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Analiza parametrov reverzibilne črpalne francisove turbine

An Analysis of the Parameters of Reversible Francis-Type Pump Turbines

Milo Mrkić

Pri projektiranju reverzibilnih hidroelektrarn (RHE) mora imeti projektant na voljo čim bolj podrobne podatke o parametrih hidravličnih strojev, ki bodo v hidroelektrarno vgrajeni. Ta pogoj je še posebej pomemben, kadar gre za reverzibilne črpalno-turbinske agregate, saj morajo le-ti optimalno ustrezati režimu obratovanja v obeh smereh pretoka (črpalni in turbinski režim), da bi bila lahko vgrajena moč agregata optimalno izrabljena, tako pri polnjenju kakor tudi pri praznjenju zgornje akumulacije, in to v skladu z zahtevami elektroenergetskega sistema.

Ko se projektant loti projektiranja RHE, izbere po nomenklaturi ustrezen tip turbine, potem pa – upoštevajoč splošne karakteristike in nomenklaturne diagrame – določa osnovne parametre reverzibilne črpalke – turbine (RPT).

Glede na to, da je nomenklatura RPT pomanjkljiva in da obstajajo splošne karakteristike samo za omejeno število tipov, je primerno v začetni fazi projekta najprej definirati osnovne parametre RPT, v prvem koraku na temelju specifične vrtilne frekvence.

V prispevku je podanih nekaj rezultatov študij razpoložljive tehnične literature kakor tudi rezultatov teoretičnega dela, modelnih preiskav in preiskav v dejanskih razmerah, ki so bile ob sodelovanju avtorja prispevka opravljene na Katedri za izkoriščanje vodnih virov v Moskvi (Inštitut MISI).

(Ključne besede: projektiranje hidroelektrarn, turbine francis, črpalke reverzibilne, analize parameterske)

When designing pumping reservoir hydroelectric power stations the designer must have available detailed data on the parameters of the hydraulic machines that will be installed in the power plant. This is particularly important when reversible pumping-turbine units are installed, since they must best suit the working mode in both directions of the flow (pumping and turbine mode) so that the installed power of the unit is best utilized in the case of filling as well as emptying the upstream reservoir in accordance with the requirements of the public electric power system.

When the planning engineer starts to project, according to the nomenclature he chooses the appropriate type of turbine and then determines the basic parameters for the reversible pump-turbine (RPT) by using universal characteristics or graphical nomenclature.

Since the RPT nomenclature still does not exist and the universal characteristics only exist for a limited number of types it is appropriate at the initial stage of the design to define the basic parameters of the RPT, initially according to the specific number of revolutions.

In the work in this context some results of the study of available technical literature as well as the results of theoretical works are given. The results of model researches and researches in real conditions which were performed in "exploitation of water power" university department in Moscow (institute MISI), with participation of the author are also presented.

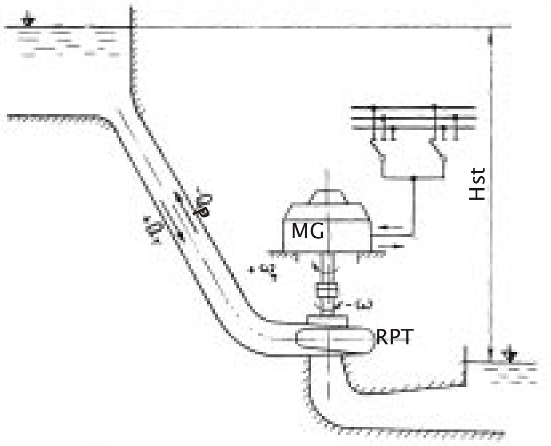
(Keywords: hydroelectric power stations, Francis turbines, reversible pump turbines, parameter analysis)

1 OSNOVNE KARAKTERISTIKE DELOVNEGA PROCESA RPT

Reverzibilni hidravlični stroj francisovega tipa ima značilnosti, ki se kažejo pri razliki njihovih osnovnih geometrijskih parametrov in obratovalnih karakteristik glede na klasične črpalke in turbine.

1 BASIC CHARACTERISTICS OF WORKING PROCESS RPT THEORY

A reversible hydraulic machine of the Francis type has some specific features whose basic geometrical parameters and operating characteristics differ from the conventional pump and turbine.



Sl. 1. *Shema reverzibilne hidroelektrarne*
Fig. 1. *Working plan of the reversible hydro power station*

Predvsem je treba vedeti, da sta padec vode reverzibilne hidroelektrarne (RHE) v turbinskem režimu (H_t) in višina RHE v črpalnem režimu (H_p) različna (sl. 1).

V turbinskem režimu je padec vode določen kot:

In particualr, it is necessary to realise that the fall of the reversible hydroelectric power plant in the turbine working mode (H_t) and the head in the pump working (H_p) mode are different (Fig. 1).

In the turbine working mode the fall is defined as follows.

$$H_t = H_{st} - h_t \quad (1),$$

v črpalnem režimu pa višina kot:

and the head in the pump working mode is defined as:

$$H_p = H_{st} + h_p \quad (2),$$

pri čemer so:
H_{st} - hidrostatični padec vode (višina)
h_t, h_p - hidravlične izgube.

Hidravlične izgube pri turbinskem obratovanju (h_t) niso enake izgubam v črpalnem režimu (h_p) obratovanja, ker sta pretoka v enem in drugem režimu v osnovi različna in ker tudi koeficienti lokalnih izgub v dovodnem in odvodnem sistemu niso enaki v turbinski in črpalni smeri pretoka vode (sl. 3).

Eulerjeva enačba za hidravlični reverzibilni stroj ima obliko:
- za turbinski režim

where:
H_{st} is the water head
h_t, h_p are the hydraulic losses

The hydraulic losses during turbine operation (h_t) are not the same as the losses in the pump working mode (h_p) because in principle the two flows in the first and second modes are different, and also because the coefficients of the local losses in the conduit and in the outflow system are not identical in the turbine direction and in the pump direction of the water flow (Fig. 3).

Euler´s equation for a reversible hydraulic machine has the following form:
- for the turbine working mode

$$\eta_t = \frac{u_1 \cdot c_0 \cdot \cos\ \alpha_0 - u_2 \cdot c_3 \cdot \cos\ \alpha_3}{g \cdot H_t} = \frac{\omega_t \left(\Gamma_1 - \Gamma_2\right)}{2 \cdot \pi \cdot g \cdot H_t} = \frac{\Delta\Gamma_t \cdot \omega_t}{2 \cdot \pi \cdot g \cdot H_t} \quad (3),$$

- za črpalni režim

- for the pump working mode

$$\eta_p = \frac{g \cdot H_p}{u_1 \cdot c_0 \cdot \cos\ \alpha_0 - u_2 \cdot c_3 \cdot \cos\ \alpha_3} = \frac{2 \cdot \pi \cdot g \cdot H_p}{\omega_p \left(\Gamma_1 - \Gamma_2\right)} = \frac{2 \cdot \pi \cdot g \cdot H_p}{\Delta\Gamma_p \cdot \omega_p} \quad (4),$$

pri čemer je:
$\Gamma = 2 \cdot \pi \cdot r \cdot c_u$ - obtok (cirkulacija),

$\omega = \frac{\pi \cdot n}{30}$ - kotna hitrost.

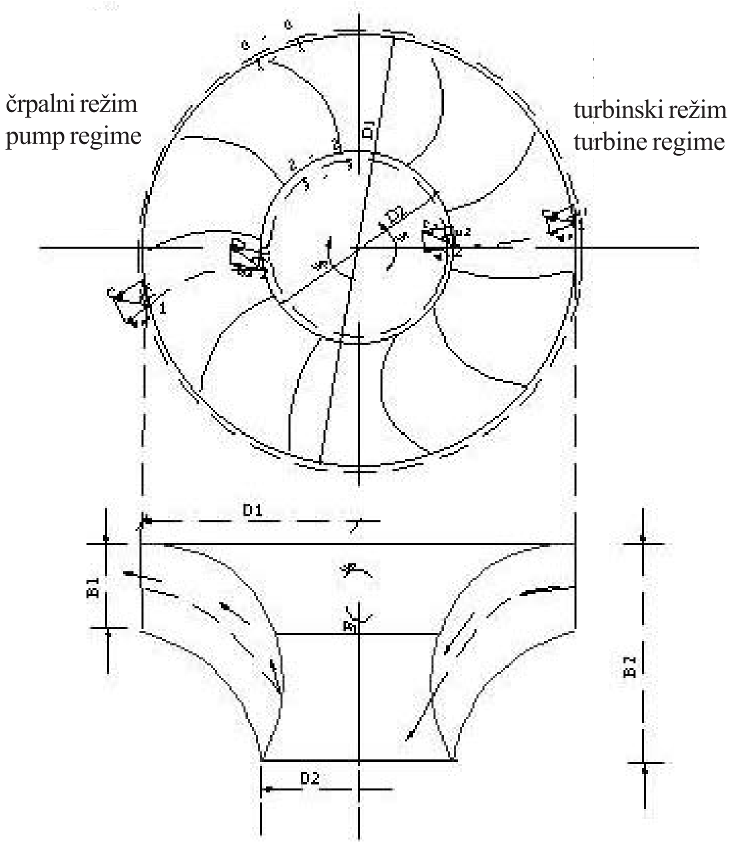
Indeks 1 se nanaša na vhod v delovno kolo, indeks 2 pa na izhod iz delovnega kolesa v turbinskem režimu. Ustrezajoči trikotniki hitrosti za

where:
$\Gamma = 2 \cdot \pi \cdot r \cdot c_u$ is the circulation

$\omega = \frac{\pi \cdot n}{30}$ is the angular speed

The index 1 refers to the inlet to the working wheel and the index 2 refers to the outlet from the working wheel in the turbine working mode. Figure 1 črpalni in turbinski režim so prikazani na sliki 2. Črte vhodnih (0-0) in izhodnih (3-3) površin delovnega področja kolesa so glede na črte vhodnih (1-1) in izhodnih (2-2) robov lopatic pomaknjene, da bi se izognili vplivu končnega števila lopatic na pretok vhodnih in izhodnih prerezov (sl. 2.).

shows the relevant triangles of the speeds in the pump and turbine working modes. The contours of the input (0-0) and outputs (3-3) working area surfaces are shifted with respect to the contours of the input (1-1) and output (2-2) edges of the moving blades to avoid influence at definitive numbers of blades on the flow in input and output cross sections (Fig. 2.).



Sl. 2. *Osnovni geometrijski parametri RPT francisovega tipa in trikotniki hitrosti v črpalnem in turbinskem režimu*

Fig. 2. *Basic geometrical parameters of the reversible pump - turbine of Francis type and triangles of speeds in the pump and turbine working modes*

Če vpeljemo oznaki: $\Gamma_1 - \Gamma_2 = \Delta\Gamma$ oziroma $\Gamma_1 - \Gamma_2 = \Delta\Gamma_p$, dobita enačbi (3) in (4) obliko:

If the terms $\Gamma_1 - \Gamma_2 = \Delta\Gamma$ and $\Gamma_1 - \Gamma_2 = \Delta\Gamma_p$ are introduced, equations (3) and (4) assume the following form :

$$\Delta\Gamma_t \cdot \omega_t = 2 \cdot \pi \cdot g \cdot H_t \cdot \eta_t \tag{3'}$$

$$\Delta\Gamma_p \cdot \omega_p = \frac{2 \cdot \pi \cdot g \cdot H_p}{\eta_p} \tag{4'}$$

oziroma razmerje:

and/or the ratio :

$$\frac{\Delta\Gamma_p \cdot \omega_p}{\Delta\Gamma_t \cdot \omega_t} = \frac{H_p}{\eta_p \cdot \eta_t \cdot H_t} \tag{5}$$

Če predpostavimo, da so izgube višine v turbinskem in črpalnem režimu enake in znašajo $h_t = h_p = 0{,}05 \cdot H_{st}$, potem je skladno z (1) in (2):

If it is assumed that the head losses in the turbine and the pump working modes are identical and amount to $h_t = h_p = 0.05 \cdot H_{st}$, the following applies in accordance with (1) and (2):

$$H_t = 0{,}95 \cdot H_{st} \quad ; \quad H_p = 1{,}05 \cdot H_{st}$$

2 ODVISNOST OSNOVNIH PARAMETROV RPT OD SPECIFIČNE VRTILNE FREKVENCE

Glede na to, da v obratovanjih RHE ni mogoče doseči enako velike stopnje izkoristka, je ugodneje imeti večjo stopnjo izkoristka v turbinskem režimu obratovanja kakor v črpalnem režimu (sl. 3), saj je cena vršne električne energije nekajkrat večja od cene električne energije v obdobjih najmanjše obremenitve elektroenergetskega sistema (sl. 4).

Predpostavimo, da sta $\eta_t = 0,93$ in $\eta_p = 0,90$, ti dve vrednosti vstavimo v enačbo (5), dobimo:

2 DEPENDENCE OF BASIC RPT PARAMETERS ON SPECIFIC NUMBER OF REVOLUTIONS

As an identical degree of efficiency cannot be reached with reversible hydroelectric power plants in both working modes (Fig. 3.) it is more convenient to have a higher degree of efficiency in the turbine working mode than in the pump working mode since the price of peak electric power is several times higher than the price of electric power during the period of least loading of the public electric power system, i.e. the price of free energy in the public electric power system (at the time when the reversible power plant operates in the pump working mode) Fig. 4.

If it is assumed accordingly that $\eta_t = 0,93$ and $\eta_p = 0,90$ and if these values are entered into equation (5), the following is obtained:

$$\frac{\Delta\Gamma_p \cdot \omega_p}{\Delta\Gamma_t \cdot \omega_t} = \frac{1,05 \cdot H_{st}}{0,90 \cdot 0,93 \cdot 0,95 \cdot H_{st}} \cong 1,3 \qquad (5')$$

Na podlagi analize (5′) lahko povzamemo, da moramo pri definiranju obratovalnih karakteristik reverzibilne črpalne turbine upoštevati naslednji predpostavki:

1. Če predpostavimo, da je razlika obtoka (cirkulacije) na vstopu in izstopu iz delovnega kolesa v turbinskem in črpalnem režimu enaka, to je $\Delta\Gamma_p = \Delta\Gamma_t$, potem mora biti vrtilna frekvenca v črpalnem režimu večje od vrtilne frekvence v turbinskem režimu ($\omega_p \cong 1,3 \cdot \omega_t$). V praksi pomeni to uporabo dvohitrostnih generatorjev (MG), ki imajo dve vrtilni frekvenci, vendar v nasprotnih smereh. Pri tem je treba pri prehodu iz enega v drugi režim obratovanja zamenjati število parov polov, ki so trenutno v obratovanju. Pomanjkljivost te rešitve je, da se cena generatorja, električnih aparatov, sistema avtomatike in zaščite v tem primeru poveča za 25 do 30 % ob hkratnem zmanjšanju stopnje izkoristka generatorja (sl. 1).
2. V primeru enake vrtilne frekvence $\left(\omega_p = \omega_t\right)$ je treba zagotoviti pogoj $\Delta\Gamma_p \cong 1,3 \cdot \Delta\Gamma_t$. To je mogoče doseči samo s predpostavko, da je premer delovnega kolesa v črpalnem režimu večji od premera delovnega kolesa v turbinskem režimu.

Za rešitev tega problema je uporabljenih več konstrukcijskih rešitev reverzibilnih hidravličnih strojev z dvema delovnima kolesoma (črpalnim in turbinskim), ki se s posebnimi napravami vključujeta v en ali drug režim obratovanja. Primeri take rešitve so reverzibilne črpalne turbine Isogyre (Švica), Hone (ČSSR) in druge. Znane so tudi konstrukcijske rešitve s samo enim vgrajenim delovnim kolesom, katerega premer se spreminja glede na vrsto obratovanja. Vse te rešitve pa so dokaj zapletene, zato pridejo v poštev samo za agregate manjših moči.

V svetovni praksi gradnje reverzibilnih agregatov velikih moči prevladuje uporaba

After analyzing equation (5') we come to the conclusion that for defining the operating characteristics of the reversible pump-turbine the following assumptions must be taken into account:

1. If it is assumed that the difference of circulation at the entry into and the exit from the working wheel in the turbine and pump working mode is identical, i.e. $\Delta\Gamma_p = \Delta\Gamma_t$, then the number of revolutions in the pump working mode must be greater than the number of revolutions in the turbine working mode ($\omega_p \cong 1,3 \cdot \omega_t$). In practice this imposes the use of two-speed generators (MG), having two rotating speeds in opposite directions. When switching from one to the other working mode it is necessary to change the number of pole pairs currently in operation. A disadvantage of this solution is that in this case the price of the generator, the electrical equipment and the automatic control system and protection is increased by 25–30 %, whereas the degree of efficiency of the generator is reduced (Fig. 1.).
2. In the case of an identical number of revolutions $\left(\omega_p = \omega_t\right)$ it is necessary to ensure the condition $\Delta\Gamma_p \cong 1,3 \cdot \Delta\Gamma_t$. This can only be reached with the assumption that the diameter of the working wheel in the pump working mode is greater than the diameter of the working wheel in the turbine working mode.

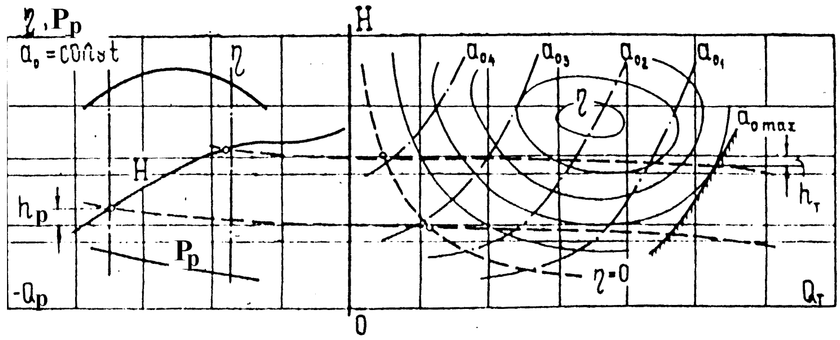
In order to solve this problem in practice several design solutions for reversible hydraulic machines with two working wheels (pump and turbine working wheel), activated in one or other operating mode by special devices, are used. Examples of such a solution are the reversible pump-turbines Isogyre (Switzerland) and Hone (Czechoslovakia). In addition, design solutions with one incorporated working wheel, whose diameter changes with respect to the working mode, are well known. However, all these solutions are rather complicated, and so they can only be considered for low-power generating units.

In most parts of the world, when building reversible power-generation units, the prevailing generatorjev z eno hitrostjo z izbiro primerne konstrukcijske rešitve delovnega kolesa hidravličnega reverzibilnega stroja z visokimi energijskimi lastnostmi v turbinskem in črpalnem režimu obratovanja (zaradi pravilne izbire profila lopatic delovnega kolesa in lopatic vodilnika).

Toda iz navedenih razlogov tudi v tem primeru ni mogoče doseči optimalnih karakteristik črpalnega im turbinskega režima. To je razvidno s slike 3, na kateri so v koordinatah Q - H predstavljene tipične delovne karakteristike črpalke - turbine, pri čemer ima pretok v črpalnem režimu negativni predznak. Posebno konstrukcijo je razvil prof. Krivčenko [6]. Ta ima zaradi učinka delno pomičnih lopatic delovnega kolesa skoraj optimalno vrtljivo rešetko delovnih lopatic v turbinskem in črpalnem režimu.

concept is the use of one-speed generators with the selection of a compromise design solution of the working wheel of the hydraulic reversible machine with high-power properties in the turbine and pump working mode (as a consequence of the correct selection of the contour of the working wheel blades and flow device blades).

However, for the above reasons it is also not possible in this case to achieve the optimum characteristics of the pump and turbine working mode. This can be seen in Figure 3, showing in coordinates Q – H the typical working characteristics of the pump-turbine where the flow in the pump working mode has a negative sign. A special structure of RPT was developed by Prof. Krivchenko, who has a nearly optimal rotation grating of the working blade in the turbine and pump modes, on the basis of the effect at partly moveable mobile blades of the working wheel (1).



Sl. 3. *Tipične delovne karakteristike francisove RPT v obeh režimih obratovanja*

Fig. 3. *Typical operating characteristics of the Francis reversible pump-turbine in both working modes*

Na sliki 3 je karakteristika črpalnega režima prikazana za primer nspremenljivega odprtja vodilnika (a_0), kakor je to na RHE običajno. Sprememba a_0 malo vpliva na vrednost pretoka, moč in stopnje izkoristka, odstopanje od optimalne vrednosti a_0 pa povzroča pojav precejšnjih utripov tlaka v pretočnem prostoru turbine. Za turbinski režim so podane krivulje nespremenljivih odprtij vodilnika a_0 in stopnje izkoristka do $\eta = 0$, to je do režima pobega turbine. Povečanje višine z namenom prehajanja delovnega področja na področje optimalnega izkoristka turbine pomeni hkrati prehod na področje zelo majhnih pretokov in nizkih stopenj izkoristka v primeru črpalnega režima (sl. 3).

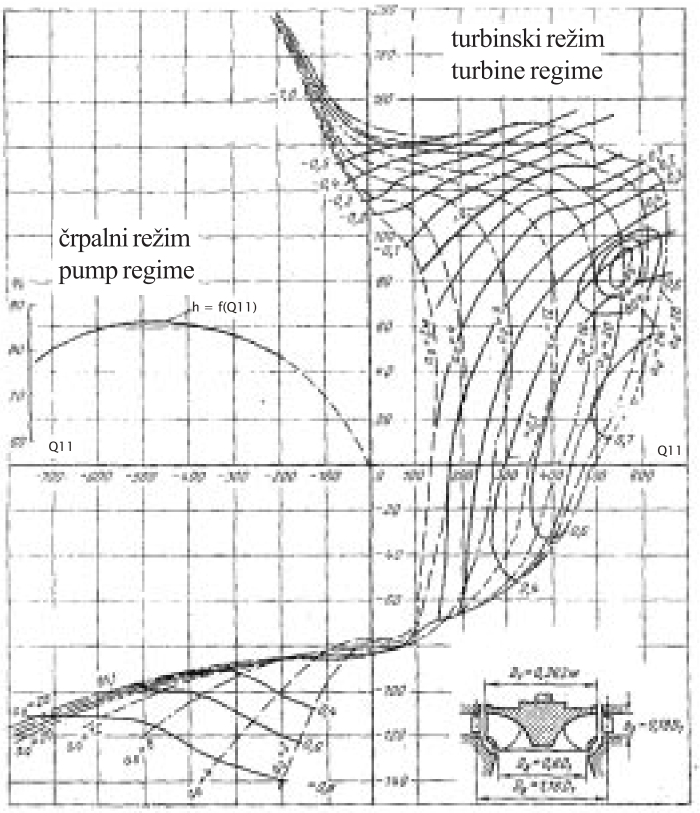
Nomenklatura reverzibilnih črpalnih turbin je pomanjkljiva, obstajajo pa glavne univerzalne karakteristike za zelo omejeno število tipov. Toda, glede na to, da se dandanes projektira vrsta RHE za zelo širok pas višin od 100 do 1200 metrov, se pokaže potreba po določanju osnovnih parametrov črpalnih turbin, v odvisnosti od teh parametrov pa tudi potreba po določanju samih RHE.

Na današnji stopnji raziskav lahko izbiro reverzibilnih črpalnih turbin opravljamo na temelju

In addition, Figure 3 shows the characteristic pump working mode for the case of a constant cross section of the flow device (a_0), as is usual for reversible hydroelectric power plants, since the change $a_{0\ only}$ slightly influences the value of the flow, the power and the degree of efficiency, and the deviation from the optimum value a_0 results in considerable fluctuations of the pressure in the turbine flow space. For the turbine working mode the isoclines of constant cross section of the flow device a_0 and of the degree of efficiency up to $\eta = 0$ i.e. up to the turbine over speed are given. The increase of the head, aimed at the working area passing into the range of optimum efficiency of the turbine, simultaneously implies passing into the range of very small flows and low degrees of efficiency in the case of the pump working mode (Fig. 3).

The parts lists of reversible pump-turbines are not yet available, whereas the principal universal characteristics for a very limited number of types are available. However, considering the fact that nowadays the type of reversible hydroelectric power plants for a very wide range of heads from 100 to 1200 m is designed, the need for determining the basic parameters of the pump-turbines and, depending on those parameters, the need for determining the reversible hydroelectric power plant itself are imposed.

At today's level of research the selection of reversible pump-turbines can be made on the basis of



Sl. 4. *Glavna splošna karakteristika RPT*
Fig. 4. *Main universal characteristic of RPT (1)*

sistematizacije in analize statističnih podatkov sedanjih RHE, ki so že v obratovanju ali pa so v fazi projektiranja oziroma gradnje.

Na inštitutu MISI v Moskvi, na Katedri za izkoriščanje vodne moči, je bila pod vodstvom prof. Aršenevskega in ob sodelovanju avtorja tega prispevka opravljena analiza več ko 40 reverzibilnih agregatov različnih zahodnih izdelovalcev. Ob tej priložnosti je bila ugotovljena naslednja odvisnost specifične vrtilne frekvence francisovih reverzibilnih črpalnih turbin v turbinskem režimu:

the systematization and analysis of statistical data from the reversible hydroelectric power plants that are already in operation, being designed, or being built.

At the MISI institute in Moscow, in the Department of Utilization of Water Power, an analysis of more than 40 reversible power generation units from different Western manufactures was made under the leadership of Professor Arshenevski, in cooperation with the author of this paper. On that occasion the following dependence of the specific number of revolutions of the Francis reversible pump-turbines in the turbine working mode was found:

$$n_{s_{RPT}} = \frac{n \cdot \sqrt{1,36 \cdot P}}{H_{t\max} \sqrt[4]{H_{t\max}}} = \frac{1212}{H_{t\max}^{0.4}} \qquad (6),$$

pri čemer so:
n - vrtilna frekvenca, min^{-1}
P - največja moč, kW
H_t - največji turbinski padec RHE, m.

where
n is the rated number of revolutions (min^{-1})
P is the maximum power (kW)
H_t is the maximum turbine fall on the reversible hydroelectric power plant (m)

Z analizo podatkov [1] se dobi naslednja enačba:

If these facts are processed (1) the following ratio is obtained:

$$n_{s_{RPT}} = \frac{1000 \div 1300}{H_{t\max}^{0,4}} \qquad (6').$$

Analiza parametrov reverzibilnih hidravličnih strojev nekaterih RHE v nekdanji ZSSR in v ZDA je pokazala, da obstaja tendenca povečanja specifične vrtilne frekvence, zato je bolj

However, an analysis of the parameters of reversible hydraulic machines on some reversible hydroelectric power plants in the former USSR and the USA showed that there is a tendency towards an primeren naslednji izraz:

increase in the specific number of revolutions, therefore the following relation is more appropriate:

$$n_{s_{RPT}} = \frac{1200 \div 1500}{H_{t\max}^{0.4}} \qquad (6'').$$

Zanimivo je primerjati specifično vrtilno frekvenco običajnih turbin (n_{st}) in reverzibilnih črpalnih turbin $(n_{s_{RPT}})$. Za klasične HE s francisovimi turbinami lahko ta parameter izrazimo kot funkcijo imenskega padca [3]:

It is interesting to make a comparison of the specific number of revolutions of a conventional turbine (n_{st}) and that of reversible pump-turbines $(n_{s_{RPT}})$. For conventional hydroelectric power plants with Francis turbines this parameter can be expressed as a function of the rated fall:

$$n_{st} = \frac{2300}{\sqrt{H_{opt}}} \qquad (7).$$

Za klasične HE velja razmerje:

On the other hand, the following ratio applies for conventional hydroelectric power plants:

$$\frac{H_{opt}}{H_{\max}} = 0,78 - 0,95$$

Če vzamemo srednjo vrednost $H_{opt}/H_{max} = 0,88$, dobi enačba (7) obliko:

If the mean value $H_{opt}/H_{max} = 0.88$ is adopted, equation (7) assumes the following form:

$$n_{st} = \frac{2070}{\sqrt{H_{\max}}} \qquad (8).$$

Razmerje med specifično vrtilno frekvenco reverzibilnih in klasičnih francisovih turbin tako izračunamo z enačbo:

The ratio of the specific number of revolutions of the reversible and conventional Francis turbines gives the following relation:

$$\frac{n_{s_{RPT}}}{n_{st}} = 0,58 H_{t\max}^{0.1} \qquad (9).$$

Na podlagi H_{tmax} in P lahko po enačbi (6) določimo vrtilno frekvenco turbine:

As H_{tmax}, and P are known, the number of turbine revolutions can be determined according to equation (6):

$$n = 1040 \frac{H_{t\max}^{0.5}}{\sqrt{P}} \qquad (10).$$

Dobljeno vrednost zaokrožimo na najbližjo sinhrono vrtilno frekvenco.

Na sliki 5 so podani rezultati analize odvisnosti glavnih izmer delovnih koles nekaterih že izvedenih reverzibilnih črpalnih turbin (RPT) pri H_{tmax} od specifične vrtilne frekvence $n_{s_{RPT}}$.

Odvisnost enotske vrtilne frekvence $n_{11} = f\left(n_{s_{RPT}}\right)$ lahko na temelju izvedene analize priporočimo v obliki:

The value obtained is approximated to the nearest synchronous number of revolutions.

Figure 5 gives the results of the analysis of dependence of the main working wheel dimensions of some reversible pump-turbines already in operation, with H_{tmax} on the specific number of revolutions $n_{s_{RPT}}$.

The dependence $n_{11} = f\left(n_{s_{RPT}}\right)$ can be recommended in the following form on the basis of the analysis carried out:

$$n_{11} = 82 + 0,05 \cdot n_{s_{RPT}} \qquad (11).$$

V tem primeru bo premer delovnega kolesa:

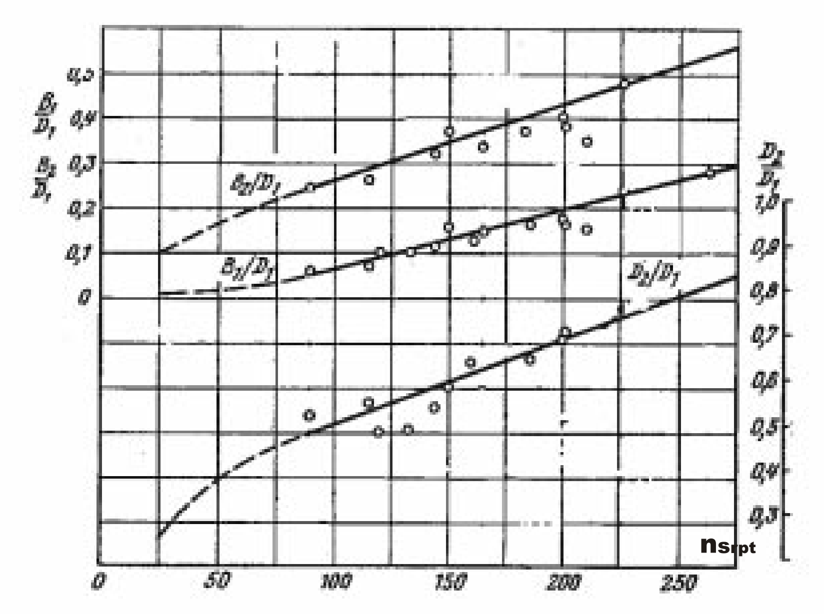
In this case the working wheel diameter will be:

$$D_1 = \frac{n_{11}\sqrt{H_{t\max}}}{n} = \frac{\left(82 + 0,05 \cdot n_{s_{RPT}}\right)\sqrt{P \cdot H_{t\max}}}{1040 \cdot H_{t\max}^{0.85}} \qquad (12).$$

Če dobljeno vrednost D_1 zaokrožimo na višjo vrednost do 0,1 m, moramo preveriti, ali smo dobili največjo višino v črpalnem režimu. Iz teorije črpalk je poznano razmerje [5]:

If the obtained value D_1 is approximated to a higher value of up to 0.1 m it is necessary to check whether the maximum head in the pump working mode has been obtained. The following ratio is known from the theory of pumps:

$$H_{p\max} = K\frac{u_1^2}{2g} = \frac{K}{2g}\left(\frac{\pi \cdot D_1 \cdot n}{60}\right)^2 \qquad (13),$$



Sl. 5. *Odvisnost izmer delovnega kolesa RPT od n_s [2]*
Fig. 5. *Dependence of RPT working wheel dimensions on n_s [2]*

pri čemer je K=0,8 - 0,9, iz tega izhaja:

where K=0.8–0.9, therefore:

$$H_{p\max} = \left(0,000111 - 0,000126\right) \cdot n^2 \cdot D_1^2 \quad (14).$$

Tako dobimo premer D_1, ki ne sme biti manjši od:

Thus the diameter D_1 is obtained, which must not be smaller than:

$$D_1 = \frac{\left(89 - 95\right)\sqrt{H_{p\max}}}{n} \quad (15).$$

Obdelava statističnih podatkov že izdelanih reverzibilnih hidravličnih strojev je pripeljala do izkustvene odvisnosti enotnega pretoka Q_{11} (l/s) pri obratovanju v turbinskem režimu pri H_{tmax} v obliki [2]:

The processing of statistical data on reversible hydraulic machines that are already built has led to the experimental dependence of the unit flow Q_{11} (l/s) in the case of maximum operation in the turbine working mode in the following form [2]:

$$Q_{11} = \left(0,008 - 0,012\right) n_{s_{RPT}}^2 \quad (16).$$

Po drugi strani pa lahko vrednost Q_{11} določimo po enačbi za specifično vrtilno frekvenco [2]:

On the other hand, the value Q_{11} can be determined according to the equation for the specific number of revolutions:

$$n_s = 3,65 \cdot n_{11} \cdot \sqrt{Q_{11} \cdot \eta} \quad (17),$$

pri čemer ima Q_{11} mero m^3/s. Če vzamemo vrednost η = 0,9, dobimo povezavo:

where Q_{11} has the dimension m^3/s. If the value η = 0.9 is assumed, the following relation is obtained:

$$Q_{11} = \left(0,029 - 0,032\right) n_{s_{RPT}}^{1.8} \quad (18).$$

Glede na to je premer delovnega kolesa upoštevajoč (11) in (16):

So the diameter of the working wheel considering (16) and (11) is:

$$D_1 = \frac{1,166 \cdot n_{11} \cdot \sqrt{P}}{n_{s_{RPT}} \cdot H_{t\max}^{3/4}} \quad (19).$$

Premer delovnega kolesa se lahko izračuna tudi iz enačbe za moč in enotski pretok Q_{11} z upoštevanjem enačb (16) in (18):

On the otherr hand, from the equation for the power and unit flow Q_{11} and by taking into account the relation (16) and (18) it is also possible to calculate the diameter of the working wheel:

$$D_1 = \sqrt{\frac{P}{9,81 \cdot H\sqrt{H} \cdot Q_{11} \cdot \eta}} \quad (20).$$

Bodimo pozorni na koeficiente v števcu enačbe (15). To so vrednosti n_{11} v črpalnem režimu za H_{pmax}, kjer je $n_{11t} > n_{11p}$, saj je $H_{t\max} < H_{p\max}$. Ob znani vrednosti D_1 lahko izhodni premer delovnega kolesa (v turbinskem režimu) D_2, višino dovodnega aparata B_1 in celotno višino delovnega kolesa B_2 (sl. 2) določimo po diagramu na sliki 5.

Let us look at the coefficients in the numerator of equation (15). These are the values n_{11} in the pump working mode for H_{pmax}, where $n_{11t} > n_{11p}$, because of $H_{t\max} < H_{p\max}$. Since we know the value D_1 the outlet diameter of the working wheel D_2 (with respect to the turbine working mode), the head of the flow device B_1 and the total head of the working wheel B_2 (Fig. 2) can be determined according to the graphics in Figure 5.

3 DOLOČITEV SESALNE VIŠINE RPT

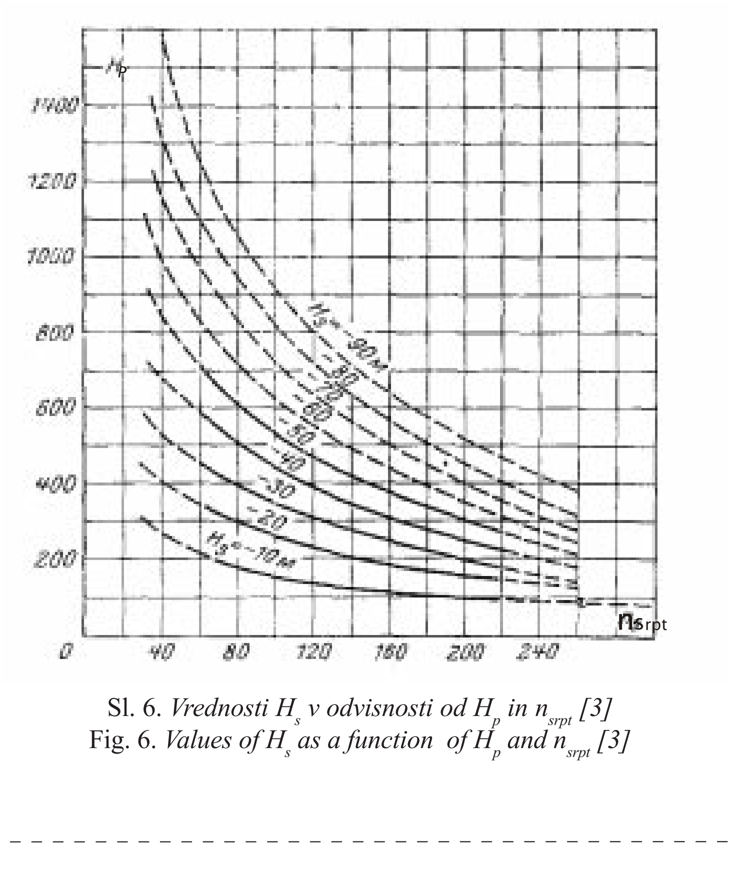
3 DETERMINATION OF THE INTAKE ALTITUDE RPT

Eden izmed najpomembnejših parametrov, ki odločujoče vpliva tudi na zasnovo RHE, je lega delovnega kolesa reverzibilnega stroja glede na najmanjšo koto vode v spodnji akumulaciji oziroma sesalna višina reverzibilnih hidravličnih strojev. Za RHE je značilno, da je treba sesalno višino določiti izhajajoč iz pogojev obratovanja v črpalnem režimu. Koeficient kavitacije je v tem režimu večji kakor v turbinskem režimu. Tako je na primer za RPT Kijevske RHE kavitacijski koeficient v optimalnem turbinskem obratovanju σ_T = 0,12 v črpalnem pa σ_P = 0,33. To je razlog, da je črpalka v primeru trojnih agregatov vedno vgrajena pod turbino.

Po drugi strani pa je vgradnja delovnega kolesa na globinah 20 do 50 m ali več odvisna od zasnove podzemeljske ali polpodzemeljske strojnice RHE, kar ima za posledico povečanje

One of the most important parameters that also decisively influences the concept of reversible hydroelectric power plants is the altitude position of the working wheel of the reversible machine in relation to the minimum level of the downstream reservoir and/or the suction height of the reversible hydraulic machines. It is characteristic of reversible hydroelectric powers plants that it is necessary to determine the suction height starting from the operating condition in the pump working mode, taking into account that the cavitations coefficient is greater in this working mode than in the turbine working mode. In this way for the RPT of the Kiev RHEPP the coefficient of cavitations in the optimum turbine regime σ_T = 0.12 in the turbine working mode and σ_P = 0.33 in the pump working mode.

On the other hand, the requirement for the installation of the working wheel at 20–50-m depths or more, conditions the concept of an underground or semi-underground powerhouse of the reversible hydroelectric



Sl. 6. *Vrednosti H_s v odvisnosti od H_p in n_{srpt} [3]*
Fig. 6. *Values of H_s as a function of H_p and n_{srpt} [3]*

dolžine odvodnih sistemov. Ti sistemi so zaradi precejšnjih nihanj nivoja vode v spodnji akumulaciji pogosto izvedeni pod tlakom [4].

Če analiziramo podatke že zgrajenih RHE in RHE v fazi projektiranja, opazimo, da sesalna višina v mnogočem prekaša vrednosti, ki jih srečujemo pri običajnih turbinah z enako vrtilno frekvenco. V preglednici 1 so prikazane sesalne višine nekaterih že izvedenih in karakterističnih RHE [3].

power plant, which results in the increase of the length of outflow systems which are frequently designed pressurized due to considerable fluctuation of the water level in the downstream reservoir [4].

If the data on the reversible hydroelectric power plants already constructed or under construction are analyzed it can be seen that the suction height, in many aspects, exceeds the values occurring on the conventional turbines with an identical number of specific revolutions. Table 1 shows the suction heights of some characteristic reversible hydroelectric power plants already constructed [3].

Preglednica 1
Table 1

RHE Reversible hydroelectric power plant	Država Country	Sesalna višina Suction height (m)	Črpalna višina Head (m)	Pretok v črpalni smeri Flow rate at pumping (m^3/s)	Vrtilna frekvneca Number of revolutions (min^{-1})
Vianden	Luksemburg Luxemburg	- 26,0	300	71,1	333,30
Krauchan	Velika Britanija Great Britain	- 45,2	358	28,6	500,00
Numapara	Japonska Japan	- 55,0	528	50,0	375,00
Katrua-Pon.	Belgija Belgium	- 20,0	259	46,0	300,00

V prvem približku lahko sesalno višino RPT določimo po diagramu na sliki 6.

Analitična enačba za izračun sesalne višine je:

The suction height RPT can be determined according to figure 6 as a first approximation.

The analytical expression for the calculation of the intake altitude is:

$$H_s = 10 - \frac{\dot{\nabla}}{900} - h_{us} - k_\sigma \cdot H_p \cdot \sigma_p \qquad (21),$$

pri čemer so:

$\dot{\nabla}$ - absolutna kota vgradnje delovnega kolesa,
h_{us} - hidravlične izgube v odvodnem sistemu pod pritiskom,
σ_p - koeficient kavitacije v črpalnem režimu,
H_p - črpalna višina RHE,
$k\sigma$ - koeficient rezerve.

Glede na to, da obstaja zelo majhno število modelnih univerzalnih karakteristik reverzibilnih hidravličnih strojev, ki vsebujejo podatke o koeficientu kavitacije, koeficienta kavitacije ni mogoče definirati na način, kakor je to običajno pri običajnih turbinah. Zato ostaja možnost uporabe empiričnih enačb. Iz teorije črpalk je znana enačba Rudneva za kritični koeficient kavitacije črpalk v optimalnem režimu obratovanja, in sicer v odvisnosti od vrtilne frekvence [6]:

where:

$\dot{\nabla}$ is the absolute elevation of the installation of the working wheel
h_{us} is the hydraulic loss in the pressurized outflow system
σ_p is the cavitations coefficient in the pump working mode
H_p is the head of the reversible hydroelectric power plant
$k\sigma$ is the coefficient of reserve

As there are only a small number of universal characteristics for the model of reversible hydraulic machines and, especially with information about the coefficient of cavitations, the cavitations coefficient cannot be defined in the usual way for conventional turbines. Therefore, the possibility for using on experimental equation remains. Rudnev's equation for the critical coefficient of the pump cavitations in the optimum working mode depending on the specific number of revolutions is known from the theory of pumps [6]:

$$\sigma_p = n^{4/3} / \mathrm{A}\ ;\ \ \mathrm{A} = 4700 \div 6300 \qquad (22),$$

pri čemer je koeficient A odvisen od konstrukcijske izvedbe delovnega kolesa in n_{sp}, A=4700 pri n_s=110; A=6300 pri n_s=180 [8]. Nekaj večjo vrednost

where the coefficient A depends on the structural design of the working wheel and n_{sp}, A=4700 at n_s=110; A=6300 at n_s=180 [8]. A somewhat higher koeficienta kavitacije dobimo, če uporabimo enačbo:

value of the cavitations coefficient is obtained if the following equation is used:

$$\sigma_p = n^{4/3} / 4000 \tag{23}$$

Koeficient rezerve $k\sigma$ definiramo analogno kakor pri določanju sesalne višine turbine. Predvsem moramo upoštevati možne napake pri povzemanju modelnih kavitacijskih karakteristik, prav tako pa tudi odstopanja, ki so posledica dejstva, da ne moremo zagotoviti popolne geometrijske in dinamične podobnosti modela in dejanskega stroja. V primeru kaplanovih turbin običajno jemljemo koeficient $k\sigma$= 1,1; pri francisovih turbinah za padce do 250 m je $k\sigma$ = 1,15 do 1,2, za padce prek 250 m pa se ta koeficient ne upošteva, to je $k\sigma$= 1,0.

Ker za reverzibilne hidravlične stroje do sedaj še ni na voljo dovolj podatkov za njihovo nomenklaturo, se vrednost tega koeficienta jemlje analogno nomenklaturi običajnih hidravličnih turbin velikih moči: $k\sigma$ = 1,1 do 1,15 [2].

Japonski strokovnjaki priporočajo enačbo (23) v obliki [3]:

The reserve coefficient kσ is defined analogously to the determination of the turbine suction height. In particular, it is necessary to take into consideration the mistakes in obtaining the model cavitations characteristic as well as the deviation resulting from the fact that it is not possible to ensure complete geometrical and dynamic equality of the model and the actual machine. For the case of Kaplans turbines the coefficient $k\sigma$= 1.1 is usually assumed; in the case of Francis turbines for 250 m falls the coefficient $k\sigma$= 1.15–1.2 , whereas for falls over 250 m that coefficient is not taken into account, i.e. $k\sigma$= 1.0.

For the time being, sufficient data are not available for parts lists of the reversible hydraulic machines, the value of that coefficient is taken in the same way as the parts lists of conventional hydraulic turbines of high powers: $k\sigma$= 1.1–1.15 [2].

Japanese experts recommend equation (23) in the following form [3]:

$$H_s \le 10 - \frac{\left(n\sqrt{Q_p}\right)^{4/3}}{1000} = 10 - \frac{n_s^{4/3} \cdot Hp}{5620} \tag{24}$$

kjer velja: $n_s = 3{,}65 \cdot \frac{n\sqrt{Q}}{H^{3/4}}$ in $\sigma_p = \frac{n_{sp}^{4/3}}{5620}$, kjer je Q_p srednji pretok v črpalnem režimu.

V območju črpalnih višin do 100 m se vrednosti dejanskih sesalnih višin večine RHE dobro ujemajo z vrednostmi, po enačbi (24). S povečanjem črpalne višine se vrednost H_S po enačbi (24) zmanjša. Zato je treba enačbo (24) popraviti takole [2]:

considering relation: $n_s = 3{,}65 \cdot \frac{n\sqrt{Q}}{H^{3/4}}$, that is $\sigma_p = \frac{n_{sp}^{4/3}}{5620}$, where Q_p is the mean flow rate in the pump working mode.

In the range of falls of up to 100 m the value of the actual suction height on most reversible hydroelectric power plants coincides well with the value obtained from equation (24). With an increase of the water head, however, the value H_s according to equation (24) decreases. Therefore, a correction must be entered in the water head and the following equation is obtained [2]:

$$H_s \le 10 - \frac{n_s^{4/3} \cdot H_p}{5620 - 3{,}94 \cdot H_{p\max}} \tag{25}$$

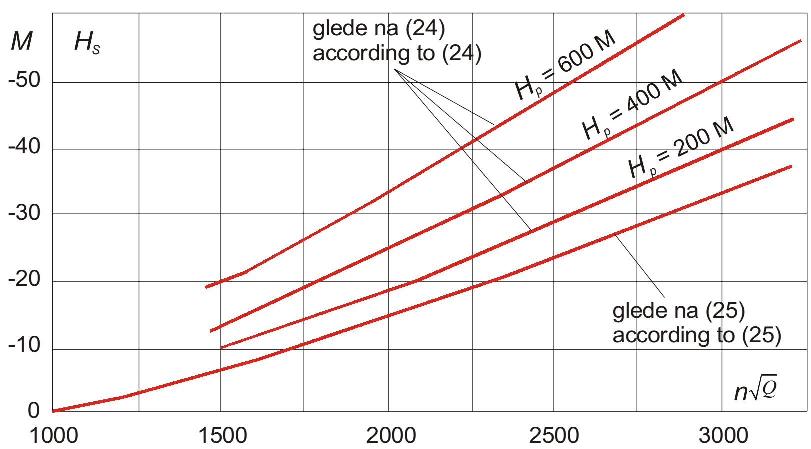


Fig. 7. *Primerjava teoretičnih in praktičnih vrednosti Hs [3]*
Fig. 7. *Comparison between Hs from theoretical and practical values [3]*

S primerjavo vrednosti H_S, dobljenih po enačbah (24) in (25), z dejanskimi vrednostmi na že izdelanih RHE pridemo do ugotovitve, da daje enačba (25) dobre rezultate za črpalne višine od 100 do 500 m. Iz te primerjave je tudi očitno, da se vrednost H_S z večanjem črpalne višine močno poveča.

By comparing the H_s values obtained according to equations (24) and (25) with the actual values on the reversible hydroelectric power plants already constructed we find that equation (25) gives good results for the heads from 100 to 500 m, and is also recommendable for that range of falls. The comparison shows that the value H_s strongly increases with an increase of the head and becomes infinite.

4 SKLEP

Reverzibilna hidroelektrarna (RHE) je vsekakor učinkovit vir vršne energije. Ta tip elektrarne zato že dalj časa vzbuja povečano zanimanje pri mnogih projektantskih organizacijah, raziskovalcih in konstrukterjih.

V prispevku je podana analiza nekaterih vprašanj, ki se pojavijo pri projektiranju tovrstnega tipa hidroelektrarne. Podane so namreč osnovne teorije delovnega procesa reverzibilnih črpalnih turbin. Razen tega je podana tudi analiza osnovnih rešitev nekaterih že izvedenih RPT francisovega tipa.

V nadaljevanju so obravnavana razmerja med parametri, ki se uporabljajo pri običajnih črpalkah in turbinah ter na podlagi le-teh opravljena analiza režimov, v katerih lahko obratuje tudi RPT.

Dognana je povezava med osnovnimi parametri RTP francisovega tipa in vrtilno frekvenco, opravljena pa je tudi analiza sesalne višine v odvisnosti od specifične vrtilne frekvence.

4 CONCLUSION

A reversible hydroelectric power plant (RHEPP) is surely the most real and the most efficient source of uppermost energy, and that is why many project departments, researchers and designers have, for a long time, shown great interest in this type of power plant.

In this context, an analysis of some questions concerning the planning of hydro-electric power plants of such type are given in this paper. Namely, basic theories of the working process of reversible pump turbines are given and conceptual solutions of some performed RTP of the Francis type are analyzed.

Then, relations between parameters are given, which are applied to classical pumps and turbines, and concerning that the regimes in which RPT can be found are analyzed.

A connection between basic parameters of RPT of the Francis type and a specific number of revolutions is established. An analysis of the intake altitude of sucking as a function of n_s is made.

5 OZNAKE
5 SYMBOLS

hidravlične izgube v črpalnem režimu	h_p	hydraulic losses in pump working mode
hidravlične izgube v turbinskem režimu	h_t	hydraulic losses in turbine working mode
hidravlične izgube v odvodnem sistemu pod pritiskom	h_{us}	hydraulic losses in pressurized outflow system
padec reverzibilne hidroelektrarne v črpalnem režimu	H_p	head of the reversible hydraulic power plant in pump working mode
sesalna višina	H_s	suction height
hidrostatični padec (višina)	H_{st}	head
padec reverzibilne hidroelektrarne v turbinskem režimu	H_t	head of reversible HEPP in turbine working mode
koeficient rezerve	$k\sigma$	coefficient of reserve
število vrtljajev	n	number of revolutions
specifična vrtilna frekvenca	$n_{s_{RPT}}$	specific rotating speed
največja moč	P	maximum power
srednji pretok v črpalnem režimu	Q_p	average flow in pump working mode
koeficient kavitacije v črpalnem režimu	σ_p	cavitation coefficient in pump working mode
kotna hitrost	ω	angular speed
absolutna kota vgradnje delovnega kolesa	$\dot{\nabla}$	elevation of installation of working wheel

6 LITERATURA
6 REFERENCES

[1] Krivčenko, G.I. (1975) Gidromehaničeskie perehodnye processy v gidroenergetičeskih ustanovkah, *Energija*, Moskva.
[2] Aršenevskij, N.N. (1977) Obratimiye gidromašiny gidroakumulirujuščih elektrostancij, *Energija*, Moskva.
[3] Trudy Instituta MISI - Moskva.
[4] Mrkić, M. (1985) Koncepcije reševanja RHE, Magistarski rad, Mašinski fakultet Beograd.
[5] Lomakin, A.A. (1966) Centrobežnye i osevye nasosy, *Mašinostroenije*, Moskva.
[6] Krivčenko, G.I. (1970) Nasosy i gidroturbiny, *Energija*, Moskva.
[7] Floriniskij, G.I. i dr. (1967) Nasosy i nasosnye stancii, Moskva.
[8] Gidroelektričeskie stancii: Pod redakcijej Gubina, *Energija*, Moskva, 1972.

Naslov avtorjev: prof.dr. Milo Mrkić
Univerzitet Crne Gore
Mašinski fakultet Podgorica
Cetinjski put bb
81000 Podgorica, ZRJ
milom@cg.ac.yu

Authors' Address: Prof. Dr. Milo Mrkić
University of Montenegro
Faculty of Mech. Eng. Podgorica
Cetinjski put bb
81000 Podgorica, ZRJ
milom@cg.ac.yu

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