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Some design characteristics of micro steam turbines for agricultural biomass energy conversion

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Abstract. The paper continues the study of reconversion of a 400 kW hot water boiler in a steam generator suitable to valorise the energy content of briquettes and pellets of agricultural biomass. After steam parameters selection (pressure, temperature, mass-flow rate), an overview of main steam machines types (axial, radial, screw, piston engine) is done. Further, a parallel design of most wide-spread ones (Laval, Curtis and radial) were performed, at different rotation speeds, in order to find the best configuration in respect with the flow section dimensions, internal efficiency and power (electrical and thermal) output. The results of the paper could be very useful for the investors in agricultural "waste-to-energy" projects in order to select appropriate technology and equipment.

1 Introduction

During last 15 years, the authors belong to a large research team who studied the biomass by-products (solid, liquid and gaseous) in useful energy (thermal and electrical) to be used locally or remote. One of these researches [1] was focused on mini thermal steam power plant able to give economic value to new energy willow crops developed in our Country.

In a previous recent paper [2] we focused on the feasibility to convert a hot water boiler powered by solid biomass waste into a steam generator, in order to couple it with a condensing or backpressure steam turbine. The purpose of current paper is to select the type of steam engine suitable for this task, based on design feasibility and efficiency estimation.

In a "white paper" edited by a manufacturer [3], is stipulated that the steam microturbines are commonly used to reduce the pressure of process steam, working in parallel to a conventional pressure reducing station with high reliability with a service life of 15-25 years and important cost savings related to the generated electricity.

Other equipment producers [4-5] underline the advantages of the microturbines used in district heating systems such as challenging space limitations, satisfying the reliability requirements, operational simplicity, meeting the sound criteria of equipment rooms; and a

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2 to 5 year payback investment. With all the parts into one seamless part, the volume is vastly reduced, allowing the maintenance simple, compact and affordable.

The specification of such axial (Laval and Curtis) micro-turbine-generators are revealed by other manufacturer [6-7]:

- Inlet pressure: 2 – 30 [bar]
- Back Pressure: 0,1 – 15 [bar]
- Turbine speed: 12000 – 14400 [rpm]
- Frequency: 50 / 60 Hz
- Voltage: 380 V / 415V

Kubawara et al [8] present the design characteristics of a micro steam expander based on the screw principle, with outputs of 12-132 kW, with internal efficiencies of 60-70 %.

Another manufacturer [9] gives the range of electric power for its products (50 – 200 kWe) and announces other advantages:

- Efficient operation in the conditions of wet steam
- Excellent ratio power/weight
- Working life min. 100 000 hours
- Full power reaction time in 10 minutes

Finally, Alford et al [10] offer a comprehensive analysis of a microturbine based upon an integral high speed air compressor, optimised by means of numerical modelling, designed, manufactured and tested in a wide range of conditions with good efficiency and endurance results.

2 Design characteristics of axial turbines

2.1 Input data and initial hypothesis

According to Figure 1 that shows the main thermal circuit of the plant, the input parameters revealed in paper [1] are:

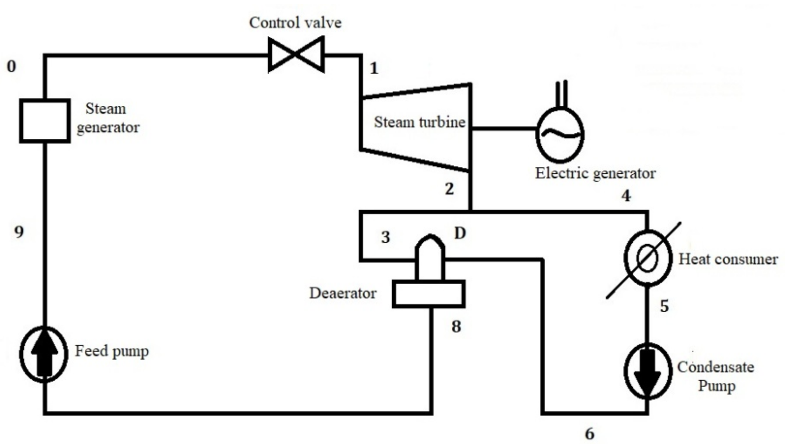

Fig. 1. Thermal circuit of the plant

- Live steam pressure p_0= 8 [bar];
- Live steam vapour content x_0=1 [-] (saturated);

- Exhaust pressure to the consumer p_2= 1.2 [bar];
- Live steam mass flow rate $\dot{m}$=0.35 [kg/s].

The two axial turbines selected for this analyse were the well known one stage Rateau and Curtis types, presented in figure 1

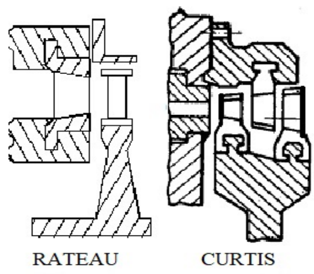

Fig. 2. One stage axial turbines Rateau and Curtis

Due to the very small expansion ratio β=0.16 < 0.546 = β_{cr}, a convergent-divergent nozzle type was selected, in order to obtain high steam velocities. The other common feature is the impulse stage type.

Meanwhile, in order be able to convert efficiently and to have small dimension, the rotation speed should be higher than 50 rps. Thus, three values were selected for this parameter, respectively 100, 150 and 200 rps. It is obvious that the system should be provided with a reducing gearbox or with a frequency convertor in the electric generator.

2.2 Methodology

The designed methodology is inspired from [11] and is currently used in didactic and applied research activities in our University.

First of all, the optimal velocity ratio is calculated:

$$x_{opt} = \frac{u}{c_1} = k_x \cdot \frac{\cos \alpha_1}{2 \cdot i \cdot (1-\rho)} \tag{1}$$

where k_x = 0,8…1;
ρ = 0 (reaction degree)
α_1- effective exit nozzle angle [°]
i – number of mobile rows of blades (1 for Rateau and 2 for Curtis)

Secondly, the absolute nozzle exit velocity can be found:

$$c_1 = \varphi \sqrt{2(H_a + H_0)} \tag{2}$$

where φ – nozzle losses coefficient
H_a – nozzle isentropic drop, equal to the stage isentropic drop H_t [J/kg]
H_0 – input kinetic energy [J/kg]

Combining eq. (1) and (2) the peripheral stage velocity u [m/s] allows to compute the average stage diameter d_1 [m]:

$$d_1 = \frac{u}{\pi \cdot n} \tag{3}$$

Imposing a minimal (related to the acceptable flow losses) and constant (respecting the speed rotation n) nozzle height L_{n1} of 10 [mm] for Rateau and 20 [mm] for Curtis, from flow equation it results the criterion for establishing the optimum rate of admission ε [-], meaning the ratio between the surface of nozzles exit and the whole circular crown surface available:

$$\varepsilon L_{n1} = \frac{\dot{m} \cdot v_1}{\mu \cdot \pi \cdot d_1 \cdot c_1 \cdot \sin\ _1} \tag{4}$$

where μ – filling surface coefficient depending on φ;
v_1 – specific volume at nozzles exit [m^3/kg].

It's now possible to compute the components of velocity diagram for the two selected turbines, in order to assess both main losses (ΔH_m [kJ/kg] related to the nozzles, blades, exit kinetic energy and secondary losses (ΔH_s [kJ/kg] related to the friction and ventilation, partial admission and humidity), and also the internal enthalpy drop H_i [kJ/kg], the internal turbine efficiency η_i [-] and the electric generator power P_g [kW]:

$$H_i = H_t - \Delta H_m - \Delta H_s \tag{5}$$

$$\eta_\mathrm{i} = \frac{H_i}{H_t} \tag{6}$$

$$P_g = \eta_m \cdot \eta_\mathrm{g} \cdot \dot{m} \cdot H_i \tag{7}$$

where η_m – mechanical efficiency [-];
η_g – electric generator efficiency [-];

2.3 Results and discussions

The common data of the two turbines with no connection with the rotation speed are presented in Table 1:

Table 1. Common parameters non related to the speed

Parameter	Unit	Rateau	Curtis
x_{opt}	[-]	0.442	0.215
c_1	[m/s]	734.3	743.8
u	[m/s]	324.3	160.0
α_1	[º]	11	17
L_{n1}	[m]	0.01	0.02

The other parameters depending to the rotation speed are shown in the following figures:

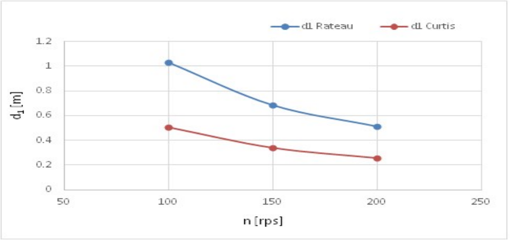

Fig. **3** Average diameter versus rotation speed

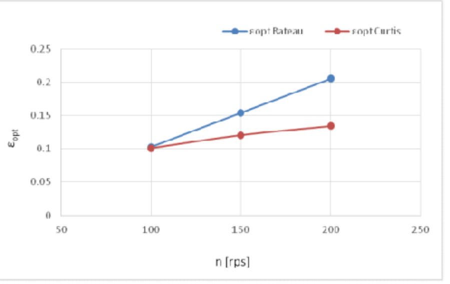

Fig. 4. Admission rate versus rotation speed

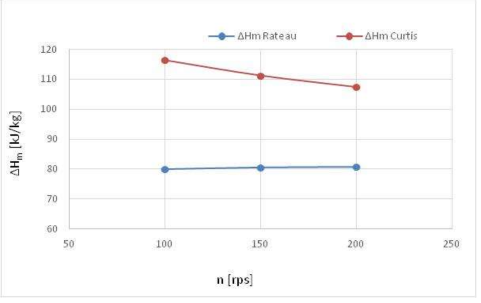

Fig. **5** Main losses versus rotation speed

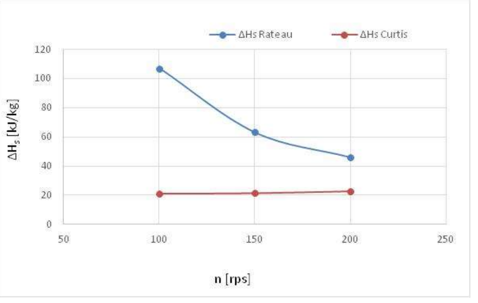

Fig. **6** Secondary losses versus rotation speed

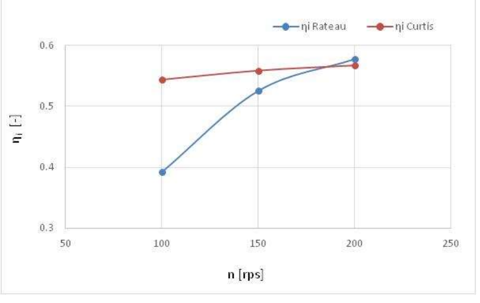

Fig. 7. Internal efficiency versus rotation speed

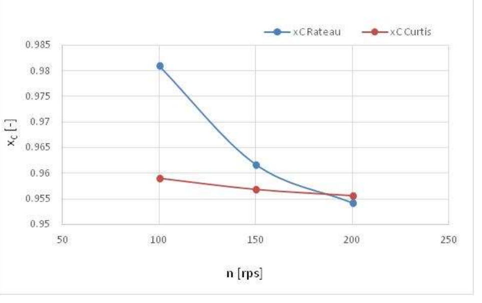

Fig. 8 Exhaust vapour content versus rotation speed

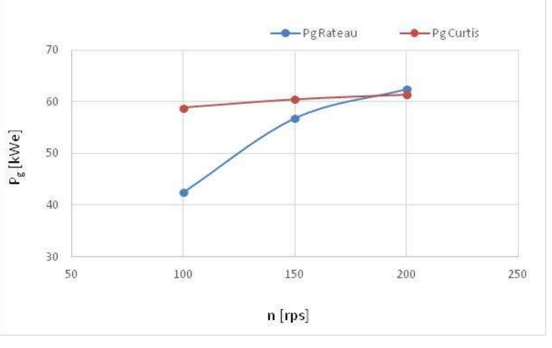

Fig. 9. Electric power versus rotation speed

In Figure 3 is displayed the average diameter of the stage versus the rotation speed. It can be seen the diameter decrease when the speed increases, and also the smaller diameter for Curtis than Rateau.

The admission rate (Figure 4) seems to be higher for Curtis than for Rateau, but only at rotation speeds smaller than 200 rps.

The main losses, presented in Figure 5, are higher for Curtis than for Rateau, due to the three rows of blades instead of only one.

But the secondary losses, shown in Figure 6 are significant more important for Rateau than for Curtiss, due to the weight of the friction and ventilation loss, depending with u^3.

Corroborating the two last sentences, the internal efficiencies are the same at 200 rps, but Curtiss beats Rateau at lower speeds. (Figure 7)

An important parameter to be checked is represented by the vapour content at the turbine's exhaust. In order to protect the blades against the water droplets erosion its value should be higher than 0.88. In the Figure 8, this condition is fully respected, the smaller value registered for both turbines is about 0.955 at 200 rps. A supplementary erosion test with Esher-Wyss and Hitachi criteria confirmed this affirmation.

The final and important parameter is represented by the power delivered by the electric generator (Figure 9). At 200 rps, the values are identical for both turbines (about 62 kW_e), but at lower speeds Curtiss is more powerful than Rateau.

The overall efficiency (electric+thermal) is the same for all configuration at the value of 0.914, that means that the biomass conversion into energy has a high efficiency in comparison with other energy conversion chains.

3 Design characteristics of radial turbines

A third solution analyzed was a radial-inflow turbine type. The centripetal effect makes possible to change as much energy as possible from the fluid into useful work. Some other significant advantages can be mentioned in comparison with the axial-flow turbine: a greater amount of work per stage, a simplest manufacturing, a high ruggedness and a lower cost [12]. The radial turbines are more efficient than the axial one for small mass flows and can perform large enthalpy drops with relatively low peripheral speeds. The turbine performance is a function of the similarity parameters expressed by specific speed n_s and specific diameter D_s, and for compressible flow Reynolds number R_e^* and Mach number M_a^*.

$$n_S = \frac{n \cdot \dot{V}_3^{1/2}}{H_t^{3/4}} \tag{8}$$

$$D_S = \frac{D \cdot H_t^{1/4}}{\dot{V}_3^{1/2}} \tag{9}$$

For the project data the isentropic enthalpy drop is H_t = 325.6 kJ/kg and the volumetric rate of flow relative to the rotor exit is V_3 = 0.22 m^3/s; the spouting velocity c_0 = 807 m/s. In the case of ideal radial turbine, with complete recovery of the exhaust energy the optimum velocity ratio is 0.707 and the typical range is indicated $0.55 < u_2^2/c_0 < 0.77$ [13].

If the normal speed is considered (n = 50 rps), than the specific speed is n_s = 0.103 and for Imperial units n_s = 1,395 or Ω_s = 0.0107 rad. The Balje diagrams (n_sD_s turbine chart) shows that this low value of specific speed is outside of the domain for the radial-inflow turbine with good performance [12-15]. The solution with maximum efficiency is indicated to be a partial admission axial turbine.

A high speed could be a solution for the inflow-radial turbine. A desired minimum efficiency and a corresponding specific speed can be determine on the Balje N_S – D_S and Rohlick diagrams, like as n_s = 0.6 rad. for a total to static efficiency η_{ts} = 0.8. Substituting this value, the imposed isentropic enthalpy drop and mass rate of steam flow into the n_s equation the rotating speed of about n = 3320 rps is found. The thermal computation of the stage for a reaction degree of ρ = 0.55 gives the fluid velocities and geometric dimensions. Mach numbers relative to the absolute velocities are slightly larger than 1, to the relative velocity M_{w1} = 1.08, M_{w3} =1.4 but the inlet blade length has an unacceptable value of about 2 mm. In these conditions a new evaluation of stage losses can decrease substantially the efficiency and the turbine solution became inacceptable.

As a conclusion the very low steam flow and the too large enthalpy drop are incompatible with the inflow radial turbine type. Splitting the enthalpy drop between two radial turbines does not significantly improve the results.

4 Conclusions

A lot of agricultural waste are generated yearly. The farmers use it as fuel for stoves and hot water boilers. Some of them need electricity and heat for adjacent activities (bakery, milk/meat processing, wood industry, etc.)

A good opportunity is to transform the flame tubes hot water boilers into steam generators and to couple it with microturbines.

The novelty of this paper is represented by the comparative analysis of axial and radial turbines regarding the design characteristics and operating performances. The main parameter is represented by the rotation speed.

One of the conclusions was that axial turbines are more suitable for this task in comparison with radial ones. Rateau is more simple and cheaper, Curtis is more complicate and expensive, but more robust.

The second conclusion is that higher rotation speeds (200 rps) offers compactness and better efficiencies than lower rotation speeds (100 rps).

The market offers microturbines in the power range of 15-300 kW_e, in multiple constructive technologies and performances. This paper can help the investors to choose one of it.

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