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MODEL FOR ANALYSIS OF THE IMPACT OF HYDRAULIC PARAMETERS IN EFFICIENCY OF A PUMP-TURBINE SYSTEM

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ABSTRACT
Using electrical power generated in the thermal power plants is variable depending of the daily, weekly and seasonal cycles of industrial and domestic requirements. Principally to mitigate the difference between power production and electric power required in many cases used by adjusting hydraulic with pump-turbine systems that will be addressed in this paper. Ways of hydraulic adjustment similar to this study are currently installed in various regions. Thus the analysis of hydraulic parameters of a pump-turbine system, in particular their impact on the performance of this system are the interest to technical and economic analysis
Keywords: *pump-turbines, power, performance*

1. INTRODUCTION
Electric power produced by a thermal power plant should remain constant to satisfy the technical and economic optimum conditions of production. Use of this power varies according to function of requirements. Way of hydraulic regulating with pump-turbine system that will be analyzed in this paper, in schematic form shown in Figure 1.

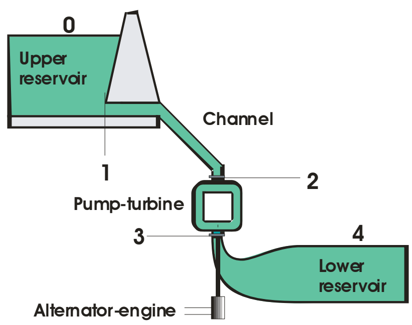

Figure 1.

Installing the pump-turbine system consists of functional group either as turbine that plays role of an alternator in regime of emptying the upper reservoir, or as pump driven by an electric motor with load mode. The level of filling of the upper and lower reservoir assumed that is constant. From figure 1 seems that are known the heights of positions (z_0, z_1, z_2, z_3 and z_4) of the five sections of the tube 0, 1, 2, 3 and 4 with diameter D of the system [3]. Power transmitted over the block axis is $(P_c)_t$ and mechanical efficiency of turbine and pump are η_t and η_p.

2. DISCHARGE REGIME

In this case the system works as turbine. The speed v of water in section 1, 2 and 3 calculated using the mechanical energy balance [2] written between section 0 and 4 if not ignore the loss of load in terms of section 2 and 3 by the following equation:

$$\dot{W} - \dot{E}_v = \dot{m}\left[\left(\frac{1}{2}v_4^2 + \frac{P_4}{p} + gz_4\right) - \left(\frac{1}{2}v_0^2 + \frac{p_0}{p} + gz_0\right)\right] \qquad (1)$$

Where are: $\dot{W}$ - communicative power of the moving fluid through walls, $\dot{E}_v$ - energy distribution in units of time. Load losses are zero in upstream 2 and in downstream 3, thus energy distribution has only in the pump. Turbine efficiency η_t defined as the ratio of effective power sending on what is taken from the flow according to equation:

$$\eta_t = (P_e)_t / (\dot{W} - \dot{E}_v) \qquad (2)$$

After that $v_4 = v_0 = 0,\ and\ \ p_4 = p_0 = p_a\ \ and\ \ v_1 = v_2 = v_3$ then the speed of flow through the tube is

$$v = v_1 = v_2 = v_3 = \frac{4}{\pi\rho D^2}\frac{1}{\eta_t}\frac{(P_e)_t}{z_4 - z_0}\frac{1}{g} \qquad (3)$$

Bernoul's theorem shows that the load is constant in all flow lines [1]. Therefore:

$$h_0 = h_1 = h_2 = z_0 + \frac{p_0}{pg} + \frac{v_0^2}{2g} = z_0 + \frac{p_a}{pg} \qquad (4)$$

$$h_3 = h_4 = z_4 + \frac{p_4}{pg} + \frac{v_4^2}{2g} = z_4 + \frac{p_a}{pg} \qquad (5)$$

3. LOADING REGIME

In this case the group operates as a pump. Flow rate $\dot{m}$ is identical to that of the discharge regime evolved, but the flow circulating in the opposite direction. Under these conditions, there is a loss of load at the exit of section 1 due to significant expansion. Required the calculation of loss of corresponding load to this expansion as well as the power $(P)_p$ delivered by the axis group in pump.
Its performance is η_p. It is assumed that there will be no loss of load between sections 2 and 1 and the other in 4 and 3. Corresponding load loss calculated with the formula as follows:

$$e_S = k\frac{v_1^2}{2} = \left(1 - \frac{A_{gjere}}{A_{ngushte}}\right)^2 \frac{v_1^2}{2} \approx \frac{v_1^2}{2}\frac{J}{kg} \qquad (6)$$

A_{wide} and A_{narrow} present mutually narrow and wide section. Since the ratio of areas is A_{narrow} / A_{wide} $<<1$ then the coefficient of loss of load is $k = 1$.
Mechanical energy balance between sections 4 and 0 states in this form:

$$\dot{W} - \dot{E}_v = \dot{m}\left[\left(\frac{v_0^2}{2} + \frac{p_0}{p} + gz_0\right) - \left(\frac{v_4^2}{2} + \frac{p_4}{p} + gz_4\right)\right] \qquad (7)$$

Where $\dot{E}_V = \dot{m}e_S$. Since $v_4 = v_0 = 0,\ \ p_4 = p_0 = p_a$ and power for driving the pump is:

$$(P)_P = \frac{\dot{W}}{\eta_P} = \frac{\dot{m}[e_S + g(z_0 - z_4)]}{\eta_P} \quad (8)$$

Compared to the case for calculations of the regime of discharge, changes belongs only to pressure in section 1 and 2; while the speed and height do not change.
According Bernoul's theorem for sections 1 and 2 and between sections 3 and 4 we get:

$$p_1 = p_a + pg(z_0 - z_1) \quad (9)$$
$$p_2 = p_3 + pe_S + pg(z_0 - z_4 - z_2 + z_3) \quad (10)$$
$$p_3 = p_a - pgz_3 \quad (11)$$

4. GLOBAL BALANCE

For global synthesis of system, modes of loading and unloading can be considered as isotherm. The balance of mechanical energy and internal energy based on first principle of thermodynamics for open systems in permanent regime has the form:

$$\dot{W} + \dot{Q} = \Delta\dot{H} + \Delta\dot{E}_c + \Delta\dot{E}_{pot} \quad (12)$$

Where $\dot{W}$ shows the mechanical work for open system, accepted with a unit of time and Q heat generated per unit time. While $\Delta\dot{H}$ is enthalpy flow ($\Delta\dot{H} = \dot{m}\Delta h(T,p) = \dot{m}(\Delta p/\rho)$) than $\Delta\dot{E}_c$ is flow of kinetic energy ($\Delta\dot{E}_c = \dot{m}(v^2/2)$) and $\Delta\dot{H}_{pot}$ is flow of potential energy ($\Delta\dot{H}_{pot} = \dot{m}gz$) between two divisions of thought. While, between sections 0 and 4 have $\Delta p = 0$, $\Delta\dot{E}_c = 0$ therefore equation (12) takes the following form:

$$\dot{W} + \dot{Q} = \Delta\dot{E}_{pot} = \dot{m}g(z_0 - z_4) \quad (13)$$

A mechanical energy balance on the other side gives:

$$\dot{W} - \dot{E}_v = \dot{m}g(z_0 - z_4) \quad (14)$$

By equating the two preliminary expressions obtained $\dot{Q} = -\dot{E}_v$. All dispersed energy is released out in the form of heat.
Efficiency for the regime of loading, unloading and it global given with the following terms:

$$\eta_c = (\dot{m}g\Delta h)/P = (\dot{W} - \dot{E}_v)/P_{\sup plied} \quad (15)$$
$$\eta_d = P/(\dot{m}g\Delta h) = Ptaken/(\dot{W} - \dot{E}_v) \quad (16)$$
$$\eta_g = Pdownload/P_{load} = \eta_c \cdot \eta_d. \quad (17)$$

5. CALCULATIONS AND ANALYSIS OF RESULTS

Input values in the program are: D=*1.200 m*, ρ=*1000 kg/m*^{3}, η_T=*0.85*, η_P=*0.9*, P_t=*25 MW*, z_0=*320m* z_1=*250 m*, z_2=*7 m*, z_3=*4 m*, z_4=*0m*, p_0=*100000 Pa*. The values of calculated parameters are: S= *1.131m*^{2}, m= *9369 kg/s*, $v_m=v_1=v_2=v_3$=*8.284 m/s*, $v_0=v_4$=*0*, $h_0=h_1=h_2$=*330.194 m*, $h_3=h_4$= *10.194 m* e_s=*34.314 J/kg*, P_P= *33.037 MW*, p_3=*26446 Pa*, (W/m)= *3174 m*^{2}*/s*^{2}, p_2=*3170530 Pa*, p_1=*786700 Pa*.
Change of pressure in the sections along the tube for the case of turbine pump system is provided in the diagram shown in figure 2.
For the case of dismissal regime calculated values of changing position height, velocity height, pressure height and total height of across the tube sections is given in the diagram shown in figure 3, while tor the case of discharged regime in figure 4. While the values of hydraulic power, mechanical power, distribution of power, heat released and internal energy for discharging and charging regime

and the global balance is reflected by the histogram as in figure 5 and figure 6. Global balance is given in figure 7. Hydraulic power is defined equal to $\dot{m}g\Delta z$.

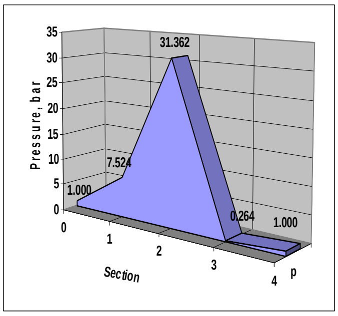

Figure 2

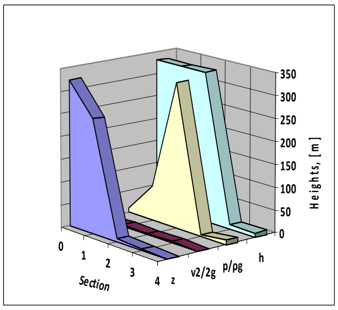

Figure 3.

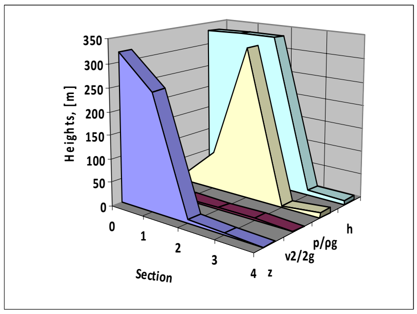

Figure 4.

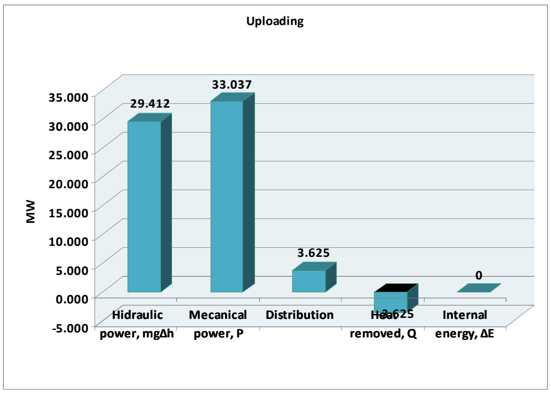

Figure 5.

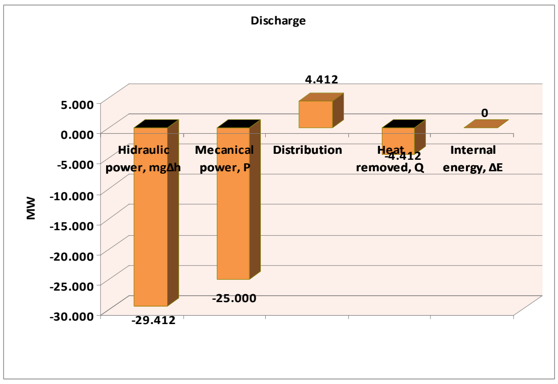

Figure 6.

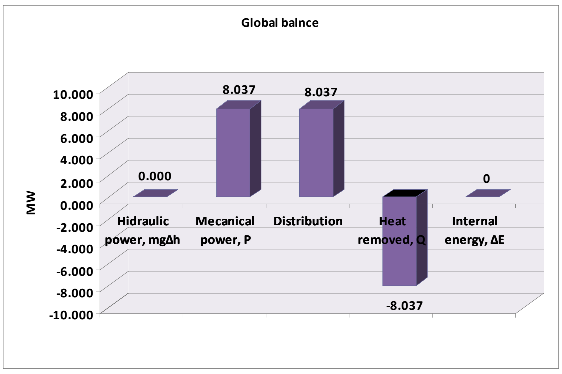

Figure 7.

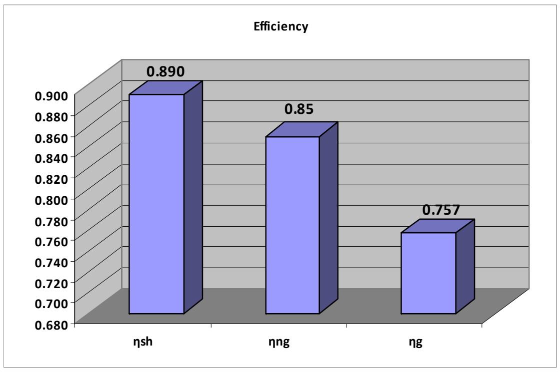

Figure 8.

6. CONCLUSIONS

Hydraulic power in the regime of loading and unloading is equal *(29.412 MW).*

Mechanical power of discharge *(33.037 MW)* is greater than in the case of loading *(-25 MW)* for *8.037 MW.*

Maximum pressure occurs in section 2 (31 362 bar) while the minimal in section 3 (0264 bar). Global efficiency pump - turbine system is $\eta_g=0.757$ implied that is smaller than the regime of discharge $\eta_{sh}=0.890$ and loading $\eta_{ng}=0.850$.

7. REFERENCES

[1] Xh. Fejzullahu, F. Krasniqi, Hidraulika dhe Termodinamika, Prishtinë, 1986.
[2] Wictor L. Steeter, E. Benjamin Wylie, Mc Graw-Hill, Book Company, 1983.
[3] S. Candel, Problemes resolus de Mecanique des fluides, Dunod, Paris 1995.