

# DEVELOPMENT OF TURBINE-PUMP AGGREGATE

by

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*With increasing needs for energy, renewable energy sources have been nowadays the one of the main subject of interest all around the world. Therefore, we made an effort to develop a gamma of axial micro turbines, for a purpose of generation of electrical energy on small water derivation, as well as a part of an irrigation turbine-pump aggregate. In this paper are presented constructive solution of a turbine-pump aggregate with micro turbine and standard norm centrifugal pump. Based on project calculation, expected operating parameters are obtained.*

*After defining conceptual solution of the aggregate detailed development of model on computer and numerical simulation of fluid flow in axial turbine are conducted.*

Key words: *turbine-pump aggregate, working parameters, irrigation*

## Introduction

At the beginning of the research process different new solutions which enable the most optimal results for synchronous work of turbine and pump are considered. Two types of turbines are investigated – axial and Banky, and standard centrifugal norm-pump and during that research we look after next terms:

- location for aggregate installation,
- net head and flow rate during vegetation period,
- approximated calculation of turbine number of revolution,
- power obtained from turbine,
- standard norm-pumps which can be used from different manufacturers programs,
- construction and dimensions of over gear which can give desired number of revolution on pump shaft,
- simplicity of aggregate installation,
- simple and cheap maintenance,
- aggregate economy (price, life time, time for investment pay out),
- simple handling and aggregate installation, and
- little supplemental investment and simple outfitting of aggregate location.

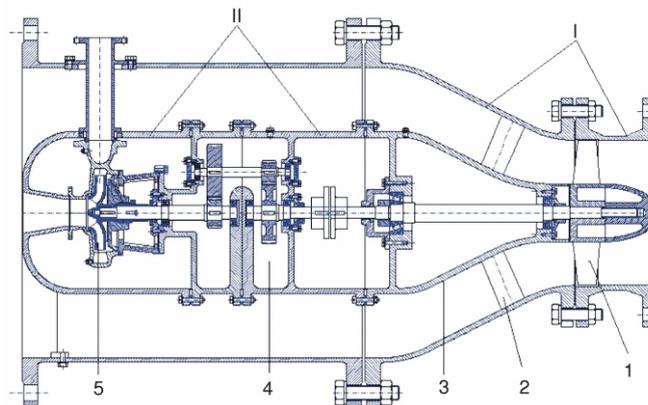
After analysis of stated aspects it is concluded that is most suitable to use axial turbine which give the power to the pump via over gear. Entire assemble will be installed

in casing which will be the part of conveying conduit to the turbine, and pump and over gear will be built in capsule which will be connected to the casing with ribs. It is predicted that aggregate could be easily transported to the location.

### Conceptual solution of aggregate

In this paper are presented conceptual solutions of turbine-pump aggregate and expected working parameters. Aggregate have the unregulated axial turbine with capsule in which over gear and centrifugal pump are placed. During the design process special effort is made to make the simplest possible construction which will also be the cheapest one. In order to make investment cost for water supply arrangement (moving gate, derivation pipeline) very low, aggregate is designed for turbine heads from 1 to 2 meters and for diameters of turbine impellers from 250 to 500 mm.

Two stage over gear have cylindrical gears. On lead-out shaft of over gear pump impeller is placed. The over gear also have the function of pump bearing case with console and spiral pump casing attached to it (fig. 1).



**Figure 1. Conceptual solution of aggregate**

*I – turbine part, II – capsule, 1 – turbine impeller, 2 – guide vane blades, 3 – turbine bearing case, 4 – over gear, 5 – pump*

Number of revolution of turbine impeller is not necessary to keep constant, and guide vane blades and impeller blades are fixed. Inexistence of regulating equipment substantially simplify turbine construction and lowering the turbine price.

From hole amount of water which passing trough the aggregate pump use 4% of flow rate, so it can be concluded that this flow doesn't influence on flow trough the turbine.

On fig. 2 the 3-D model of aggregate is shown.

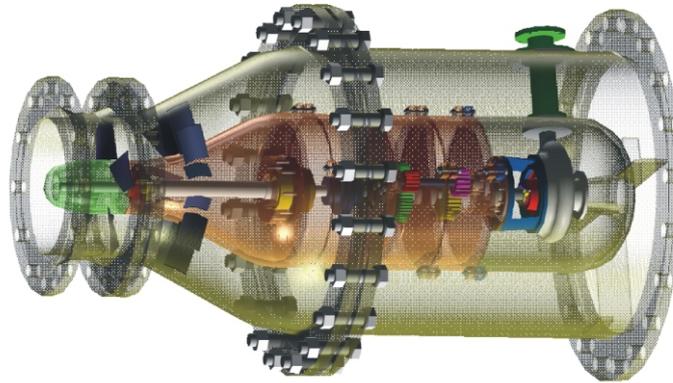


Figure 2. 3D model of aggregate

### Basic working parameters of aggregate

Notations:

$H_T$  – net turbine head [m],  $H_P$  – pump head [m],  $Q_T$  – flow trough turbine [ $m^3/s$ ],  $Q_P$  – pump flow [ $m^3/s$ ],  $\eta_T$  – turbine efficiency [-],  $\eta_P$  – pump efficiency [-],  $\eta_m$  – over gear efficiency [-],  $\eta_{TP}$  – aggregate efficiency [-],  $N_T$  – turbine power [W],  $N_P$  – pump power [W],  $N_{tT}$  – theoretical turbine power [W],  $N_{efP}$  – effective pump power [W].

Equations:

$$N_{tT} = \rho g Q_T H_T, \quad N_{efP} = \rho g Q_P H_P$$

$$N_T = N_{tT} \eta_T = \rho g Q_T H_T \eta_T, \quad N_P = \frac{N_{efP}}{\eta_P} = \frac{\rho g Q_P H_P}{\eta_P} \quad (1)$$

$$\eta_{TP} = \frac{N_{efP}}{N_{tT}} = \frac{Q_P H_P}{Q_T H_T} = \eta_T \eta_P \eta_m \quad (2)$$

$$Q_P = \frac{\eta_{TP}}{(H_P/H_T)} Q_T \quad (3)$$

For expected efficiency rates:

$$\eta_T = 0.75 - 0.75 \quad (0.72)$$

$$\eta_P = 0.60 - 0.65 \quad (0.62)$$

$$\eta_m = 0.95 - 0.97 \quad (0.96)$$

$$\eta_{TP} = 0.40 - 0.47 \quad (0.43)$$

eq. (3) obtain the following form:

$$Q_P = \frac{0,43}{(H_P/H_T)} Q_T, \text{ or } \frac{Q_P}{Q_T} [\%] = \frac{43}{(H_P/H_T)} \quad (3')$$

Turbine-pump aggregates are developed for net turbine heads:

$$H_T = 1, 1.5, \text{ and } 2 \text{ m} \quad \text{and} \quad H_P/H_T = 10, 20, \text{ and } 30.$$

Using eqs. (3') following results are obtained:

**Table 1**

$H_P/H_T$	10	20	30
$Q_P/Q_T$ [%]	4.3	2.15	1.075

And according to these results it can be concluded that flow rate which use pump doesn't influence on flow trough the turbine.

Heads for pump choosing ( $H_P$ ) depend of net turbine head ( $H_T$ ) and ratio  $H_P/H_T$  and they have the following values:

**Table 2**

$H_T$ [m]	$H_P$ [m]		
	$H_P/H_T = 10$	$H_P/H_T = 20$	$H_P/H_T = 30$
1	10	20	30
1.5	15	30	45
2	20	40	60

### Expected nominal (calculated) specific speed of turbine

Specific speed of turbine is defined by equation:

$$n_{sT} = \frac{n_T \sqrt{N_T}}{\sqrt[4]{H_T^5}} \quad (4)$$

where:  $N_T$  – turbine power [kW],  $n_T$  – turbine [rpm], and  $H_T$  – net turbine head [m].

On the base of equation  $N_T = \rho g Q_T H_T \eta_T$ , ( $g = 9,81 \text{ m/s}^2$ ), specific turbine speed can be defined by formula:

$$n_{sT} = 3.13 \frac{n_T \sqrt{Q_T}}{\sqrt[4]{H_T^5}} \sqrt{\eta_T} \quad (4')$$

where:  $Q_T$  is the turbine flow rate [ $\text{m}^3/\text{s}$ ].

Unit turbine number of revolution ( $n_1$ ) and unit turbine flow rate ( $Q_1$ ) are determined with following terms:

$$n_1 = \frac{n_T D_T}{\sqrt{H_T}}, \quad Q_1 = \frac{Q_T}{D_T^2 \sqrt{H_T}} \quad (5)$$

where  $D_T$  is the turbine impeller diameter [m].

Working parameters  $n_{sT}$ , and  $Q_1$  are linked with equation:

$$n_{sT} = 3.13 n_1 \sqrt{Q_1 \eta_T} \quad (6)$$

Further analysis of axial turbines working characteristics and using of systematized data for  $n_{sT}(H_T)$  and  $Q_T(H_T)$  from scientific literature, for turbine heads  $H_T = 1\text{-}2 \text{ m}$  with respect that micro turbine is in question, expected nominal working regime is

$$n_{sT} \sim 700 \quad \text{and} \quad Q_1 \sim 2.1 \text{ m}^3/\text{s}$$

According to eq. (6) for estimated turbine efficiency rate  $\eta_T = 0.72(0.70 - 0.75)$ , expected unit number of revolution is  $n_1 \sim 180$ .

### Expected nominal (calculated) turbine working parameters

Turbine working parameters ( $Q_T$ ,  $n_T$ ,  $N_T$ ), for  $Q_1 = 2.1 \text{ m}^3/\text{s}$  and  $n_1 = 180 \text{ rpm}$ , are calculated using next equations:

$$Q_T [\text{m}^3/\text{s}] = Q_1 D_T^2 \sqrt{H_T} = 2.1 D_T^2 \sqrt{H_T}$$

$$n_T [\text{rpm}] = \frac{n_1 \sqrt{H_T}}{D_T} = \frac{180 \sqrt{H_T}}{D_T}$$

$$N_T [\text{kW}] = 9.81 \rho \eta_T Q_T H_T = 6.87 \rho Q_T H_T$$

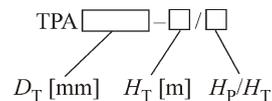
Results of calculations for turbine impellers diameters  $D_T = 0.25$  and  $0.32$ , and net turbine heads  $H_T = 1, 1.5,$  and  $2$  m are given in tab. 3.

**Table 3. Expected nominal working parameters of turbine**

$D_T$ [mm]	$H_T$ [m]	$Q_T$ [m <sup>3</sup> /s]	$n_T$ [rpm]	$N_T$ [kW]
250	1	0.125	720	0.86
	1.5	0.161	882	1.66
320	1	0.215	562	1.48
	1.5	0.263	689	2.71
	2	0.304	795	4.18

### Parameters for pump selection

Label of turbine-pump aggregate for irrigation is:



For example: TPA 500-1,5/20 – turbine-pump aggregate with impeller diameter 500 mm, designed for net turbine head 1,5 m and pump head which is 20 times larger from turbine head ( $H_p = 20 \times 1,5 = 30$  m).

For turbine impeller diameters  $D_T = 250, 300, 400,$  and  $500$  mm, according to turbine heads given in second row of tab. 3 and accepted ratios  $H_p/H_T = 10, 20,$  and  $30$ , it is possible to combine 27 aggregates (6 for  $D_T = 200, 400$  and  $500$  mm and 9 for  $D_T = 320$  mm). For every aggregate pump must be selected and determined transmission ratio of over gear (according which gears diameters are determined).

Required pump heads are given in tab. 2, and according to turbine flow rates given in tab. 3, using eq. (3') required pump flow rates are determined. During pump selection special attention is given to spiral case dimension which must be placed in capsule which diameter is 10% bigger then turbine impeller diameter.

Toward pump working parameters, determined within prior described method, application of pumps from product range of Pump Factory “Jastrebac” from Niš, Serbia, is considered. Centrifugal NORM pumps SCP, in the case of working regimes with 2900 rpm, satisfied all aggregates, except aggregates with labels TPA250-1/30, TPA250-1,5/30, and TPA320-2/30 (for which NORM pumps SCP are to big). For this aggregates pumps from manufacturer Pedrollo, Italy, can be used, and probably from some other manufacturers.

The biggest number of revolution of pump is  $n_p = 2900$  rpm, a least expected number of revolution of turbine is  $n_T = 441$  rpm (tab. 3), and according to this values minimal transmission ratio of over gear is  $i_{\min} = n_{T\min}/n_p = 0,152$  (which can be achieved with two stage over gear with cylindrical gears, placed in aggregate capsule).

## Numerical simulations

With the development of CFD software's turbomachinery designing has been considerably changed. Nowadays, it can be made a numerical experiment in order to test if operating characteristics of turbomachinery satisfy our requirements. Also we can make as many changes in turbomachinery design (such as blade profile shapes, or some other constructive parameters) as we need, and test turbomachinery every time to examine influence of these modifications, using numerical simulations.

For discretisation and numerical solving of Navier-Stokes equations it is used a finite volume method. There are four phases of CFD modeling:

- defining of geometry and mesh,
- defining of initial and boundary conditions, fluid properties and other physical parameters,
- numerical solving of governing equations, and
- post-processing of numerical simulation results.

## Guide vane and turbine blade defining

Preliminary calculation of the axial micro turbine is done with an assumption of axisymmetrical fluid flow in the turbine. Axisymmetrical flow surfaces in meridional cross-section are determined using a model of potential flow of nonviscous fluid. Guide vanes and turbine blades are designed with aerodynamic profiles in six different coaxial cross-sections (1-1, 1'-1', 2-2, 3-3, 4-4, 5-5), fig. 3.

Profile cascades of the turbine rotor are defined using a method of lift force and Kaplan profiles. Lift coefficients of profiles in a profile cascade are derived to lift coefficients of single profiles (Veining diagram and Numachi diagram).

Circumferential velocities on the guide vane outlet (that we use for determination of the profile angles on the guide vane outlet) are obtained by the formula:

$$(r_{ik}c_{u_{ik}})_{i-i} = K_{i-i} (r_o c_{u_o})_{i-i}, \quad i = 1, 1', 2, 3, 4, 5 \quad (K_{i-i} = 1)$$

where:  $r_{ik}c_{u_{ik}}$  – circulation on the turbine rotor inlet and  $r_o c_{u_o}$  – circulation on the guide vane outlet.

It is assumed that coefficient of circulation loss between the guide vane and the turbine rotor are:  $K_{1-1} = K_{1'-1'} = 0.95$  and  $K_{2-2} = K_{3-3} = K_{4-4} = K_{5-5} = 0.98$

Guide vane is consists of 12 vanes and it is assumed that angle of stream deflection is  $4-6^\circ$  (from the hub to the shroud of the guide vane, respectively).

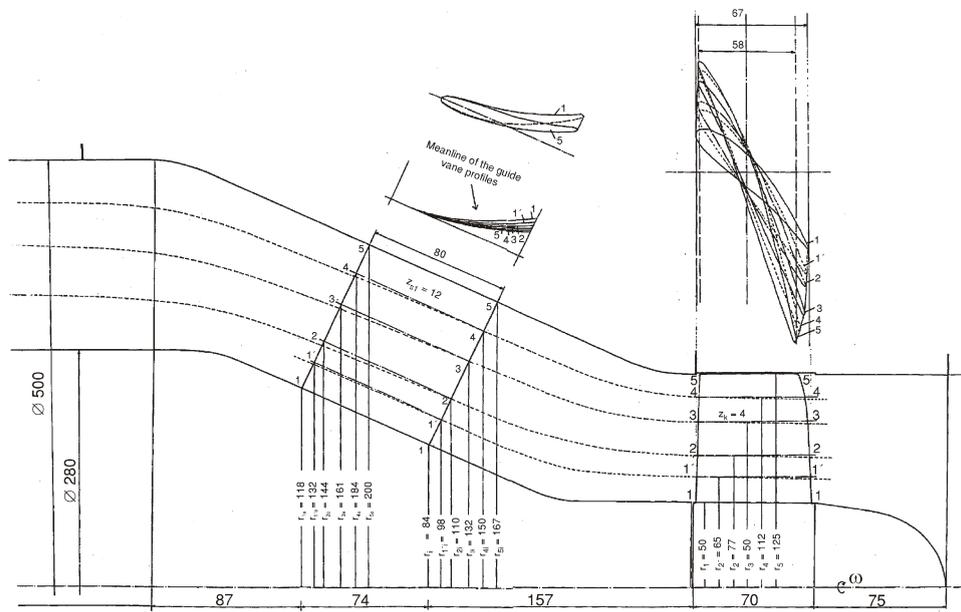


Figure 3. Meridional cross-section of the axial micro turbine and calculated meridional streamlines

### Defining of geometry and mesh

There are large number of various methods for discretisation and solving Navier-Stokes partial differential equations, which are nowadays used in CFD codes. The most often is used the finite volume method which divides a domain in subdomains (control volumes). Governing equations are discretised and solved for each control volume, and as a result, an approximate value of any quantity can be obtained in every point of the observed domain. Thus it can be obtain a full picture of the considered fluid flow.

First thing to be done was defining of geometry and creating a 3-D model of the guide vane and axial micro turbine. Diameter of turbine rotor is  $D_T = 250$  mm, flow rate through the turbine is  $Q_T = 0.166$  m<sup>3</sup>/s, calculated turbine head is  $H_T = 1.5$  m and it rotates with  $n_T = 880$  rpm.

Further on it was defining a mesh, thicker around a blade surface and wrapped geometry area, with more than 1.500,000 tetrahedral elements.

The next step was defining of initial and boundary conditions, fluid properties and other physical parameters. For simulations real viscous fluid (water) is used. A

three-dimensional simulation of flow in turbine is carried out with  $k-\varepsilon$  turbulent model for which appropriate solver control parameters are tuned.

For solving of partial differential equations it was used “high resolution” procedure, and numerical simulations convergence criteria was that root mean square values of the equation residuals were  $10^{-5}$ .

### Correction of the guide vanes

Analyzing results of numerical simulation for the calculated axial micro turbine it is obtained a low hydraulic efficiency ( $\eta_h = 0.72$ ), and reasons for that are:

- (a) Great loss of velocity circulation  $rc_u$ , between the guide vanes and the turbine rotor, on the flow surfaces near the turbine shroud.
- (b) The flow has a larger angles in the outlet of the guide vanes (compared to the preliminary calculated angles), on the flow surfaces near the turbine case.
- (c) Relatively large circumferential velocities on the outlet of the turbine rotor (from the value  $c_u = 0.58$  m/s, near the hub, to  $c_u = 0.99$  m/s, near the shroud), which means that turbine blades dos not achieve required inclination.

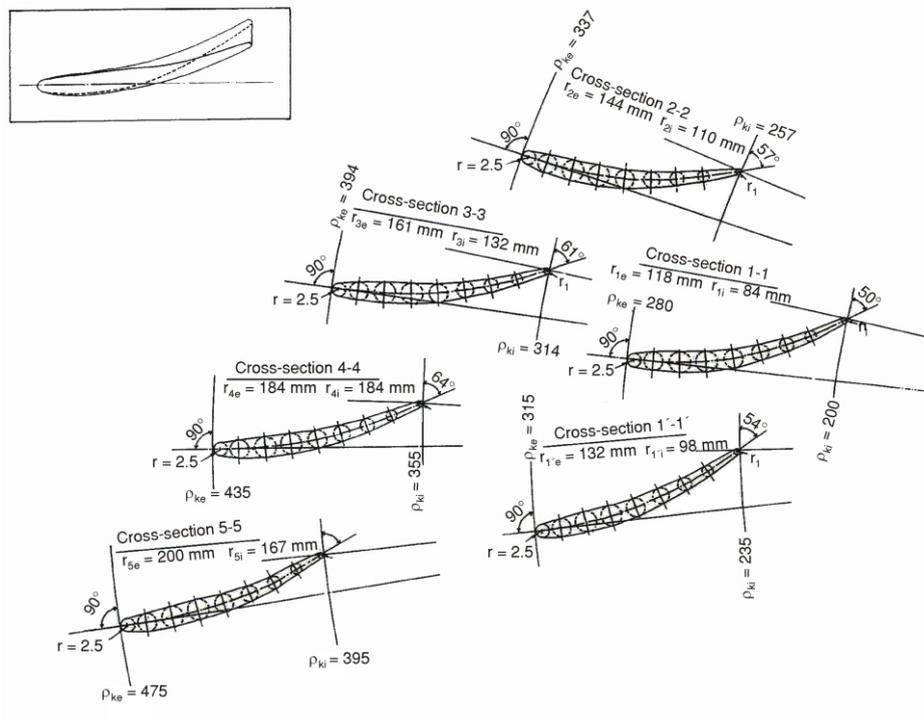


Figure 4. Profiles of first guide vane (turbine  $D_T = 250$  mm,  $H_T = 1.5$  m)

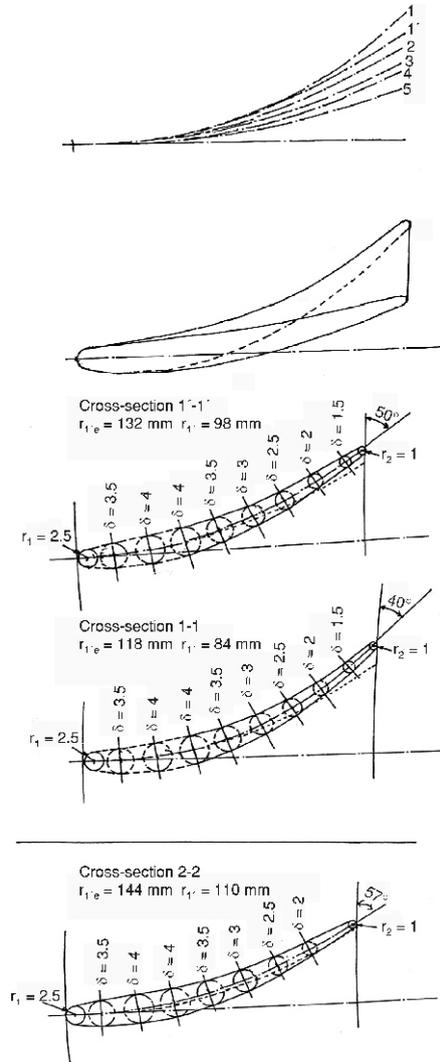


Figure 5. Profiles of new guide

According to numerical simulation results it is calculated coefficients of circulation loss between the guide vane and the turbine rotor:  $K_1 = 0.80$ ,  $K_{1'} = 0.95$ , and  $K_2 = K_3 = K_4 = K_5$  (0.98-0.99). It means that previously calculated angles of stream deflection in cross-sections 1-1 and 1'-1' are  $4^\circ$  larger than angles obtained with corrected blade angles.

Guide vane profiles are corrected in cross-section 1-1 and 1'-1' by outlet angle reduction of  $10^\circ$ . Also, the chamber line of the cross-section 2-2 has been changed. Vane profiles of the other cross-sections were not changed (figs. 4 and 5).

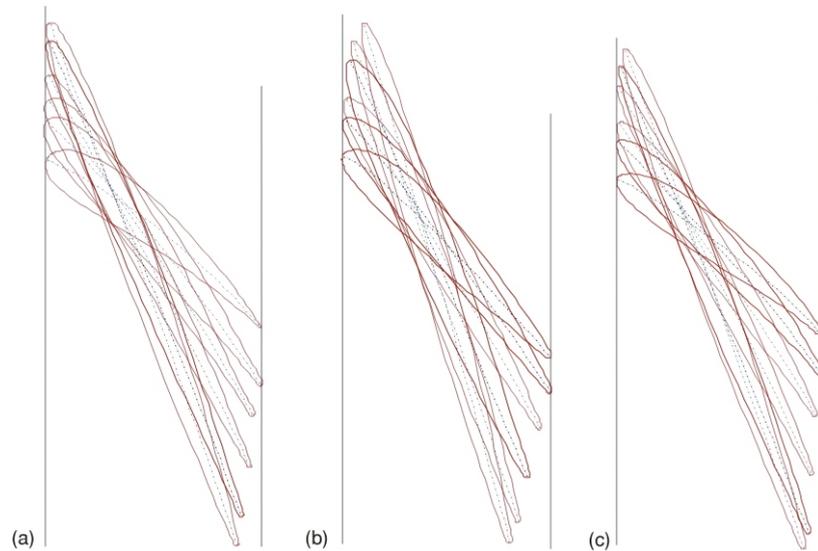
### Correction of the turbine rotor blades

Analyzing the results of the numerical simulation of the flow through the axial micro turbine, with correction of the guide vanes, it is obtained a hydraulic efficiency of the turbine  $\eta_h = 0.80$ . Fluid flow through the turbine is better after the correction of the guide vanes, but still there was a space for turbine blade correction. Since, there was not a possibility to change blade profiles, solution is found in changing the angle of attack, *i. e.* changing angle of blade profiles.

Blade angle of a blade profile in one calculated flow surface affects significantly on blade angles on other flow surfaces. Therefore, seeking for the optimal angles of blade profiles on calculated flow surfaces is achieved by variation of different angles of blade profiles. After many variation, and numerical simulations for each variation, it is

finally accepted a solution (fig. 6).

Chosen turbine blades are completely different than turbine blades obtained by preliminary calculation. According to the numerical simulation, working parameters of the turbine would be: hydraulic efficiency  $\eta_h = 0.84$ , turbine head  $H_T = 1.5$  m, flow rate  $Q_T = 0.166$  m<sup>3</sup>/s, rotation speed  $n_T = 860$  rpm, torque of the turbine rotor  $M_{kT} = 22.8$ , and turbine rotor power  $N_{kT} = 2050$  W.

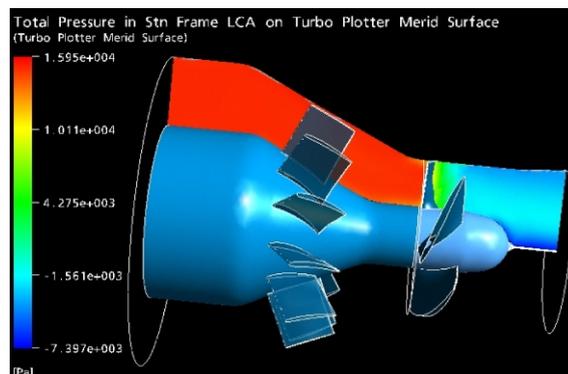


**Figure 6. Turbine profiles**  
*(a) preliminary calculated, (b) one of variation, (c) final correction*

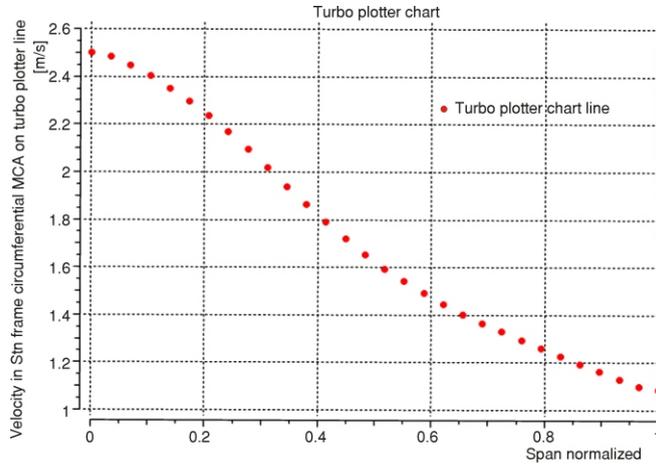
## Results of numerical simulations

Our approach in axial turbine design was in iterative procedure that is consisting of blade correction, after preliminary calculation was done and numerical simulation afterward.

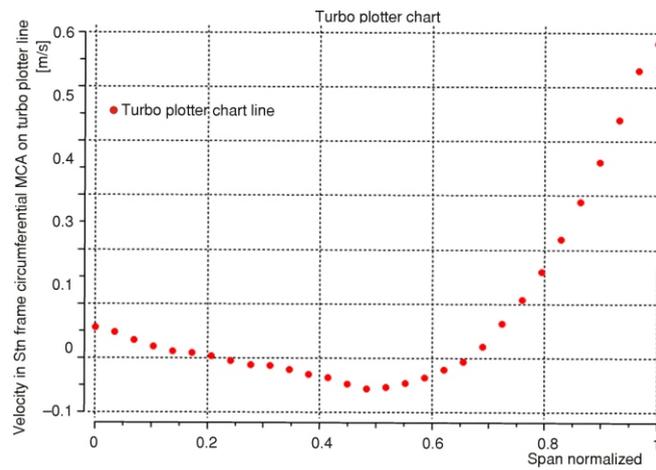
With such a numerical experiments we could obtain a wide range of parameters, such as total pressure distribution (fig. 7), velocity component distribution (figs. 8 and 9),



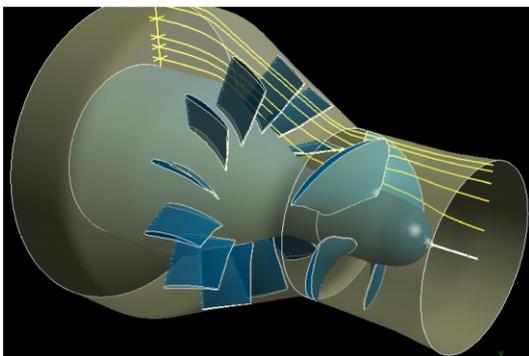
**Figure 7. Pressure distribution in meridional surface**



**Figure 8. Velocity circumferential at the turbine rotor inlet**



**Figure 9. Velocity circumferential at the turbine rotor outlet**



**Figure 10. Meridional streamlines**

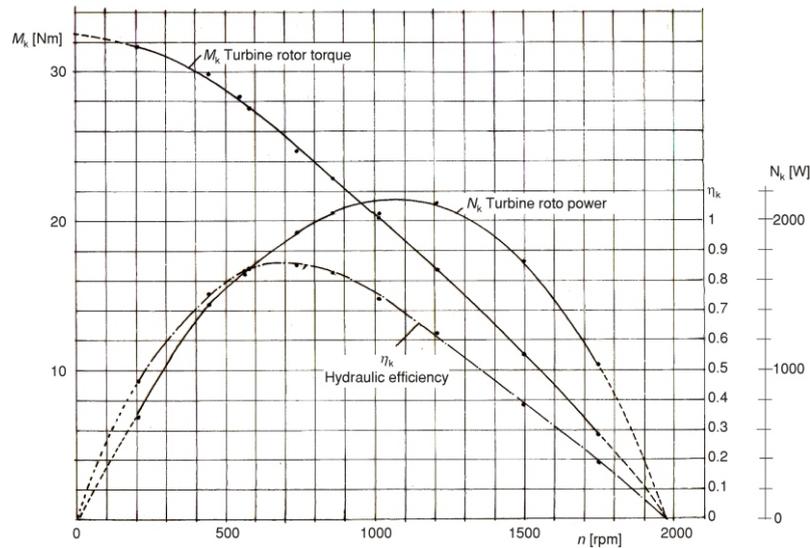


Figure 11. Turbine characteristics  $D_T = 250$  mm,  $H_T = 1,5$  m,  $n = 880$  min.

*etc.*, in the whole domain of the fluid flow. That enables analyzing results in the flow domain and making necessary corrections each time until obtaining optimal results.

### Conclusions

Turbine-pump aggregates can be used for irrigation on the rivers where flow rates in vegetation period (April-September) are not smaller than flow rates given in tab. 3, where requested turbine heads can be achieved with relative simple moving gates and derivation conduit. To reduce the investment costs it is recommended to use the aggregates on the rivers where flow rates in vegetation period are not few times bigger then turbine flow rate.

The price of turbine-pump aggregate should be equal to price of pump with electromotor or price of pump aggregate with diesel motor. Energy saving for irrigation with this aggregate during few years will be equal to investment cost for building of moving gate and derivation conduit.

Besides the energy saving such turbine-pump aggregate also provides safe environment.

According to the numerical simulation results, the shape of the turbine blades doesn't influence considerably on the fluid flow between the guide vanes and turbine rotor, but the shape of the guide vane does.

Iterative procedure that is consist of blade correction and numerical simulation gives good results in optimization of the axial micro turbine and, what isn't less important, there is no need for making physical prototype of the turbine. Such numerical experiment enable us to correct all miscalculations before making a prototype, which saves us money as well as time.

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