Commentary on the Efficiency of Selected Structural Designs of Low Head Micro Hydraulic Power Plants

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Abstract

This article compares, first of all, structural designs of low head micro hydraulic power plants using Kaplan turbines. The authors have chosen this turbine class because siphon turbine systems suck the water up over the dams, can operate at temperatures below 0°C and are associated with less capital costs than turbine systems of other types. Besides it, mobile micro power plants can be easily assembled in a modular way and be installed on floating platforms thus providing appeal to a growing market niche. The authors discuss the issue of identity of possible structural designs of siphon micro hydraulic power plants in terms of their efficiency and selection of optimal pressure heads. They have developed a criterion of comparison of efficiency of various designs of siphon micro hydraulic power plants and provide evidence of virtual identity of selected structural designs with identical parameters in terms of efficiency. This criterion allowed them to make estimates in search for an optimal solution in terms of gross head (vertical distance between the highest and lowest water surface) versus net head (effective head available for power generation which is gross head less all the losses in the water conductor system including penstocks), which can be useful for designers of micro hydraulic power plants.

Keywords: potential capacity of small-scale hydro power, micro hydraulic power station, low head micro hydraulic power station, power generation efficiency, Kaplan turbine, siphon micro hydraulic power station, siphon

1. Introduction

According to estimates (Chang et al., 2004; Taylor, 2004; Kaygusuz, 2004) the worldwide technically exploitable hydro power potential of hydro energy is about 14000 TWh/year and the economically exploitable potential is about 8000 TWh/year, whereas the present global hydro power generation stands at account about 35 %; the rest can be contributed by small scale hydro power based on available small-scale energy sources. The latter include small rivers and rivulets flowing through the plains with only low gross heads not exceeding 1-4 meters (Date et al., 2012). Low-head micro hydraulic power stations are intended for such energy sources.

The existing structural designs of micro hydraulic power plants employ hydro turbines of all known types (Nasir, 2013). Many publications worldwide offer estimates of the efficiency of various kinds of turbines for available hydro energy sources. The analysis of this publications provides evidence that Kaplan-turbines (Stark et al., 2011) and Banki turbines (Sinagra et al., 2014) are the best suited for small pressure heads. Banki turbines have a number of disadvantages compared to Kaplan turbines. First of all, the speed at which a Banki turbine typically rotates is slower than that at which a Kaplan turbine typically rotates. Secondly, Kaplan turbines are fully submerged in water and Banki turbines interact with both water and air of the environment. This can result in icing of components of Banki turbines in winter. Therefore in countries with cold winters it is desirable to employ micro hydraulic power plants using Kaplan turbines installed in penstocks, which automatically suck water from the upper pool and convey it to the lower pool.

Penstocks used in micro hydraulic power plants generally come in two types: those with common pit design and those operating on the siphon principle. The first of them are associated with large capital costs because either the penstocks pass through the dam wall to supply water from the reservoir to the turbine or one has to build a diversion channel and both solutions require large costly civil works. In siphon power plants penstocks are situated over the dam and therefore are associated with lower capital costs. Besides it, mobile micro power plants can be easily assembled in a modular way and be installed on floating platforms thus providing appeal to a

growing market niche. That is why in this article it is only the structural designs of siphon micro hydraulic powers plants that are compared.

There exists a number of different structural designs of siphon micro hydraulic power plants. The most relevant of them into practice different from each other (Figure 1) location of turbines in the siphon.
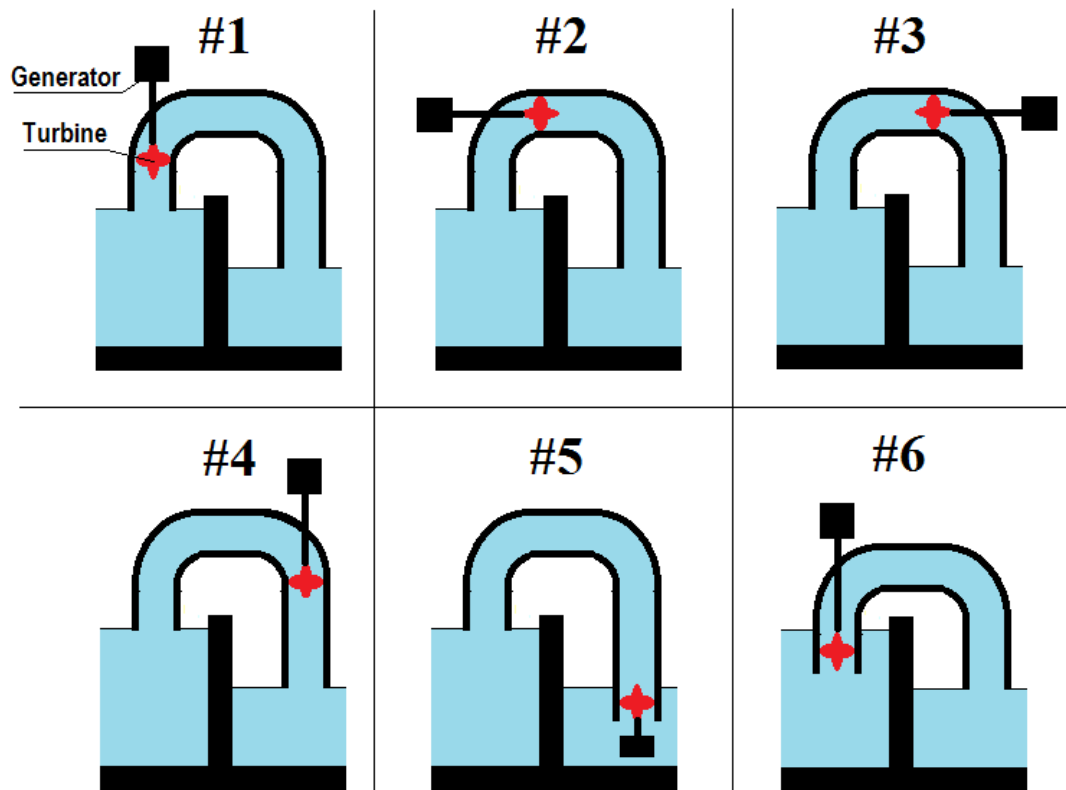

Figure 1. The constructive scheme of siphon micro hydraulic power plants

When choosing among these designs, the question arises as to whether they are identical in terms of efficiency of conversion of hydraulic energy into mechanical power on the turbine's shaft. Unfortunately, the literature scarcely sees this problem (at least the authors of this article failed to find publications, which would directly answer this question). Besides it, the determination of an optimal pressure head for a turbine is of great importance for practical designing of siphon micro hydraulic power plants. If such an optimal solution exists but is ignored by developers of a micro hydraulic power plant, the lacking efficiency must be compensated extensively, i.e. through an increased water intake thus causing increase in capital costs. We failed to find any discussion of this matter in open publications. When writing this article, we aimed at closing these knowledge gaps.

2. Method

When making estimates of efficiency of various structural designs of micro hydraulic power plants, we make the following basic assumptions:

- the water source of hydraulic energy is composed of the upper and lower pools;

- the volumes of both, the upper and the lower pools, are indefinitely large so that we can assume that the water levels in both pools remain unchanged during the operation of the siphon;

- the pressure on the surface of the pools is equal to the atmospheric pressure;

- the cross section of a siphon penstock does not change along its entire length, i.e. the area F of the cross section perpendicular to the axis of the penstock is constant ($F = const$);

- the siphon penstock is absolutely hard; it cannot be deformed;

- the working medium, i.e. water, is a viscous uncompressible fluid;

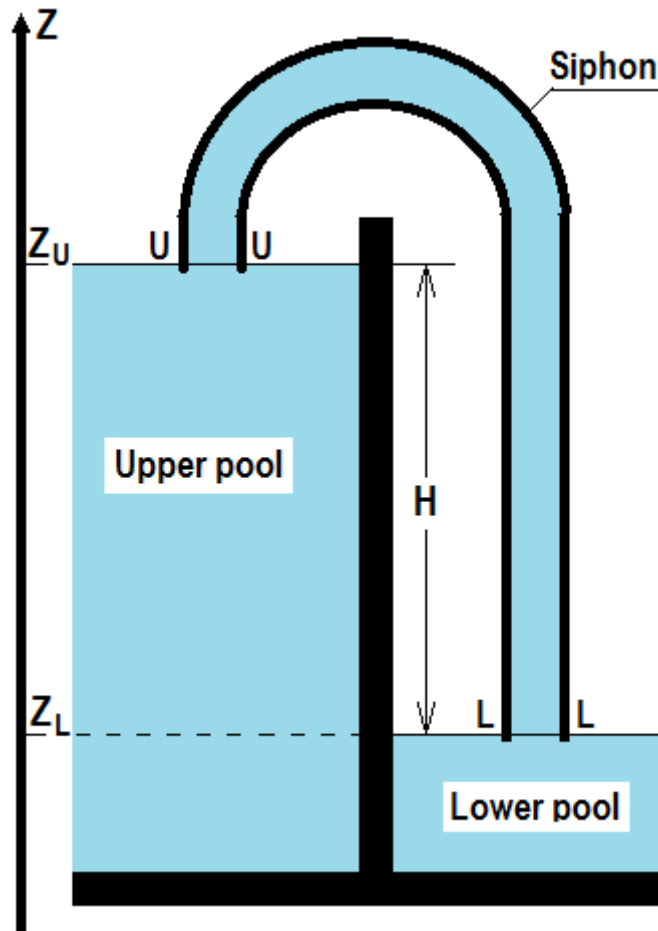- the flow of fluid (water) in the siphon penstock and turbine is axially symmetric.

Figure 2. Computational model for a siphon

In order to compare the efficiency of various structural designs of siphon micro hydraulic power plants we use a well-known equation for determination of the speed V of the flow of water inside the siphon (Yemtsev B.T., 1978; Kramarenko V.V., Savichev O.G., 2009; Landau L.D., Lifshitz E.M., 2006):

$$V^2 = 2 \times g \times H/(1 + \xi_S), \tag{1}$$

where g is the gravitational constant (acceleration of gravity), $H = Z_U - Z_L$ is the vertical distance between the highest and lowest water surface (difference between the water levels in the upper and lower pools) (see Figure 3) expressed in meters; *[ksi]*$_S$ is the sum of all friction coefficients of the fluid flow in all sections of the siphon penstock along its length including the friction at its inlet ("total friction"). In this article we do not make quantitative estimates of the hydraulic friction; that is why we do not refer to well-known methods of determination of hydraulic friction coefficients.

Equation (1) is usually derived from the Bernoulli equations for cross sections of the siphon penstock at the levels of the upper and lower pools with making well-known assumptions (Landau et al., 2003). However this method does not offer a comprehensive picture of operation of a siphon. Strictly speaking, a siphon cannot be regarded as a device, which is isolated from its environment, i.e. its operation cannot de depicted using only the assumption that the static pressure in the cross sections of the upper and lower pools remains unchanged and equal to the atmospheric pressure (in reality it is influenced by the environment). That is why in our research we use equation (1) derived for the computational model shown in Figure 3 based on the law of conservation of linear momentum (Lojcânskij, 2003; Landau & Lifshitz, 2006; Altschul et al., 1987), according to which the rate of change in the quantity of motion of a stream tube is equal to the sum of all forces exerted on its surface.
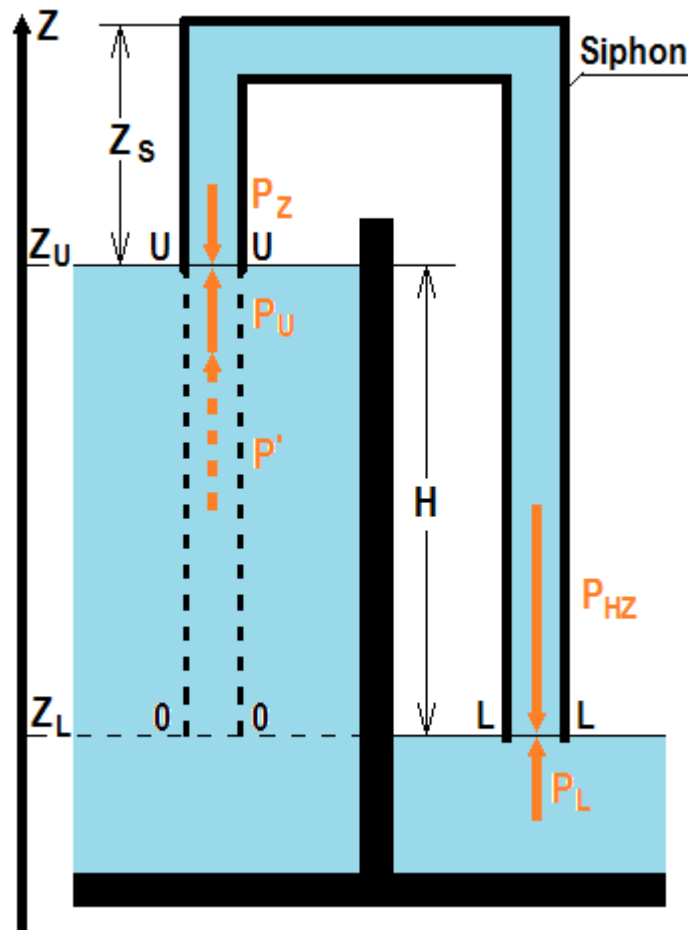
Figure 3. Scheme of the forces operating in the siphon

According to Figure 3, in the "upper reservoir- siphon system" the following forces are exerted on the cross section of the siphon penstock at the water level in the lower pool: from below – pressure of the power pool $P_L = p_L \times F = p_A \times F$, from above – pressure of the water column in the longer leg of the siphon $P_{HZ} = \rho \times (H + Z_S) \times F \times g$, where *[ro]* is the density of water. The following forces are exerted on the cross section of the siphon penstock at the water level in the upper pool: from below – pressure of water in the upper pool $P_U = p_U \times F = p_A \times F$, from above – pressure of the water column in the shorter leg of the siphon $P_Z = \rho \times Z_S \times F \times g$. The sum of all forced exerted on the water in the siphon in the direction of the low is equal to the following:

$$P = P_U - P_Z + P_{HZ} - P_L = p_A \times F - \rho \times Z_S \times F \times g + \rho \times (H + Z_S) \times F \times g - p_A \times F = \rho \times H \times F \times g \neq 0. \quad (2)$$

Here it is noteworthy to notice that the sum of all forces inside the siphon is not equal to zero $P \neq 0$. Such an imbalance of forces occurs at the start of the siphon effect and the force P found from equation (2) determines the initial acceleration of the water inside the siphon at the moment when the velocity of the flow is equal to zero. Inside the siphon a steady flow can occur only if the forces are fully balanced. Consequently, in case of a steady flow the absolute value of the force P' exerted from below to the cross section of the siphon at the level of water in the upper pool and pointing up (shown in Figure 3 as a dotted line) is equal to P. In Figure 3 dotted lines show the flow in a symbolic tube with a cross sectional area F that is a symbolic continuation of the tube of flow inside siphon to the region below the water level in the upper pool. The static pressure exerted on its lowest cross section 0-0 is equal to $P_0 = p_A \times F + \rho \times H \times F \times g$ and the speed of the flow at this cross section is equal to zero. In contrast to it, in the cross section at the water level in the lower pool the speed of the flow is equal to V, i.e. it is not equal to zero. Therefore the law of conservation of linear momentum for the stream tube between the cross sections 0-0 and L-L leads to the following equation:

$$M \times 0 - M \times V = -P_0 + P_Z - P_{HZ} + P_L + \xi_S \times \rho \times V^2 \times F/2 = -p_A \times F - \rho \times H \times F \times g + \rho \times Z_S \times F \times$$
$$g - \rho \times (H + Z_S) \times F \times g + p_A \times F + \xi_S \times \rho \times V^2 \times F/2 \tag{3}$$

where M is the mass flow rate of water through the siphon per second. After simplification of equation (3) with taking into account that $M = \rho \times V \times F$ we obtain the final formula for the flow rate in the form of equation (1). Besides equation (1) we use the well-known equation for power W transferred to the turbine by the flow of water, when the latter passes through the turbine (Krivchenko, 1978; Lomakin, 1966; Bulls & Emin, 1972). Here we make the following assumptions: the areas of the inlet and outlet cross sections perpendicular to the axis of the flow inside the siphon are equal being equal to the area of the siphon's cross section F; the velocities of the flow at the turbine's inlet and outlet are equal and point in the direction of the turbine's axis. Such assumptions are reasonable and admissible in case of Kaplan turbines and allow to write the equation for determination of the power N as follows:

$$N = M \times g \times H_T \tag{4}$$

where H_T is the (theoretically calculated) static pressure lost by the water flow in the turbine.

If we introduce such a concept as symbolic power of a siphon operating without a turbine $N_S = M_S \times g \times H$ and divide equation (4) by N_S, we shall obtain a relative parameter, which is called "energy utilization coefficient":

$$K_N = N/N_S = (V/V_S) \times (H_T/H) \tag{5}$$

This parameter proved to be convenient for quantitative comparison of the efficiency of structural designs of micro hydraulic power plants as it shows what fraction of the work done by the siphon can be converted into the useful work of the turbine.

In reality turbines do have certain hydraulic friction leading to loss of a part of the flow's kinetic energy. The loss of kinetic energy in the turbine divided by the mass of the water can be expressed both through hydraulic loss coefficient $[ksi]_T$, and additional head loss, i.e. hydraulic efficiency of the turbine $[atar]_T$:

$$\xi_T \times V^2/2 \equiv g \times H_T \times (1 - \eta_T) \tag{6}$$

If the turbine is situated inside the siphon penstock, one should take into account the losses inside the turbine when calculating the velocity of the flow inside the siphon. We suggest that they can be taken into account as head losses and therefore equation (1) can be rewritten as follows:

$$V^2 = 2 \times g \times [H - H_T - H_T \times (1 - \eta_T)]/(1 + \xi_S) \tag{7}$$

If the ratio of the gross pressure head ("theoretical turbine's head") to the net pressure heat ("siphon's head") defined as the pressure head utilization coefficient h, i.e. $h = H_T/H$, equation (7) can be rewritten as follows:

$$V^2 = 2 \times g \times H \times [1 - h \times (2 - \eta_T)]/(1 + \xi_S) \tag{8}$$

If we divide equation (8) by equation (1) written for the stream's speed V_S inside the siphon without the turbine, we obtain the following ratio:

$$V/V_S = \sqrt{1 - h \times (2 - \eta_T)} \tag{9}$$

If we substitute equation (9) into equation (5), we obtain the final expression for the energy utilization coefficient:

$$K_N = N/N_S = h \times \sqrt{1 - h \times (2 - \eta_T)} \tag{10}$$

3. Results of Comparison of Efficiency of Selected Structural Designs of Siphon Micro Hydraulic Power Plants

At the first stage of our research we compared the efficiency of structural designs #1, #2, #3 and #4 shown in Figure 1. The characteristic feature of all these designs is that the turbine is placed inside the siphon penstock between the cross section at the water level of the upper pool and the cross section at the level of the lower pool. That is why equation (7) is valid for structural designs #1, #2, #3 and #4 and a result equation (10) is also valid. In all of the above-mentioned cases the turbine configurations are similar; that is why we shall ignore possible insignificant differences in the values of the hydraulic loss coefficient $[ksi]_T$, associated with these structural

designs. Thus, when discussing the matter of identity in terms of efficiency of conversion of hydraulic energy into mechanical power on the turbine's shaft, we can state with little doubt that the above-mentioned designs are identical if proper parameters h и hydraulic efficiency of the turbine are identical.

As to the search for the turbine's optimal gross head, we have carried out a quantitative analysis of the two-parameter function $K_N(h, \eta_T)$. Figure 4 graphically shows the results of computation of $K_N(h, \eta_T)$ versus h varying in the range from 0 to 1 for a number of values of $\eta_T \leq 1$.
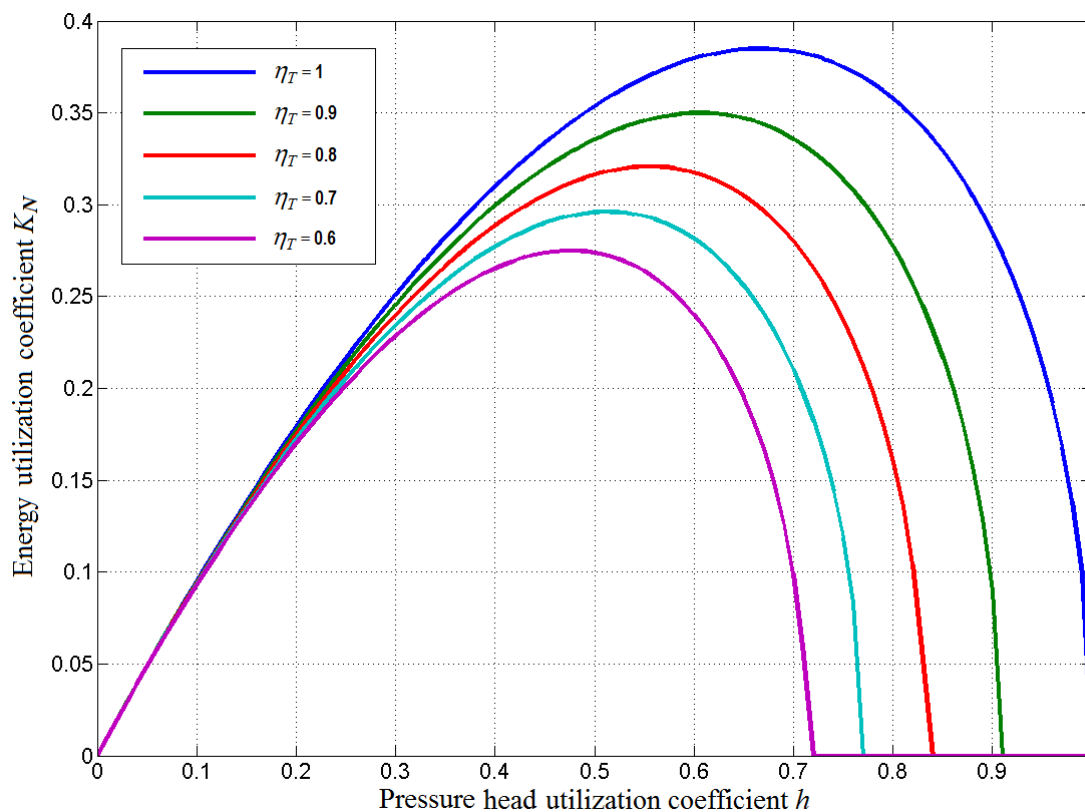

Figure 4. Graph of the function $K_N(h, \eta_T)$

Figure 4 provides evidence that the function w has a distinct maximum, which corresponds to the optimal value of h in terms of efficiency. In its turn, this optimal energy utilization coefficient is a function of the turbine's hydraulic efficiency and decreases as the turbine's hydraulic efficiency decreases. Another feature of the function $K_N(h, \eta_T)$ is that with decrease of the turbine's efficiency those values of h, for which there exist real values of K_N, rapidly decrease. In other words, in case of a siphon micro hydraulic power plant the turbine's efficiency not only poses a limit on the efficiency of the power plant, but also significantly influences the performance capability of the entire power plant. Thus, if we attempt to create a micro hydraulic power station using a turbine with $h = 0.8$ and efficiency about 70%, to all appearances, such a hydraulic power plant would fail to operate because, according to Figure 4, proper value of the energy utilization coefficient would not be a real number.

At the next stage of our research into the efficiency of micro hydraulic power plants shown in Figure 1 we compared structural design # 5 with structural designs #1, #2, #3 and #4 (which are identical to each other) under the condition of identity of the parameters h и the turbine's hydraulic efficiency. In contrast to designs #1, #2, #3 and #4 in case of design #5 the turbine is situated inside the long leg of the siphon at a depth Z_2 below the water level in the lower pool (Figure 5).
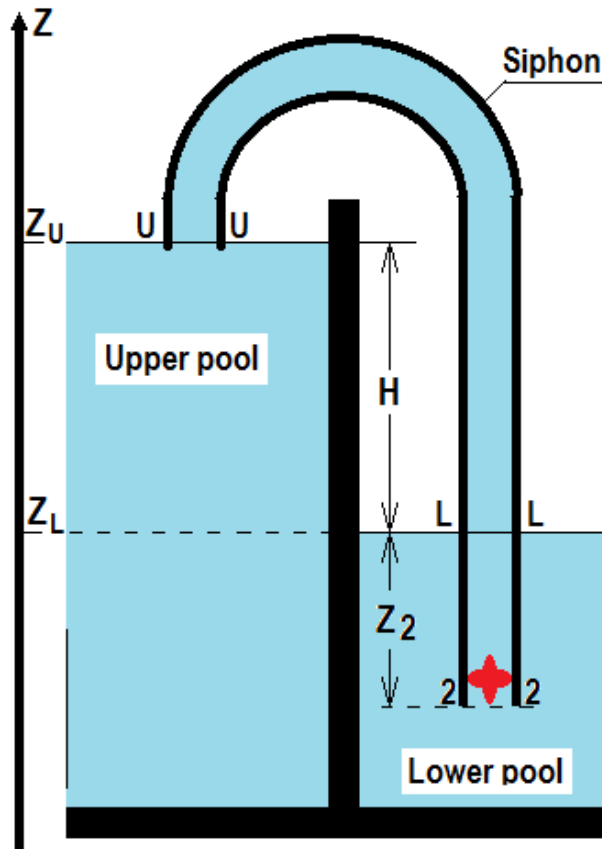
Figure 5. Design of micro hydraulic power plants with the turbine below the water level in the lower pool

It is noteworthy to notice that if the longer leg of the siphon ends below the water level of the lower pool, the pressure in the cross section of the penstock is always equal to the pressure exerted by the environment (Bachelor, 2004; Lojcânskij, 2003; Landau & Lifshitz, 2006; Altschul, 1982; Siov, 1968), i.e. to the pressure of the water at the depth Z_2 in the lower pool. In case of structural design #5 this pressure is equal to the sum $p_A + \rho \times Z_2 \times g$; such a pressure exerted on area F is counteracted by the weight of the water column inside the siphon penstock, whose height is equal to the sum $H + Z_S + Z_2$. The resulting force exerted on cross-section 2-2 is determined only by the atmospheric pressure and the height of the water column inside the siphon over the water level in the lower pool $P_2 = -\rho \times (H + Z_S) \times F \times g + p_A \times F$, similar to the situation of a siphon, whose longer leg in not immersed in the lower pool. Thus in case of design #5 with taking into account the law of conservation of linear momentum we can write the following equation for the change of linear momentum between sections 0-0 and 2-2:

$$M \times 0 - M \times V = -P_0 + P_Z - P_{HZ} + P_2 + \xi_S \times \rho \times V^2 \times F/2 \quad \text{or}$$

$$M \times 0 - M \times V = -p_A \times F - \rho \times H \times F \times g + \rho \times Z_S \times F \times g - \rho \times (H + Z_S) \times F \times g +$$
$$+ p_A \times F + \xi_S \times \rho \times V^2 \times F/2, \tag{11}$$

The members of equation (11) are virtually the same as the members of equation (3) for siphon designs #1, #2, #3 and #4. Therefore equation (10) is also valid for such a system. This fact can be regarded as a reason to agree that siphon design #5 is identical to siphon designs #1, #2, #3 and #4 in terms of efficiency if one ignores the difference in hydraulic losses due to different configurations of turbines. Here it is noteworthy to notice that there exist small differences between the values of the hydraulic friction coefficients *[ksi]*$_S$, and in practice in most cases the longer leg of the siphon ends below the water level in the lower pool. The results of computation of the optimal value of the pressure head utilization coefficient h_{OPT} is also valid for design #5.

At the next stage of our research into efficiency of structural designs of micro hydraulic power plants shown in Figure 1 we compared structural design #6 with structural designs #1, #2, #3, #4 and #5 (which are identical to each other) under the condition of identity of the parameters *h* и the turbine's hydraulic efficiency. In contrast to designs #1, #2, #3, #4 and #5 in case of design #6 the turbine is situated inside the short leg of the siphon at a

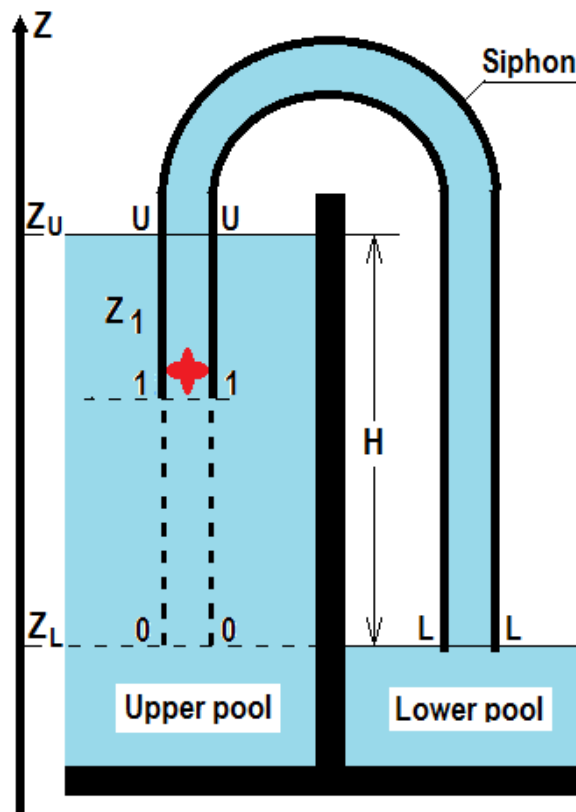depth Z_I below the water level in the upper pool (Figure 6).

Figure 6. Design of micro hydraulic power plants with the turbine below the water level in the upper pool

However, this fact does not change the computational model shown in Figure 3 and used by us, when we derive equation (1). This means that equation (1), as well as equation (3) and (10) are valid in case of design #6. With taking into account that in practice in most cases the shorter leg of the siphon ends below the water level in the upper pool and that the configuration of design #6 differs from configuration of design #1 only insignificantly, one can assume that design #6 is identical to siphon designs #1, #2, #3, #4 and #5 in terms of efficiency. Thus, the results of computation of the optimal value of the pressure head utilization coefficient h_{OPT} is also valid for design #6.

Finally one should notice that despite the virtual identity of #1, #2, #3, #4, #5 and #6 in terms of efficiency on the criterion of K_N, all other conditions being equal, they are not identical in terms of bubble cavitation in the flow channels of the micro hydraulic power plants. It is known that cavitation in the siphon tract breaks the flowing water stream and does a siphon inoperative. Thus, cavitation reduces the efficiency of the micro hydraulic power plant almost instantly to zero. Therefore, the assessment of the efficiency of design of siphon micro hydraulic power plant must necessarily be accompanied by an assessment of their cavitation stock. Cavitation begins to develop, when the static pressure in the pipe is lowered to the level of saturated vapor pressure of water p_{SV}. So we, like most researchers, to assess the safety of the cavitation stock using the difference between the pressure p_X in a cross-section X-X of the siphon pipe, normal its axis, and the pressure p_{SV}.

Reliably it is known (Landau & Lifshitz, 2006), that the static pressure p_L in the cross-section L-L at the level of the lower pool (Figure 3) equals of the atmospheric pressure p_A, when flow of water in the siphon pipe is stable. Then the full pressure of the p_L^* defined by: $p_L^* = p_A + \rho \times V^2/2$. Taking pressure p_L^* as a basis and taking a comparison plane as the surface of the lower pool compare, apply Bernoulli's equation to cross-sections L-L and X-X:

$$p_L^* = p_X^* - \xi_{XL} \times \rho \times V^2/2 + \rho \times Z_X \times g \qquad \text{or}$$

$$p_A + \rho \times V^2/2 = p_X + \rho \times V^2/2 - \xi_{XL} \times \rho \times V^2/2 + \rho \times Z_X \times g, \qquad (12)$$

where $[ksi]_{XL}$ is the summary coefficient of hydraulic losses in the siphon pipe between the cross-sections X-X

and L-L; Z_X is the excess of the cross-section X-X of the siphon pipe above the comparison plane, i.e. over the cross-section L-L. If in equation (12) instead of the pressure p_X put the saturated vapor pressure of water p_{SV}, the Z_X will give theoretical minimum excess of the cross-section X-X above the level of the lower pool, in which cavitation begins to develop:

$$\frac{p_X}{\rho \times g} = \frac{p_A}{\rho \times g} + \xi_{XL} \times \frac{V^2}{2 \times g} - Z_X \ \text{или} \ Z_X = \xi_{XL} \times \frac{V^2}{2 \times g} + \frac{p_A - p_{SV}}{\rho \times g}. \tag{13}$$

Let's call this the Z_X as the cavitation stock of the siphon H_C in the cross-section X-X. Then indicate the last member of the equation (13) as $\Delta H = \frac{p_A - p_{SV}}{\rho \times g}$ and using equation (1) to express V^2, we can think of the cavitation stock of the siphon as

$$H_C = \Delta H + H \times \xi_{XL}/(1 + \xi_S) \tag{14}$$

Note that the value of ΔH for purposes of comparison of design of the siphon micro hydraulic power plant can be considered permanent, since it depends only on the characteristics of the environment, which means we can take

$$H_C = const + H \times \xi_{XL}/(1 + \xi_S). \tag{15}$$

For comparison of cavitation stocks of siphon micro hydraulic power plants it is expedient to apply equation (15) with a cross-section X-X in the downstream cross-section of the turbine. Then, the value of H_C for each of us design of siphon micro hydraulic power plants is determined only by the value of *[ksi]*$_{XL}$, that is, total hydraulic losses in the siphon pipe after the turbine. Obviously, the farther from the lower pool is the turbine, believing against the flow, the greater will be the coefficient *[ksi]*$_{XL}$ and greater will be the H_C. In connection with this design #5 has minimum H_C as a value of the coefficient *[ksi]*$_{XL}$ almost zero and the cavitation stock has a value $H_C \approx \Delta H = \frac{p_A - p_{SV}}{\rho \times g}$. On the contrary, designs #1 and #6 has maximum H_C. It should be noted that should be designs #1 and #6 in the equation (12) take into account the increase of pressure p_X compared with the design #2 by the amount of the difference between the heights of the turbines. Thus $H_{C.6} > H_{C.1} > H_{C.2} > H_{C.3} > H_{C.4} > H_{C.5}$.

4. Discussion

In this article we have tried to express, first of all, we offer simple approaches to assessing energy efficiency of design of siphon low-head micro hydraulic power plant. It is the simplicity of the criterial equations and a small number of parameters included in them will allow designers to use them in the choice and optimization of design concept created by siphon low-head micro hydraulic power plant. The proposed criterion is the coefficient KN, expressed by equation (10) we believe accurately reflects the level of energy efficiency siphon new within the scope of the adopted assumptions.

5. Conclusion

The authors believe that they succeeded in this article to answer the questions of identity possible constructive schemes, siphon microhydroelectric power station in terms of energy efficiency and the optimal value of the head of the turbine. Provides a comparative assessment of energy efficiency design schemes and micro HPP siphon practical energy efficiency examined identity design schemes of identical design. The simple criteria allowed authors to get a quantitative assessment of the design pressure of the turbine, the optimum in terms of energy efficiency, and new rate in first approximation NPSH possible constructive schemes, siphon micro hydraulic power plants. This approach can be very useful to designers at the stage of choosing the best design of micro hydraulic power plants.

In addition, the authors believe that reviewed here design # 6 siphon micro hydraulic power plants (Figure 1) has two important advantages over other schemes. First, her acknowledgement reserve with the right choice of the design of the turbine can be maximized. Secondly, it has an important advantage over all other schemes with constructive and operational points of view.

Kaplan turbine is a reversible axial flow hydraulic machine with non-volumetric displacement. In other words it can operate both, as a turbine and as a pump without reversion of the direction of rotation of the rotor or reversion of the direction of the flow of water through it. Such a pump-turbine, when operating as a pump, can easily fill the siphon with water; so there is no need for using any external vacuum pump for starting the siphon effect. In case of all other structural designs such an external pump is needed. Therefore, in our opinion, structural design #6 best fits the needs of customers and is preferable when selecting a design of a low-head micro hydraulic power plant.

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