ALTERNATIVE FOR SMALL HYDRAULIC TURBINES

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Producerea energiei din surse regenerabile şi inepuizabile a devenit o prioritate în ultimii ani. Utilizarea micropotenţialului hidroenergetic disponibil poate fi una dintre soluţii. O alternativă la turbinele hidraulice de mică putere, în vederea obţinerii unor investiţii mai mici, o pot constitui pompele din producţia de serie. În lucrare se determină caracteristicile de funcţionare ale unei pompe centrifuge în regim de turbină şi de pompă, şi se compară, în vederea obţinerii unor indicaţii necesare alegerii acelora care pot fi utilizate cu randamente bune.

Because of the ecological and environmental restrictions in energy production, the use of small hydropower resources will be economical in the future. Standard pumps could be considered a low cost alternative for hydraulic turbines used in small hydropower plants. The purpose of the paper is to determine the characteristic curves of a centrifugal pump operating as turbine. Tests showed that pumps operating backwards could compete with turbines, with high efficiency.

Keywords: small hydraulic turbine, centrifugal pump, turbine operation regime.

1. Introduction

Considering the actual ecological and environmental restrictions in energy production, the use of small hydropower resources will become more economic in the future. Standard pumps can be used in hydro systems when installing or upgrading small hydropower plants. Tests showed that pumps operating backwards could compete with turbines.

Pumps are produced in large series, so they are easy to find, for a large domain of head and discharge. The cost is low comparing to that of a turbine, which is designed for each site. The delivery time is short, the installation and service being easily available. The entire assembly pump-motor can be used as turbine-generator unit. Though, the simple design of a pump limits the discharge

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operational domain comparing to that of a turbine.

Creating reservoirs for water storage in small hydropower plants is not a viable option from economical point of view. Therefore it is important to choose a site where the available water is enough all the year, in order to have a certain value of discharge, as close as possible to the one at best efficiency point.

The head is also an important parameter. It is determined by the vertical distance between the intake and the tailrace, less the energy losses. The chosen pump must have the head and the discharge at best efficiency point as similar as possible with the site parameters.

The conditions for turbine operating regime of a pump (discharge and head) are very different from those when operating as a pump, but the efficiencies of the two operational regimes are pretty similar.

When choosing a pump, the rotational speed must be taken in consideration. Pumps with higher rotational speed are more compact and cheaper than equivalent pumps with a lower rotational speed, but they have a shorter operational life (the bearings and the seals need often changing). A low rotational speed pump is more expensive, but has a longer operational period and less maintenance service.

The electrical machine can be a synchronous or an asynchronous generator. Many pumps are delivered with an asynchronous motor directly connected, which can be used as self exciting generator. This is a cheaper solution than a separate pump operating as turbine and a synchronous generator. The asynchronous generators are easier to find, their rotor being more robust than that of synchronous generators.

The generator can be connected with the pump direct or by belt. If the connection is by belt it is possible to use a high rotational speed electric machine. If the connection to the generator is direct there are more advantages: low mechanical losses, easy setup, simple design – low cost, fewer bearings, longer bearings life (there are no axial efforts), easier maintenance.

2. Centrifugal pump operating as turbine

A proper alternative for small hydraulic turbines could be the centrifugal pumps. They have a simple design, are easy to maintenance being used for water supplying and irrigation. A centrifugal pump can have its own bearings or can be connected with an asynchronous motor which has the necessary bearings.

All types of centrifugal pumps – from those with radial flow up to those with axial flow – can be used as turbines. In this case the flow and the rotation directions are backwards, and the electric motor can be used as generator.

In turbine operation regime, a centrifugal pump can have good efficiency, which is not valid for a turbine operating as pump. This comes from two main reasons:

- the optimal angle of turbine blades at the inlet edge of the runner is bigger than the optimal angle of pump blades at outlet edge of the impeller (figure 1.);
- for an optimal design, at the same parameters – head and rotational speed, the pump impeller is bigger than the turbine runner.

The flow in turbine runner is accelerated, while in the pump impeller is decelerated because the turbine runner has more blades than the pump impeller and because the axial efforts in the runner are bigger than those in the impeller. This makes the turbine efficiency better than the one of the pump designed for the same operation parameters.

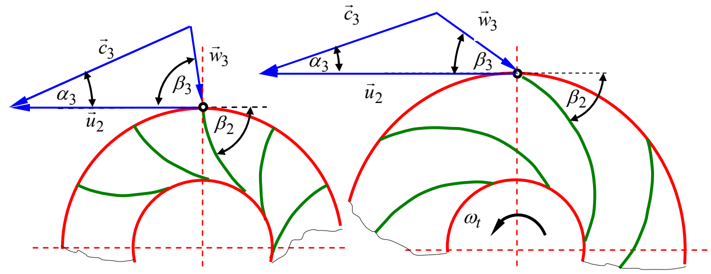

Fig.1. Comparison between turbine runner and pump impeller operating as turbine

The pump head is determined from Euler equation for pump operation

$$H^p = \frac{\eta_h(u_2 c_{2u} - u_1 c_{1u})}{g}, \tag{1}$$

where $\vec{u}$ is the peripheral velocity, $\vec{c}$ is the absolute velocity in the impeller and η_h is the hydraulic efficiency. The index 1 and 2 refer to the inlet edge and to the outlet edge of impeller blade, for pump operation.

In pumping regime, when operating at best efficiency point or close to it, the absolute velocity does not have initial rotational component, so c_{1u}=0. At the same operational conditions, because of the finite number of blades and because of the hydraulic losses, the flow angle at the impeller outlet is deflected from the blade angle, which leads to a decrease of c_{2u} and thus to a decrease in pump head (the effect of finite number of blades).

In pumping regime the deflection can be neglected. In the turbine regime of the pump is preferred that the velocity triangles at the inlet and outlet to be alike (but with the vectors in opposite directions) so the machine can have maximum efficiency in the pumping regime. This flow condition will lead to a flow with no shock at the impeller inlet and without swirling at the outlet, and in the same time to low hydraulic losses and high efficiency.

For a good decision when choosing a pump, it must be known the relation between discharge and head in pumping regime and in turbine regime, corresponding to the best efficiency point. The head for both regimes at best efficiency point, H_{bep}^{p} and H_{bep}^{t}, is determined with Euler equations

$$H_{bep}^{p} = \frac{\eta_h^p \left(u_2^p c_{2u}^p - u_1^p c_{1u}^p\right)}{g}, \tag{2}$$

$$H_{bep}^{t} = \frac{\left(u_2^t c_{2u}^t - u_1^t c_{1u}^t\right)}{g\eta_h^t} \tag{3}$$

If the velocity triangles are congruent, at the same rotational speed, relations (2) and (3) become

$$H_{bep}^{p} = H_{bep}^{t}\eta_h^t\eta_h^p. \tag{4}$$

It is also important to determine the rate between the discharge for turbine regime and the one for pumping regime, at best efficiency point, $Q_{bep}^{t} \big/ Q_{bep}^{p}$. The differences between velocity triangles at both operation regimes, for best efficiency point are shown in figure 2. The absolute flow angle, α_3, at the exterior diameter (in the point 2) is approximately the same for both regimes, considering the shape of the spiral case.

If in pumping regime the water leaves the blade with the relative flow angle, β_{3P}, smaller than the blade angle, β_2, (the effect of angle deviation), then when operating as turbine, the relative flow angle should be equal to the one of the blade, $\beta_2 = \beta_{3T}$ (to avoid energy losses due to the shocks). Thus, the radial flow velocity, $\vec{c}_{3,R}$, must be higher in turbine regime than in the pumping regime, and so must the flow.

The absolute flow angle at the interior diameter, α_0, (at point 1) is approximate equal for both operating regimes (90°), due to the zero initial rotational speed condition and to the no impeller outlet swirl condition.

Impeller blades are designed with β_1 angle, bigger than the relative flow angle β_{0P}, at maximum efficiency, in order to improve flow conditions.

In turbine regime, the relative flow angle β_{0T} is higher than the blade angle β_1, because of the outlet angle deviation. In compensation, the turbine flow regime must be smaller than the pumping flow regime, otherwise is possible to obtain a swirl at the impeller outlet, so reducing the efficiency.

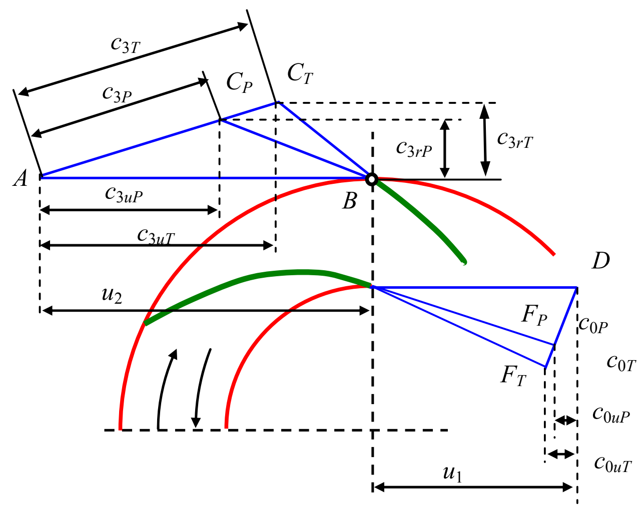

Fig.2. Velocity triangle at best efficiency point for pumping and turbine regimes

Considering all this, a pump operating backwards, has the best efficiency point at a higher discharge, comparing to pumping regime. For this reason the efficiency remains almost constant, or even gets higher in the turbine regime for the pumps with low specific speed.

The influence of secondary losses (due leaking and friction) over global efficiency is lower for higher shaft power in turbine regime.

3. Turbine operation regime

For turbine operation regime we measure the discharge, Q, the turbine upstream pressure, p_1^t, the turbine downstream pressure, p_2^t, the electric power output, $P_e{}^t$ and the turbine rotational speed, n. With this data we calculate turbine head, hydraulic power and turbine – generator efficiency, as follows

- turbine head

$$H^t = H_1{}^t - H_2{}^t = \frac{v_1^2 - v_2^2}{2g} + \frac{p_1^t - p_2^t}{\rho g} + \Delta z, \tag{5}$$

- hydraulic power

$$P_h^t = \rho g Q H^t; \tag{6}$$

- turbine – generator efficiency

$$\eta^t = P_e^t \big/ P_h^t \,. \tag{7}$$

Using the results for turbine operation regime we determine the head characteristic curve of the turbine, $H^t = H^t(Q)$, the hydraulic power characteristic curve, $P_h^t = P_h^t(Q)$ and the turbine – generator efficiency curve, $\eta^t = \eta^t(Q)$, as shown in figure 3.

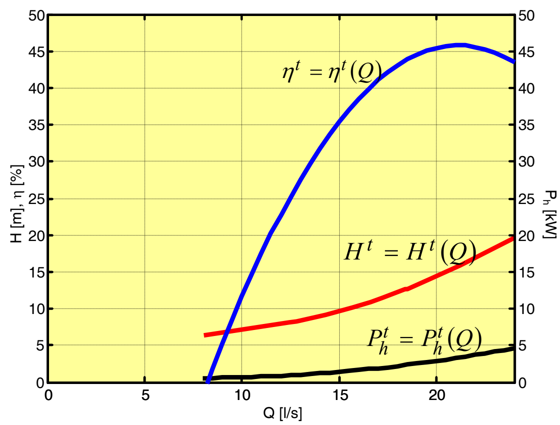

Fig. 3. Characteristic curves for turbine operation regime

4. Pump operation regime

When operating as a pump, we measure the discharge, Q, the inlet section pressure, p_1^p, the outlet section pressure, p_2^p, the consumed electric power, P_e^p and the pump rotational speed, n, in order to determine

- the pump head

$$H^p = H_2 - H_1 = \frac{p_2^p - p_1^p}{\rho g} + \frac{v_2^2 - v_1^2}{2g} + \Delta z \tag{8}$$

- the produced hydraulic power

$$P_h^p = \rho g Q H^p \,; \tag{9}$$

- the pump-motor efficiency

$$\eta^p = \left(P_h^p / P_e^p\right) 100 \ [\%]. \tag{10}$$

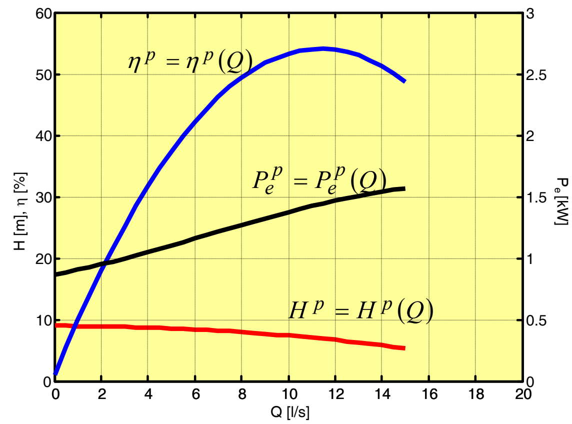

Fig. 4. Characteristic curves for pump operation regime

The head characteristic curve $H^p = H^p(Q)$, the power characteristic curve $P_e^p = P_e^p(Q)$ and the pump-motor efficiency curve $\eta^t = \eta^t(Q)$ for pump operation regime are presented in figure 4. A comparison between the head characteristic curves for turbine and pump operation regimes, the hydraulic power characteristic curve for turbine operation regime and the electric power characteristic curve for pump operation regime, and the global efficiency characteristic curves for turbine and pump operation regimes is presented in figures 5, 6 and 7.

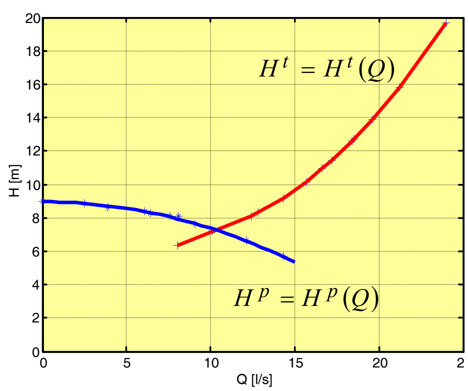

Fig.5. Head characteristic curves

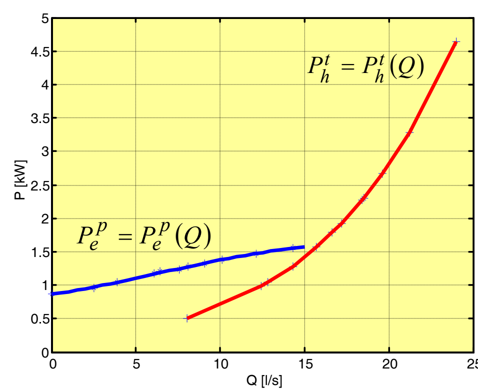

Fig.6. Power characteristic curves

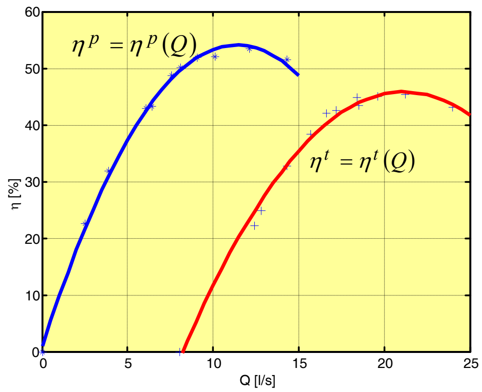

Fig.7. Global efficiency characteristic curves

5. Conclusion

In order to determine the characteristics of a pump in turbine regime we tested a small centrifugal pump, NC 80-65-160 (35m^3/h discharge, 7m head, 1.5kW electrical power motor, 1500rpm speed), installed backwards, produced by S.C. AVERSA. The characteristic curves: head $H(Q)$, power $P(Q)$ and efficiency $\eta(Q)$, are determined by testing the pump on the experimental set-up, for both operating regimes (pump and turbine).

The values for discharge Q^t and the head H^t at best efficiency point for turbine operation regime are almost double comparing to the parameters at best efficiency point for pump operation regime ($Q^t \cong 2Q^p$ and $H^t \cong 2H^p$). The efficiency is almost the same for both operation regimes, with 5 – 6% lower in turbine operation regime. This can happen due to the determination of the global efficiency of turbine-generator assembly instead of the single hydraulic machine and the use of an electric motor which operates with a lower power factor, so with higher copper losses. This solution is advantageous because the cost is consistently lower comparing to that of turbine, allowing reduced investment costs for small hydropower plants or hydropower plants with low power output.

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