%$ is reached when compression ratio is $\pi_c = 16$, which is the optimum value for GTE.

![Figure 2](image_url)

**Figure 2.** Variation of HATPP performance indicators with compression ratio.
Lower efficiencies of HATTP could be explained by analysing the variation curves of $\eta_{\text{CAH}}$ and $t_{fg}$ with $\pi_c$ (see figure 2b): the higher $\pi_c$ the higher $t_{fg}$ because compressed air temperature ($t_{\text{lna}}$, see figure 1) is higher. Thus, $q_2$ is higher and consequently $\eta_{\text{CAH}}$ is lower. In optimum conditions (when $\pi_c = 12$), $t_{fg} = 392^\circ\text{C}$ and $\eta_{\text{CAH}} = 73.6\%$ – significantly lower than efficiency of a conventional GTE combustion chamber, which was assumed 98% in the study. Not only $t_{fg}$ but also LPT exhaust air temperature is high, namely 396°C. This implies a significant heat quantity released into the atmosphere. A higher efficiency of the system could be achieved only by recovering this energy, which is possible if there is heat or / and cooling demand beside power requirement. The installation would become a cogenerative or trigenerative unit in this case.

5. Conclusions

Compared with a conventional GTE operating with the same TIT (900°C), in similar conditions, the analysed HATTP has a different optimum compression ratio, namely $\pi_c = 12$ (optimum compression ratio of GTE is $\pi_c = 16$).

HATTP is less performant than a conventional similar GTE: maximum efficiency of HATTP (when $\pi_c = 12$) is $\eta_{\text{HATPP}} = 24.7\%$, while maximum efficiency of GTE (when $\pi_c = 16$) is $\eta_{\text{GTE}} = 33.8\%$. The other performance indicators of HATTP are: SFC$_{\text{HATPP}} = 0.88 \text{ kg/kWh}$ (minimum value); $P_{\text{LPT}} = 1338 \text{ kW}$; FC$_{\text{HATPP}} = 1177 \text{ kg/h}$.

Lower performance of HATTP is caused by the substitution of the conventional combustion chamber of GTE, highly efficient, with CAH – a less efficient combustor with external combustion. But taking into account that analysed power system operates with bio-waste and not with fossil fuel (liquid or gaseous), as conventional GTE does, this drawback is a minor one. In this approach, HATPP represents a very interesting solution for straw energy conversion into electricity, especially when the change of the plant location and / or independency by the water sources are imposed.

Higher efficiency of the installation could be achieved by converting HATTP into a cogenerative / trigenerative unit. Obviously, this solution can be taken into consideration only if there is a heat or / and cooling demand.

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