Dimensional approach on hot air turbine power plant in opened cycle for straw recycling

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Abstract. Currently, disposal of straw is one of the biggest problems that crop plant producers are facing. The ideal case implies not only to get rid of straw but also to recover its energetic potential. In this context, the performance of a hot air turbine power plant operating in open cycle, with straw as fuel, was analyzed in a previous study and proved to be a very interesting solution for straw disposal. As consequence, dimensional analysis of the hot air turbine power plant is required into the next step and this makes the subject of the present study. The dimensional analysis is focused on the compressed air heater - the largest component of the Power Plant, with crucial role in what concerns its entire size and mass. Once both performance and dimensional analysis performed, the final conclusions are drawn in an overall approach, by taking also into consideration the economic aspects.

1. Introduction

Straw represents a byproduct of crop plants harvesting process. Currently, straw is one of the most significant biomass material [1] and is assumed to be environmentally harmless. In spite of this fact, straw still represents a big problem, which refers to disposal. By trying to find proper solutions to this problem, a hot air turbine power plant (HATPP) configuration, operating in open cycle, was analyzed in a previous study from performance point of view [2]. The results of this study indicate the analyzed configuration as a very interesting solution for straw disposal. In these terms, the dimensional analysis of HATPP is mandatory next. Since dimensions of HATPP are mainly decided by the compressed air heater (CAH), the analysis was focused on CAH size and mass.

2. Calculation method

Configuration of HATPP [2] corresponds to an external combustion power system and can be assumed as a Gas Turbine Engine having the conventional combustion chamber replaced by CAH; consequently, air is the working fluid for turbines instead of combustion gases (will be referred further as “gases”). This is because of the solid fuel (straw), which excludes the possibility of an internal combustion system configuration.

CAH consists in two heat exchangers in series (see figure 1a) – one parallel-flow (HE1) and one counter-flow (HE2). This configuration was established by aiming to get maximum HATPP efficiency. Thus, compressed air with lowest temperature (compressor discharge temperature) enters CAH in the flue gases section and flows counter-current with gases (HE2); this minimizes the flue gases temperature, which means it maximizes of CAH efficiency. Besides, the compressed air exiting HE2, with intermediary temperature (t_ina), enters CAH / HE1 in the hottest gases section and flows co-
current with gases; this ensures maximum tubes protection against the high temperature of gases making possible the highest temperature of air delivered by CAH (that is the Turbine Inlet Temperature – TIT) and, consequently, maximum HATPP efficiency.

In order to ensure maximum tubes protection, the value of \( t_{\text{inta}} \) results from the following condition for minimum temperature difference: \( \Delta t_{\text{inta}} = t_{\text{inta}} - t_{\text{outa}} = 50^\circ\text{C} \). In order to comply with the advanced alloys used in high temperature applications [3], it was assumed \( \text{TIT} = t_{\text{outa}} = 900^\circ\text{C} \).

\[ \text{HE1} – \text{First Gases-Air Heat Exchanger} \]
\[ \text{HE2} – \text{Second Gases-Air Heat Exchanger} \]

\[ \text{Figure 1. Configuration (a) and matrix (b) of CAH heat exchangers.} \]

The geometry of the two heat exchangers of CAH is presented in figure 1b. The basic matrix has the following characteristics:

- Specific surface \( C_v = 21.81 \text{ m}^2/\text{m}^3 \);
- Relative mass \( m_R = 31.2 \text{ kg/m}^2 \);
- Transverse relative pitch \( \sigma_t = s_t/d_c = 2.73 \);
- Longitudinal relative pitch \( \sigma_l = s_l/d_c = 1.09 \).

The aim of the study was to analyse variation of CAH / HATPP dimensional parameters with compression ratio \( \pi_c \), in the same conditions as the previous study referring to performance [2], by trying to get an overall view over the analysed power system. Since output power is varying with \( \pi_c \), only specific parameters (relative to kW of output power) could be relevant in what concerns CAH / HATPP dimensions. In this context, specific surface, specific volume and specific mass came as the most significant parameters. Their calculation procedure, described below, firstly requires calculation of the heat transfer rate in HE1 and HE2, which are expressed as

\[ Q_{\text{HE1}} = 3600^{-1} \cdot 10^{-2} \cdot \eta_{\text{he}} \cdot FC_{\text{HATPP}} \cdot (I_f - I_{\text{inag}}) \quad [\text{kW}] , \]

\[ Q_{\text{HE2}} = 3600^{-1} \cdot 10^{-2} \cdot \eta_{\text{he}} \cdot FC_{\text{HATPP}} \cdot (I_{\text{inag}} - I_{fg}) \quad [\text{kW}] , \]

where: \( \eta_{\text{he}} \) - heat exchanger efficiency; \( \eta_{\text{he}} = 95\% \);
\( FC_{\text{HATPP}} \) - fuel consumption of HATPP [kg/h];
\( I_f, I_{\text{inag}}, I_{fg} \) - enthalpy of gases into the flame, enthalpy of gases between the two heat exchangers and enthalpy of flue gases, respectively.

In order to calculate the enthalpies of gases, specific heat at constant pressure had to be determined. In this case, a polynomial temperature dependence of five degree was considered [3].

The flame temperature \( t_f \) results from the combustion process calculation [4], [5] while the temperature of gases between HE1 and HE2 is...
The flue gases temperature is given by

\[ t_{\text{fg}} = t_{\text{ina}} + \Delta t \quad [\text{°C}] \],

where \( t_{\text{ina}} \) is the temperature of air delivered by compression and is an output data of the compression process calculation; \( \Delta t \) is the final CAH temperature difference, assumed to be 30°C.

Applying the thermal balance equation to HE1 it results the intermediary enthalpy of air

\[ i_{\text{inta}} = i_{\text{outa}} - Q_{\text{HE1}} \cdot \dot{m}_a^{-1} \quad [\text{kJ} / \text{kg}] \]

Knowing \( i_{\text{inta}} \), the intermediary temperature of air \( t_{\text{inta}} \) can be determined. It must be mentioned that was considered linear temperature dependence for the specific heat at constant pressure [2].

Since all temperatures and flow rates of the two fluids changing heat (gases and air) are known, the heat exchange surfaces of HE1 and HE2 can be calculated. It was used the classic method [4], [6], based on the known formula

\[ S_{\text{HE1,2}} = 1000 Q_{\text{HE1,2}} \left( k_{1,2} \Delta t_{\text{lg1,2}} \right)^{-1} \quad [\text{m}^2] \],

where: \( k_{1,2} \) - heat transfer coefficients in HE1 and HE2, respectively \([\text{W/m}^2 \text{K}]\);
\( \Delta t_{\text{lg1,2}} \) - mean temperature difference in HE1 and HE2, respectively; it is expressed as

\[ \Delta t_{\text{lg1}} = \frac{(t_f - t_{\text{ina}}) - \Delta t_{\text{out}}}{\ln \left( \frac{t_f - t_{\text{ina}}}{\Delta t_{\text{out}}} \right)} \quad [\text{°C}] \],

\[ \Delta t_{\text{lg2}} = \frac{(t_{\text{in}} - t_{\text{ina}}) - \Delta t}{\ln \left( \frac{t_{\text{in}} - t_{\text{ina}}}{\Delta t} \right)} \quad [\text{°C}] \].

Heat transfer coefficients are calculated as

\[ k_{1,2} = \frac{\alpha_{g1,2}}{1 + (\varepsilon + \alpha_{a1,2}) \cdot \alpha_{g1,2}} \quad [\text{W/(m}^2 \text{K}]] \],

where: \( \alpha_{g1,2}, \alpha_{a1,2} \) - heat transfer coefficients of gases and air \([\text{W/(m}^2 \text{K}])\);
\( \varepsilon \) - ash deposit coefficient; it is assumed \( \varepsilon = 0.065 \text{ m}^2 \text{K/W} \) [5].

By using the general notation “\( \alpha \)” for heat transfer coefficients of gases and air, these parameters are given by

\[ \alpha = \omega (\alpha_g + \alpha_r) \quad [\text{W/(m}^2 \text{K}]] \],

where: \( \omega \) - factor for nonuniform sweeping of the gas; \( \omega = 1 \) [4];
\( \alpha_g, \alpha_r \) - convective / radiative heat transfer coefficient \([\text{W/(m}^2 \text{K}])\).

For both gases and air the radiative heat transfer coefficient is calculated as [4], [5]

\[ \alpha_r = 2.835 \cdot 10^{-8} \cdot a \cdot (a_{ad} + 1) \cdot T^3 \cdot \frac{1 - (T_{\text{aw}} \cdot T^{-1})^4}{1 - (T_{\text{aw}} \cdot T^{-1})^4} \quad [\text{W/(m}^2 \text{K}]] \],

where: \( a \) - emissivity of gases;
\( a_{ad} \) - emissivity of the external surface of tubes with ash deposits;
Convective heat transfer coefficients of gases and air are calculated as [4], [5]

\[
\alpha_{\text{g}} = \frac{\lambda_{\text{g}} \cdot d_{e}^{-1} \cdot \Re_{\text{g}}^{0.6} \cdot \Pr_{\text{g}}^{0.33}}{C_{g1} \cdot C_{g2}} \left[ \frac{W}{(m^2 \cdot K)} \right],
\]

(12)

\[
\alpha_{\text{a}} = 0.023 \cdot \lambda_{\text{a}} \cdot d_{e}^{-1} \cdot \Re_{\text{a}}^{0.8} \cdot \Pr_{\text{a}}^{0.4} \cdot C_{a1} \cdot C_{a2} \cdot C_{a3} \left[ \frac{W}{(m^2 \cdot K)} \right].
\]

(13)

In formulas (12) and (13) the following notations are used:

- \( \lambda_{\text{g}}, \lambda_{\text{a}} \) - thermal conductivity of gases and air [W/(m K)];
- \( \Re_{\text{g}}, \Re_{\text{a}} \) - Reynolds number for gases and air;
- \( \Pr_{\text{g}}, \Pr_{\text{a}} \) - Prandtl number for gases and air;
- \( C_{g1} \) - correction factor taking into account the numbers of rows of tubes;
- \( C_{g2} \) - correction factor taking into account the geometry of the bank of tubes;
- \( C_{a1} \) - correction factor taking into account the shape of the flow passage;
- \( C_{a2} \) - correction factor for the temperature difference between gases and tube wall;
- \( C_{a3} \) - correction factor taking into account the relative tube length.

Once \( S_{\text{HE1}} \) and \( S_{\text{HE2}} \) known, specific heat transfer surface of HATTTP can be calculated as

\[
s_{sp} = S \cdot P_{\text{LPT}}^{1} = (S_{\text{HE1}} + S_{\text{HE2}}) \cdot P_{\text{LPT}}^{1} \left[ \frac{m^2}{kW} \right],
\]

(14)

where \( P_{\text{LPT}} \), in kW, is HATTTP output, generated by the Low Power Turbine; it is defined in [2].

The other two relevant dimensional parameters of HATTTP, namely specific volume and specific mass, can now be expressed as

\[
v_{sp} = s_{sp} \cdot C_{v}^{-1} \left[ \frac{m^3}{kW} \right],
\]

(15)

\[
m_{sp} = s_{sp} \cdot m_{g} \left[ \frac{kg}{kW} \right].
\]

(16)

As mentioned above, the variable parameter in the present study is \( \pi_{c} \). The analysis was performed with a dedicated code based on the calculation procedure described above.

### 3. Results of the analysis and discussions

Beside variations of \( s_{sp}, v_{sp} \) and \( m_{sp} \) with \( \pi_{c} \), variations of heat transfer rate and heat transfer surface there were also represented in figure 2. It can be seen that \( Q_{\text{HE1}} > Q_{\text{HE2}} \) on the entire range of \( \pi_{c} \) (see figure 2a) but \( S_{\text{HE1}} > S_{\text{HE2}} \) (see figure 2b). This is mainly because temperature of gases in HE1 is much higher than temperature of gases in HE2, which makes \( \Delta t_{lg1} >> \Delta t_{lg2} \) – see formula (6).

As can be seen in figure 2c, minimum values of the specific dimensional parameters are reached when \( \pi_{c} = 16 \) (Case 1), which is the optimum case in what concerns CAH / HATTTP dimensions. All the dimensional parameters (specific and absolute) for this situation are presented in table 1, beside \( P_{\text{LPT}} \) and thermal efficiency of HATTPP (\( \eta_{\text{HATTPP}} \)) [2]. There are also specified the two mases (\( m_{\text{HE1}} \) and \( m_{\text{HE2}} \)) of the two heat exchangers, which are calculated taking into account the ratio \( S_{\text{HE1}}/S_{\text{HE2}} \). In addition, all the parameters mentioned above there are presented too for \( \pi_{c} = 12 \) (Case 2), which is the optimum case from performances point of view [2].

It can be seen that Case 1 requires the additional masses of 237 kg for HE1 and 963 kg for HE2 if compare with Case 2 but implies 104 kW additional power and 0.4% efficiency gain. Assuming an efficiency of 70% for \( P_{\text{LPT}} \) conversion into electric power (both internal energy consumption and efficiency of the electric generator are included here) it means that the additional electric power is 0.7 \( \times 104 \) kW = 72.8 kW = 0.0728 MW.
Figure 2. Variation of characteristic parameters of HATPP with compression ratio.

Table 1. HATPP characteristic parameters in Case 1 and Case 2.

| Unit          | $\pi_c = 12$ (case 1) | $\pi_c = 16$ (case 2) |
|---------------|-----------------------|-----------------------|
| $s_{sp}$      | m$^2$/kW              | 0.277                 | 0.269                 |
| $v_{sp}$      | m$^3$/kW              | $12.7 \times 10^{-3}$ | $12.34 \times 10^{-3}$ |
| $m_{sp}$      | kg/kW                 | 8.64                  | 8.39                  |
| $Q_{HE1}$     | kW                    | 2379                  | 2230                  |
| $Q_{HE2}$     | kW                    | 1639                  | 1391                  |
| $S_{HE1}$     | m$^2$                 | 126.7                 | 119.1                 |
| $S_{HE2}$     | m$^2$                 | 243.8                 | 212.9                 |
| $S = S_{HE1} + S_{HE2}$ | m$^2$ | 370.5                 | 332                   |
| $V$           | m$^3$                 | 17                    | 15.22                 |
| $m_{HE1}$     | kg                    | 3953                  | 3716                  |
| $m_{HE2}$     | kg                    | 7607                  | 6644                  |
| $m = m_{HE1} + m_{HE2}$ | kg | 11560                 | 10360                 |
| $P_{LPT}$     | kW                    | 1338                  | 1234                  |
| $\eta_{HATPP}$ | %                     | 24.7                  | 24.3                  |
The following considerations are made:

- Incoloy 800HT tubes are used for HE1 (the hottest side); a price of 75 Euro/kg is assumed;
- Austenitic stainless steel TP304L tubes are used for HE2; a price of 3 Euro/kg is assumed;
- In Romania, the selling price of the electric energy is fluctuating but certainly never drops below 10 Euro/MWh; the average price should be currently rated to 30...35 Euro/MWh. In what concerns the green certificates, the minimum current value is 29 Euro/MWh. Consequently, considering the most pessimistic scenario, an income of 39 Euro/MWh was assumed for the energy produced.

Following these assumptions, it results an overall supplemental cost of 237 kg × 75 Euro/kg + 963 kg × 3 Euro / kg = 20664 Euro in Case 1. This implies 20664 / 39 = 530 MWh of electric energy to be sold. Admitting that payback for supplemental cost should be entirely covered just by the additional electric power produced, this means 530 MWh / 0.0728 MW = 7280 h of full operation. In one of the most pessimistic scenarios, considering a seasonal operation period of 2160 h/year (three month / year), this means a payback period of three years and one more month of full operation in the fourth year. Obviously, this is more than reasonable, so πc = 12 (Case 1) can be assumed as optimum case not only from performances point of view but also in the overall approach.

4. Conclusions

The optimum case from dimensional point of view (Case 2) is described by πc = 16 and corresponds to the minimum dimensional specific parameters of HATPP, namely s sp = 0.269 m²/kW, v sp = 12.34·10⁻³ m³/kW and m sp = 8.39 kg/kW. In absolute units, it means S = 332 m², V = 15.22 m³ and m = 10.36 t.

In the optimum case from performance point of view (Case 1), described by πc = 12, dimensional specific parameters of HATPP are higher, namely s sp = 0.277 m²/kW, v sp = 12.7·10⁻³ m³/kW and m sp = 8.64 kg/kW. In absolute units, these means S = 370.5 m², V = 17 m³ and m = 11560 kg.

In one of the most pessimistic scenarios, assuming 75 Euro/kg for HE1 price, 3 Euro/kg for HE2 price, an income of 39 Euro/MWh for the energy produced and 70% efficiency of PLPT conversion into electric power, it results a payback period of 7280 h (full operation) for the costs of the additional heat exchange surface required in Case 1. This is very convenient, so it can be concluded that Case 1 is the optimum one, not only from performances point of view but also in the overall approach.

5. References

[1] Kerstetter J D and Lyons J K 2001 Wheat Straw for Ethanol Production in Washington: A Resource, Technical, and Economic Assessment Washington State University, Cooperative Extension Energy Program
[2] Bălănescu D T, Homutescu V M, Hrițcu C E and Atanasiu M V 2016 Air turbine – an interesting solution for straw energy conversion into electricity IOP Conf. Ser.: Mater. Sci. Eng. current issue
[3] Saravanamuttoo H, Rogers G, Cohen H and Straznicky P 2008 Gas Turbine Theory - 6th Edition (Pearson Education)
[4] Neaga C and Epure A 1988 Îndrumar - Calculul termic al generatoarelor de abur (București: Ed. Tehnică)
[5] Basu P, Kefa C and Jestin L 2000 Boilers and Burners – Design and Theory (New York: Springer)
[6] Incropera F P, Dewitt D P, Bergman T L and Lavine A S 2007 Fundamentals of Heat and Mass Transfer, 6th Edition (Hoboken: John Wiley & Sons)

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