COMPARATIVE ENERGETIC ASPECTS REGARDING STEAM DRIVEN TURBINES

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Abstract: One of the main possible methods for the increase of the thermal proficiency of a steam turbine is actually the method which acts on the heat source and namely the increase of initial pressure. The paper brings forward the determination of the absolute thermal efficiency of the specific heat and steam consumption in 2 distinct situations.

Key words: the increase of the initial pressure of steam, thermal efficiency, specific fuel consumption, specific heat consumption, Rankine cycle.

1. INTRODUCTION

Presently, the increase of the efficiency of the Rankine cycle is realised through a series of methods which act upon the heat source and methods which act upon the cold source. For a simultaneous increase of pressure and temperature with 40 bar, respectively 30°C, the speciality literature mentions a possible increase of the thermal efficiency with 5% [2]. This efficiency increase method is relieved of a series of inconveniences [1]:

- the effect of the increase of the initial pressure is an increase of the relative humidity of the steam within the final area of the turbine;
- the increase of the initial parameters implies increased investment efforts.

This method used to increase the efficiency is justified especially when: the uniform power of the group is increased; the yearly usage period of the installed power is high; the fuels used are expensive.

The paper proposes to bring forward the determination of the thermal efficiency, the actual absolute efficiency of the specific heat consumption and the specific steam consumption for 2 distinct situations: when the turbine is in operation at designed parameters and when the steam pressure is increased from 90 to 110 bar. The

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steam title within the turbine is determined, being known that this method of increasing the initial pressure leads to the increase of its title.

2. THE EQUATIONS USED FOR THE DETERMINATION OF STEAM PARAMETERS. THERMODYNAMIC PARAMETERS OF THE TURBINE

In order to establish the parameters used by the study, the following measures shall be defined: the real expansion is equal to the entropy increase:

$$H_r = \sum h_r = i_0 - i_{final\ step\ output} \tag{1}$$

The following relation is considered between the real and the theoretic expansion:

$$H_r = \eta_i \cdot H_t \tag{2}$$

The title curves x = ct. are determined using the following relations:

$$x = \frac{i_{ir} - i'}{r} \tag{3}$$

This sum also includes the losses due to friction, those occurred through spreading due to the partial admission, due to humidity and those due to steam escape.

The thermal efficiency of the Rankine cycle is the following ratio:

$$\eta_t = \frac{H_t}{i_1 - i_a} \tag{4}$$

Effective power:

$$P_e = \eta_m \cdot \dot{m} \cdot H_r \tag{5}$$

Actual efficiency:

$$\eta_e = \frac{l_e}{l_t} = \eta_m \cdot \eta_i \tag{6}$$

Specific steam consumption:

$$q_{ab} = \frac{\dot{m}_h}{P_e} = \frac{3600}{\eta_e \cdot H_t}\ \text{[kg/kWh]} \tag{7}$$

Specific heat consumption:

$$q_q = \frac{3600}{\eta_{ea}} [kJ / kWh] \tag{8}$$

Actual absolute efficiency:

$$\eta_{ea} = \frac{l_e}{q_1} = \eta_m \cdot \eta_i \cdot \eta_t = \eta_e \cdot \eta_t \tag{9}$$

3. CASE STUDY

The study follows the increase of the efficiency of the steam turbine using the method which acts of the hot source, and the increase of the initial pressure. 2 distinct situations are analysed: the first case considers the parameters of the steam in the turbine to be p_2 = 90 bar and t_1 = 500 ^{0}C. While during the second case the initial pressure is increased to 110 bar. In both cases the expansion in the condenser is realised up to the following parameters p_2 = p_{2c} = 0.04 bar and t_2 = 30 ^{0}C. The condition that the relative humidity doesn't exceed 90% is also verified, distroing the final part of the turbine.

Geometric and thermodynamic parameters of the turbine. The studied turbine is a jet propelled turbine and is composed of a speed wheel as the adjustment wheel and 17 pressure steps [3]. The turbine has been calculated to operate with live steam at a pressure of 90 bar and a temperature of 500^{0}C, measured before the inlet of the automated valve of the turbine. The temperature of the cooling water at the inlet of the condenser was measured to be 10^{0}C, the pressure of the expanded steam is p_{2c} = 0.04 bar, and the temperature t_{2c} = 30 ^{0}C.

The turbine allows a long term operation for a pressure within the range of 85 – 105 bar and a temperature within the range of 470-505^{0}C but the temperature of the water supplied to the condenser not higher than t_{1c} = 10 ^{0}C, if the parameters of the live steam (pressure and temperature) are smaller than the nominal ones. In order to improve the thermal parameters of the turbine the initial pressure of the steam is increased to 110 bar. The theoretic and real expansions in the turbine considering the nominal design parameters are presented in Figure 1. In Figure 1, H_t represents the isentropic expansion from the parameters of steam p_1 = 90 bar and t_1 = 500^{0}C and the pressure of the condenser p_2 = p_{2c}= 0.04 bar and t_{2c}= 30^{0}C.

As the constant pressure lines are divergent to the right, the real process moving away from the vertical line, the sum of the falls on a step are larger than the total theoretic fall H_t: $\sum h_t > H_t$. This is explained through the fact that due to the losses, steam is heated and the specific volume increases more than for the isotropic expansion.

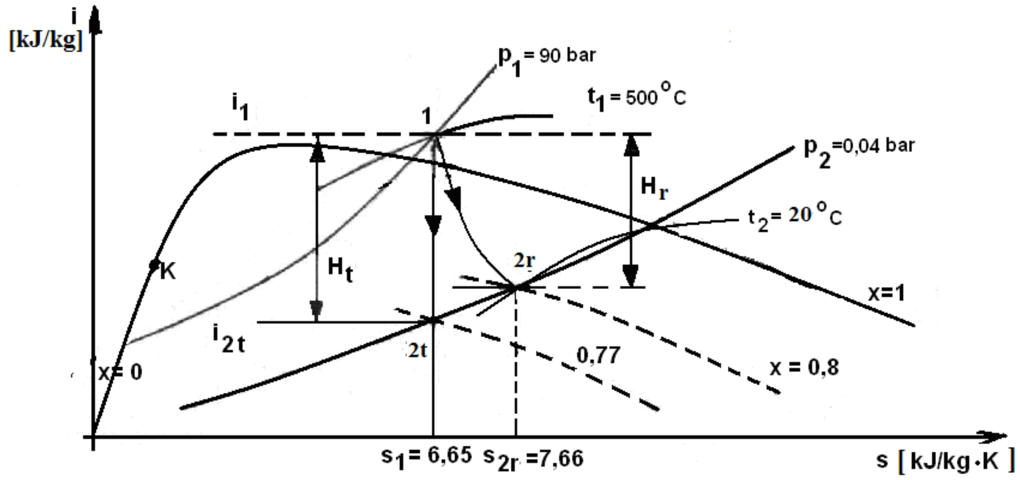

Fig. 1 Theoretic and real expansion of steam

This additional dilatation leads to an enhanced mechanical work. The phenomenon recovers heat as during the additional mechanical work a part of the lost energy is found[1].

$$\sum h_t = (1 + 1.1 \div 1.3) H_t = 1.2\ H_t$$

Therefore, the internal efficiency of the turbine becomes:

$$\eta_i = \frac{H_r}{H_t} = \frac{H_r}{1.2 \cdot H_t} = \frac{i_1 - i_{iesire\ ultima\ treapta}}{1.2\ H_t} \qquad (10)$$

Taking into consideration the values of the temperature and the initial pressure for steam (p_1 = 90 bar and t_1 = 500 ^{0}C (point 1) and the pressure of the condenser p_{2C} = 0.04 bar (point 2t), measure H_t = 1378 kJ/kg, is determined through the difference on the axis of the ordinates in diagram *i-s*.

The presented efficiency is expressed as follows:

$$\eta_i = \frac{H_t}{1.2 \cdot H_t} = \frac{1378}{1.2 \cdot 1378} = 0.87$$

The thermal efficiency of Rankine cycle is the ratio:

$$\eta_t = \frac{1378}{3386 - 30} = 0.41$$

where i_1 is the steam's enthalpy considering p_1, T_1; i_a the enthalpy of the water supplied to the boiler, the values being determined correspondingly to the table L +

VSI. The optimum pressure value of the condenser reaches the value p_{2C} = 0.04 bar to which it corresponds a $t_{2C} \approx 30\ ^0$C. The mechanical efficiency was considered from the technical specifications of the manufacturer, namely η_m = 0.9.

The actual efficiency becomes:

$$\eta_e = \eta_i \cdot \eta_m = 0.87 \cdot 0.9 = 0.783$$

while the effective power realised by the turbine at the nominal parameters becomes:

$$P_e = 0.9 \cdot 53 \cdot 1062 = 50.5\ MW$$

where the steam mass flow at the nominal load of 50 MW is 190.5 t/h = 53 kg/s; H_r = $i_1 - i_{\text{final step output}}$ = 3386 - 2324 = 1062 kJ/kg.

The absolute actual efficiency:

$$\eta_{ea} = \eta_e \cdot \eta_t = 0.783 \cdot 0.41 = 0.783 = 0.321$$

and the specific heat consumption is:

$$q_q = \frac{3600}{0.321} = 11214.25 \quad kJ/kWh$$

The value of the specific heat consumption is higher than it is foreseen in the technical documentation of the turbine, which is within the range of 9500-9700 kJ/kWh (considering the theoretic data for 50 MW.)

The specific steam consumption becomes:

$$q_{ab} = \frac{3600}{0.783 \cdot 1378} = 3.33\ kg/kWh$$

Applying the method which acts on the heat source, namely the increase of the initial pressure, to 110 bar, the efficiency gain, the specific steam and heat consumption will be determined for the new conditions. It is imposed to respect the condition that the dryness fraction in the turbine, following the increase of pressure, shall not exceed x = 0.9. Following the increase of initial pressure to 110 bar, the specific values which show any interest within the calculation is modified. The representation of the thermal processes may be observed in figure 2, and the specific energetic values were determined using diagram *i-s*. The values of the enthalpies of points 1 (90 bar; 500 ^{0}C), 2 (100 bar; 500 ^{0}C), 3 (110 bar; 500 ^{0}C), are simply determined using the corresponding L + VSI tables.

$$i_1 = 3386; \quad i_2 = 3372; \quad i_3 = 3360 \text{ kJ/kg}$$

The values of the enthalpies of the points are obtained from the isentropic expansions of diagram i-s, while for the real expansions with an increase in entropy 1r, 2r and 3r are determined using the relation of the of the interior efficiency.

$$H_r = \eta_i \cdot H_t$$

Diagram *i-s* gives the specific values of the theoretic enthalpies and entropies and the dryness fraction, considering the pressure of the condenser, p_{2C} = 0.04 bar and the fact that transformations 1-1t, 2-2t, 3-3t are isentropic.

$$i_{1t} = 2008; \quad i_{2t} = 1968; \quad i_{3t} = 1936 \text{ kJ/kg}$$

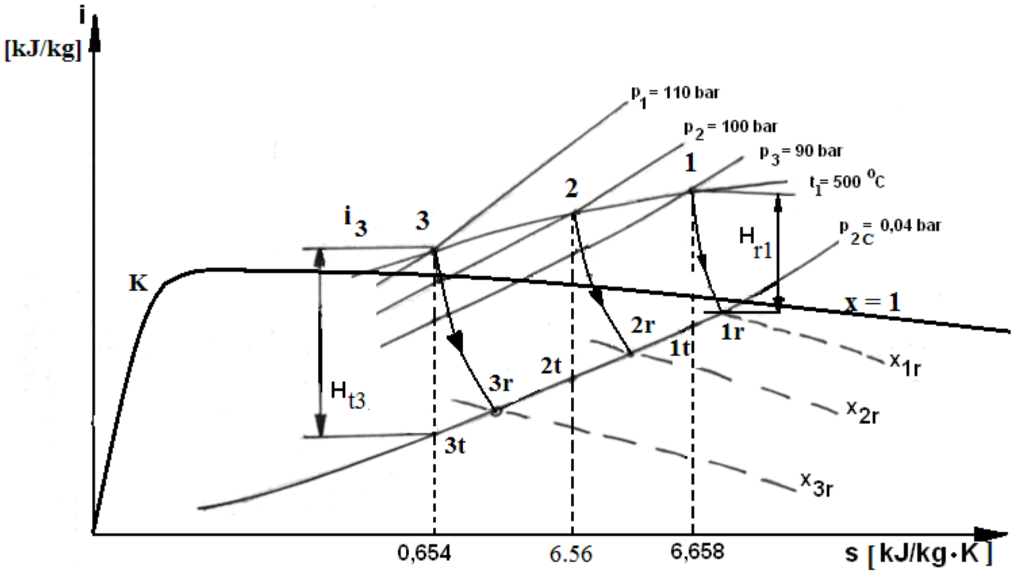

Fig. 2 The increase of the initial parameters of steam from 90 to 110 bar

$$s_{1t} = 6.658; \quad s_{2t} = 6.6; \quad s_{3t} = 0.654 \text{ kJ/(kg·K)}; \quad x_{1t} = 0.9; \quad x_{2t} = 0.768; \quad x_{3t} = 0.758$$

Therefore, the theoretic enthalpy decreases become:

$$H_{t1} = i_1 - i_{1t} = 3386 - 2008 = 1376 \text{ kJ/kg}; \; H_{t2} = i_2 - i_{2t} = 3372 - 1968 = 1404 \text{ kJ/kg}$$

$$H_{t3} = i_3 - i_{3t} = 3360 - 1936 = 1423 \text{ kJ/kg}$$

and the real specific enthalpies are obtained using the relation:

$$i_{1r} = i_1 - \eta_i \cdot (i_1 - i_{1t}) = 3386 - 0.87 \cdot 1376 = 2188.88 \text{ kJ/kg}$$
$$i_{2r} = i_2 - \eta_i \cdot (i_2 - i_{2t}) = 3372 - 0.87 \cdot 1404 = 2150.52 \text{ kJ/kg}$$
$$i_{3r} = i_3 - \eta_i \cdot (i_3 - i_{3t}) = 3360 - 0.87 \cdot 1423 = 2121.99 \text{ kJ/kg}$$

The admissible limit for the dryness fraction is x = 0.8÷0.9. In order to verify the completion of these conditions for the dryness fraction the real title is calculated for all the 3 values of the initial pressure of steam.

The latent vaporising heat, according to the output parameters of the condenser (p_{2C} = 0.04 bar; t_{2C} = 30^{0}C), is determined from table LS + VSU:

$$r = i'' - i' = 2433 \quad \left(i' = 121.42;\, i'' = 2554.4\right) \text{ kJ/kg}$$

Therefore:

$$x_{1r} = \frac{i_{1r} - i'}{r} = \frac{2188.88 - 121.42}{2430} = 0.849\,;\; x_{2r} = \frac{i_{2r} - i'}{r} = \frac{2150.52 - 121.42}{2430} = 0.833\,;$$

$$x_{3r} = \frac{i_{3r} - i'}{r} = \frac{2121.99 - 121.42}{2430} = 0.822$$

It is observed a sensitive decrease of the dryness fraction. The difference $\Delta x = x_{3r} - x_{1r} = 0.027\,(2.7\%)$ is considered to be insignificant and favours the presentation of a large number of water drops within the steam which expands at high speed (>200 m/s), which leads to a pronounced erosion phenomenon and the destruction of the rotor blades from the final area of the turbine.

The specific thermal efficiency of the cycle becomes:

$$\eta_t = \frac{H_{t3}}{i_3 - i_a} = \frac{1423}{3360 - 30} = 0.42$$

The actual absolute efficiency:

$$\eta_{ea} = \eta_e \cdot \eta_t = 0.783 \cdot 0.42 = 0.783 = 0.329$$

The specific heat consumption becomes:

$$q_q = \frac{3600}{\eta_{ea}} = \frac{3600}{0.389} = 9254.5\ kJ/kWh$$

The specific steam consumption for the following pressure case p_1 = 110 bar and H_{t3} = 1423 kJ/kg becomes:

$$q_{ab} = \frac{3600}{\eta_e \cdot H_{t3}} = \frac{3600}{0.783 \cdot 1423} = 3.23\ kg/kWh$$

The obtained values situate themselves within the technical documentation of the turbine.

4. CONCLUSIONS

The method to increase the initial pressure of steam is considered advantageous taking into account that the unitary power of the group is increased, the yearly usage period of the installed power is high and the fuel used is very expensive. The increase of the initial pressure of steam from 90 bar to 110 bar leads to the following improvements: the increase of the thermal efficiency with 2.38%; the increase of the actual absolute efficiency 3.43 %; the specific heat consumption decreases significantly with 17.82 %; the specific steam consumption decreases 3%.

In the same time, the condition that the dryness fraction doesn't decrease drastically is respected (the dryness fraction decreases together with the initial pressure of steam), this being within the limits of the design, $x = 0.83\text{-}0.9$. Contrary, the benefits obtained through the increase of the initial pressure are counterbalanced by an increase of the humidity of steam in the final area of the turbine and the significant decrease of the dryness fraction, which implies the destruction of the turbine due to the cavity phenomenon.

REFERENCES:

[1] **Pimsner V., and others**, *Termodinamica tehnica*, Problems Workbook, Didactic and Pedagogic Publishing House, Bucharest, 1976

[2] **Badea A., and others**, *Echipamente si instalatii termice*, "Tehnica" Publishing House, 2003.

[3] **Păsculescu, D., Pasculescu, V**., *Presentation and simulation of a modern distance protection from the national energy system*, 10th International Conference of Environment and Electrical Engineering, Rome, Italy, 6-11 May 2011, IEEE Catalog Number: CFP1151I-CDR, ISBN 978-1-4244-8781-3, pp. 647-650.

[4] **Păsculescu, D., Niculescu, T., Pana, L.,** *Uses of Matlab Software to size intrinsic safety barriers of the electric equipment intended for use in atmospheres with explosion hazard*, Proceedings of the International Conference on Energy and Environment Technologies and Equipment (EEETE '10), RECENT ADVANCES in ENERGY and ENVIRONMENT TECHNOLOGIES and EQUIPMENT, Bucharest, Romania, April 20-22, 2010, ISSN: 1790-5095, ISBN: 978-960-474-181-6, pp. 17-21.