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DO NOT BE AFRAID OF SMALL HIGH-SPEED FRANCIS TURBINES

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Abstract: In the first quarter of the last century hydraulic power plants were equipped with high-speed Francis turbines even in the situation when a contemporary project manager would suggest Kaplan turbine. The reason is simple. Mr. Kaplan patented his turbine only in 1912 [1], https://en.wikipedia.org/wiki/Viktor_Kaplan . Those high-speed Francis turbines have just reached their lifetime. Mainly runners need repair. Our customers' respond is that even renowned firms refuse to deliver runners with better parameters. Offer is to replace whole turbine with Kaplan or to make a copy of the existing runner. This paper presents experience and results of such a high-speed runner design. The runner substituted the one of Prokopa & sons from 1939 in powerhouse and mill at Křemže stream. Virtual prototyping technique has been used.

KEYWORDS: Francis turbine, runner, refurbishment, CFD, virtual prototype.

1 Introduction

This article deals with concrete design of a Francis runner as a replacement of an old one installed already in 1939 in powerhouse and mill at Křemže stream. Since the small power plant is not a national heritage the owner could start the refurbishment. This consisted of new generator, new guide vanes but as a copy of the old ones, repair or manufacture of some parts and fitting the turbine up with a new runner. The runner is just the subject of the paper. The event had a personal meaning for the owner since the mill has been a family private property for a long time. That is why I provide the article with some story.

The owner expected to get a runner with higher efficiency and made from stainless steel. It is easy to meet the second condition. First one was a problem. He turned to a respected Czech turbine producer and got rather surprising answer. There does not exist a better runner than he has. He was offered with an exchange for a Kaplan bulb turbine. Such an investment seemed to be too expensive. Being aware of modern CFD methods he refused to accept the verdict that there is no chance to get better design than that from 1939. He contacted Strojírny Brno and we accepted the challenge. The goal was to get maximum electric output of 13 kW. This means increase in power and efficiency by 13%. The paper is not aimed at complete methodology of Francis runner design. Nevertheless it shows main directions where to go when somebody meets the problem of refurbishment of a small high-speed Francis turbine. Also a method to estimate computational error of total power output from such limited data sources is presented.

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2 Nomenclature

Table 1

Constant or variable	Definition	Unit	Value	Description
D		m	0.35	Runner diameter.
n		/min	470	Runner speed.
ρ		kg/m^3	999	Water specific weight at 20 °C.
g		m/s^2	9.81	Gravitational acceleration.
T		Nm		Torque at turbine shaft axis.
p_a		Pa	101000	Atmospheric pressure.
p_1	$p_2 + \rho\, g\, H_G$	Pa	25455	Inlet total pressure specifying gross head. See fig. 7.
p_2	$-\rho\, g\, H_s$	Pa	-21539	Outlet total pressure specifying gross head. See fig. 7.
H_G	$\frac{p_1 - p_2}{g\,H}$	m	4.6	Gross head.
p_{in}		Pa	Computed.	Total pressure at the control volume inlet. See fig. 7.
p_{out}		Pa	Computed.	Total pressure at the control volume outlet. See fig. 7.
H	$\frac{p_{in} - p_{out}}{\rho\, g}$	m		Net head.
H_s		m	2.2	Suction head.
n_{11}	$\frac{n.D}{\sqrt{H}}$	/min	75.1	Unit runner speed [3].
n_{ED}	$\frac{n\ D}{60\sqrt{gH}}$		0.4	Speed factor [5].
ω	$\frac{2\pi n}{60}$	rad/s	49.22	Angular velocity of the runner.
Q		m^3/s		Flow rate.
Q_{11}	$\frac{Q}{D^2\sqrt{H}}$	m^3/s		Unit flow rate [3].
Q_{ED}	$\frac{Q}{D^2\sqrt{gH}}$			Discharge factor [5].
η_H	$\frac{T\omega}{\rho g H Q}$			Hydraulic (turbine) efficiency.
η_G				Generator efficiency.
η_B			0.91	Efficiency of belt transmission [4].
η	$\eta_H\, \eta_G\, \eta_B$			Turbogenerator efficiency.
P	$\rho\, g\, H\, Q\, \eta_H\, \eta_G\, \eta_B$			Total power output of turbogenerator.

3 State before reconstruction

Parameters of the turbine in powerhouse and mill at Křemže stream. Fig. 1 gives the evidence of data.

Table 2

Runner diameter.	0.35 m
Number of runner blades.	10
Number of guide vanes.	10
Gross head. Now by 0.2 m lower then projected.	4.6 m
Suction head	2.2 m
Maximum discharge.	0,37 m^3 /s
Maximum hydraulic power.	18.4 HP = 13.72 kW
Maximum electric power.	11.5 kW
Q_{11} at maximum discharge.	1.38 m^3/s
Q_{ED} at maximum discharge.	0.44

The turbine was designed at operation point n_{ED} = 0.4 (n_{11} = 75.1) and Q_{ED} = 0.44 (Q_{11} = 1.38) . Since the designer had to use Francis turbine he chose the speed of 570 /min to get suitable value of n_{ED} . If we take into consideration the small gross head it is quite clear that Kaplan bulb turbine with runner speed about 800 /min is the best solution.

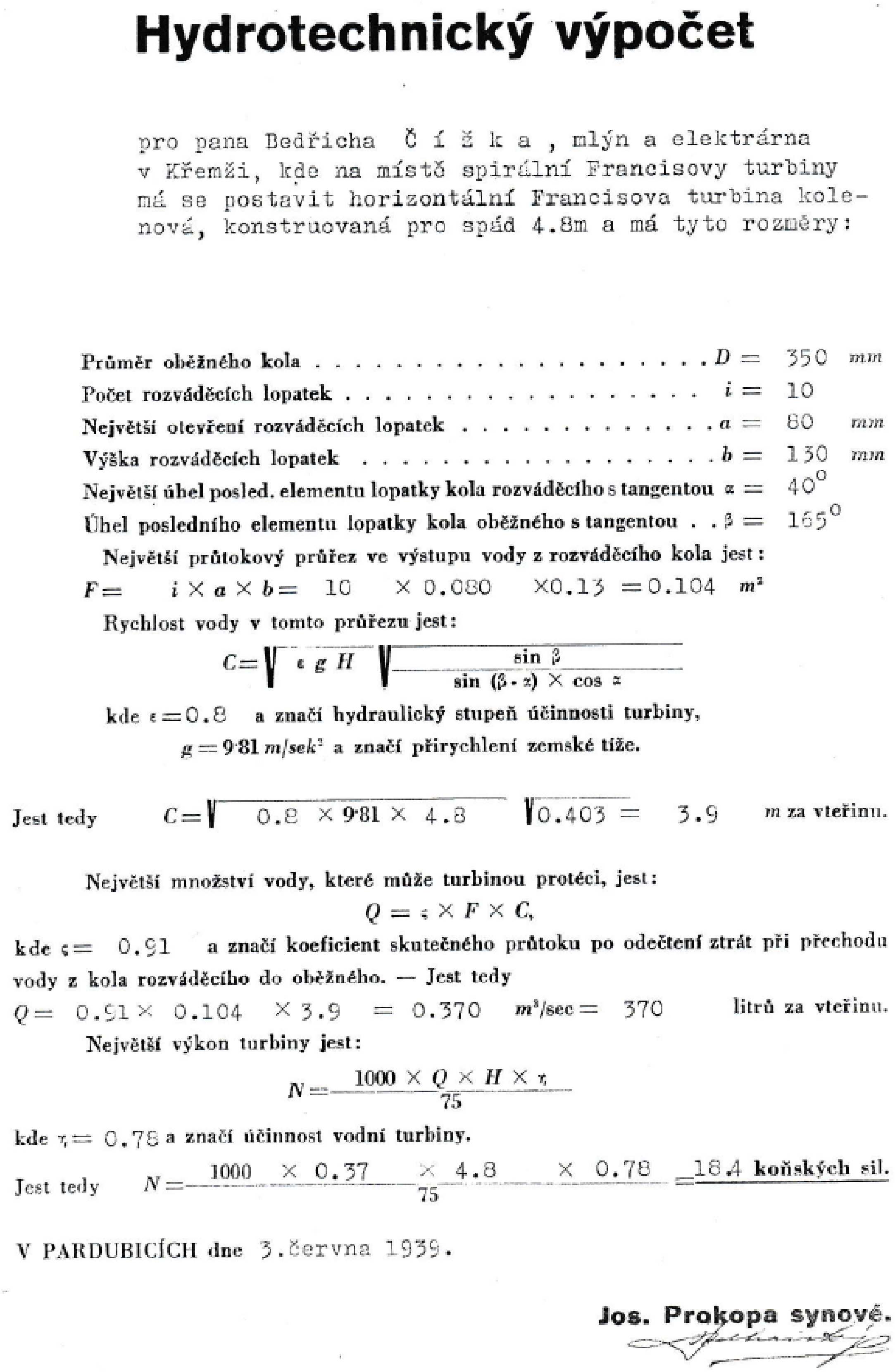
It is necessary to give translation to English of some nomenclature used in Fig. 1.

Table 3

Expression	**English meaning**
Hydrotechnický výpočet.	Calculation of hydraulic design.
D	Runner diameter.
i	Number of guide vanes.
a	Maximum opening of guide vanes. Known as maximum a_0 .
b	Height of guide vane.
Q	Maximum discharge in litres.
N	Maximum hydraulic power in HP.

4 Reverse engineering

To be able to do an analysis of a difference between the turbine with old and new runner it was of great importance to get 3D model of the existing runner. Other reason was to get fitting dimensions for the new one. Inlet height and diameters, overall height and outlet diameter. Old runner was scanned and 3D model was prepared.



Hydrotechnický výpočet

pro pana Bedřicha Č í ž k a , mlýn a elektrárna v Křemži, kde na místě spirální Francisovy turbiny má se postavit horizontální Francisova turbina kolenová, konstruovaná pro spád 4.8m a má tyto rozměry:

Průměr oběžného kola	$D =$	350	*mm*
Počet rozváděcích lopatek	$i =$	10	
Největší otevření rozváděcích lopatek	$a =$	80	*mm*
Výška rozváděcích lopatek	$b =$	130	*mm*
Největší úhel posled. elementu lopatky kola rozváděcího s tangentou	$\alpha =$	40°	
Úhel posledního elementu lopatky kola oběžného s tangentou . .	$\beta =$	165°	

Největší průtokový průřez ve výstupu vody z rozváděcího kola jest:

$F = i \times a \times b = 10 \times 0.080 \times 0.13 = 0.104$ m^2

Rychlost vody v tomto průřezu jest:

$$C = \sqrt{\varepsilon\, g\, H} \sqrt{\frac{\sin \beta}{\sin (\beta - \alpha) \times \cos \alpha}}$$

kde $\varepsilon = 0.8$ a značí hydraulický stupeň účinnosti turbiny,

$g = 9{\cdot}81\ m/sek^2$ a značí přirychlení zemské tíže.

Jest tedy $C = \sqrt{0.8 \times 9{\cdot}81 \times 4.8}\ \sqrt{0.403} = 3.9$ *m* za vteřinu.

Největší množství vody, které může turbinou protéci, jest:

$$Q = \varsigma \times F \times C,$$

kde $\varsigma = 0.91$ a značí koeficient skutečného průtoku po odečtení ztrát při přechodu vody z kola rozváděcího do oběžného. — Jest tedy

$Q = 0.91 \times 0.104 \times 3.9 = 0.370$ $m^3/sec = 370$ litrů za vteřinu.

Největší výkon turbiny jest:

$$N = \frac{1000 \times Q \times H \times \eta}{75}$$

kde $\eta = 0.78$ a značí účinnost vodní turbiny.

Jest tedy $N = \dfrac{1000 \times 0.37 \times 4.8 \times 0.78}{75} = \underline{18.4}$ **koňských sil.**

V PARDUBICÍCH dne 3.června 1939.

Jos. Prokopa synové.

Fig. 1 Original document with hydraulic calculation [2].

Fig. 2 Original runner.

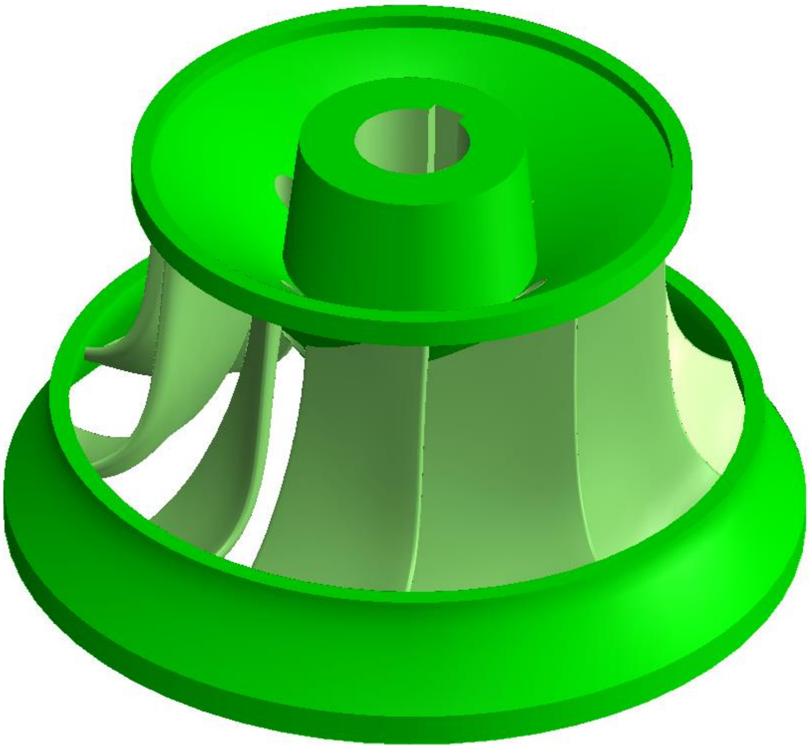


Fig. 3 Scan of the original runner.

Designer equipped with runner model can easily cut out meridian curve and one blade. These serve as input to Numeca Autogrid. See fig. 4.

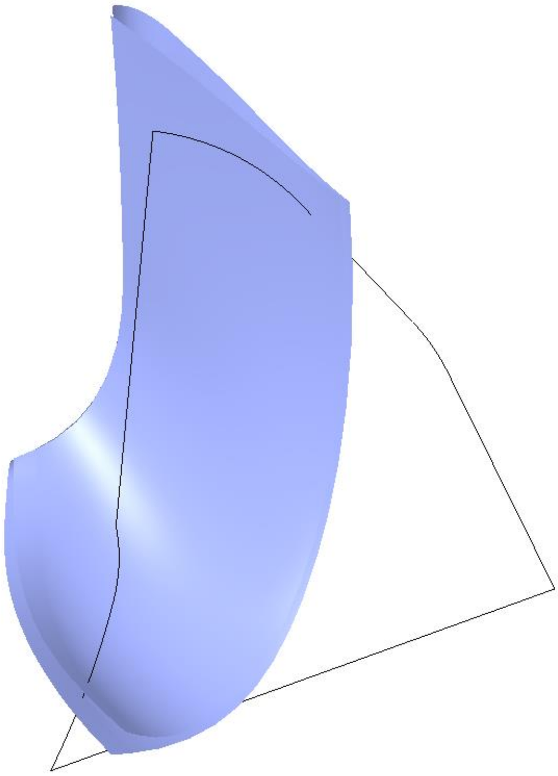


Fig. 4 Meridian section curve and blade surface.

Resulting cyclical symmetric section of the runner can be seen in fig. 5. There one periodic and runner inlet surfaces are removed.

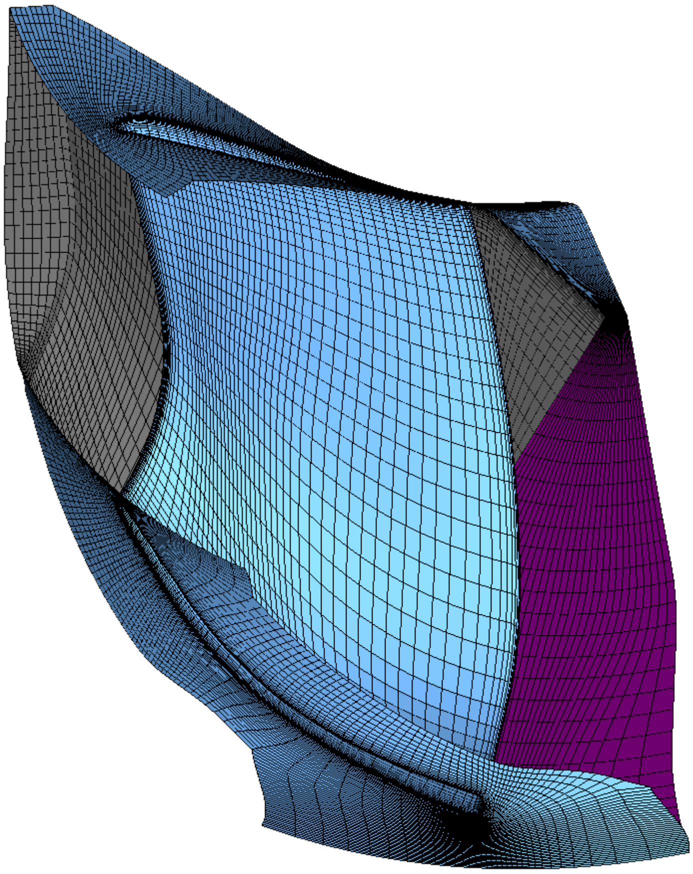


Fig. 5

5 Runner design

During hydraulic design following software has been used.

Table 4

Software	Producer	Purpose
Ansys CFD	Ansys	CFD computation, pre and post processing.
Autogrid	Numeca	Blade meshing.
Solid Works	Dassault Systems	3D modeling.
Creo	PTC	3D modeling.
Runner blade	In-house	Runner blade shape computation.

5.1 Virtual prototype

Virtual prototyping is so strong tool that small water turbines do not undergo any experimental verification. Anyway it is necessary to have drawings to be able to prepare the prototype. It was a piece of luck that there was enough documentation to enable to get 3D model. Part of it can be seen in following figure [2].

Virtual prototype parameters correspond with real site conditions. See also table 1.

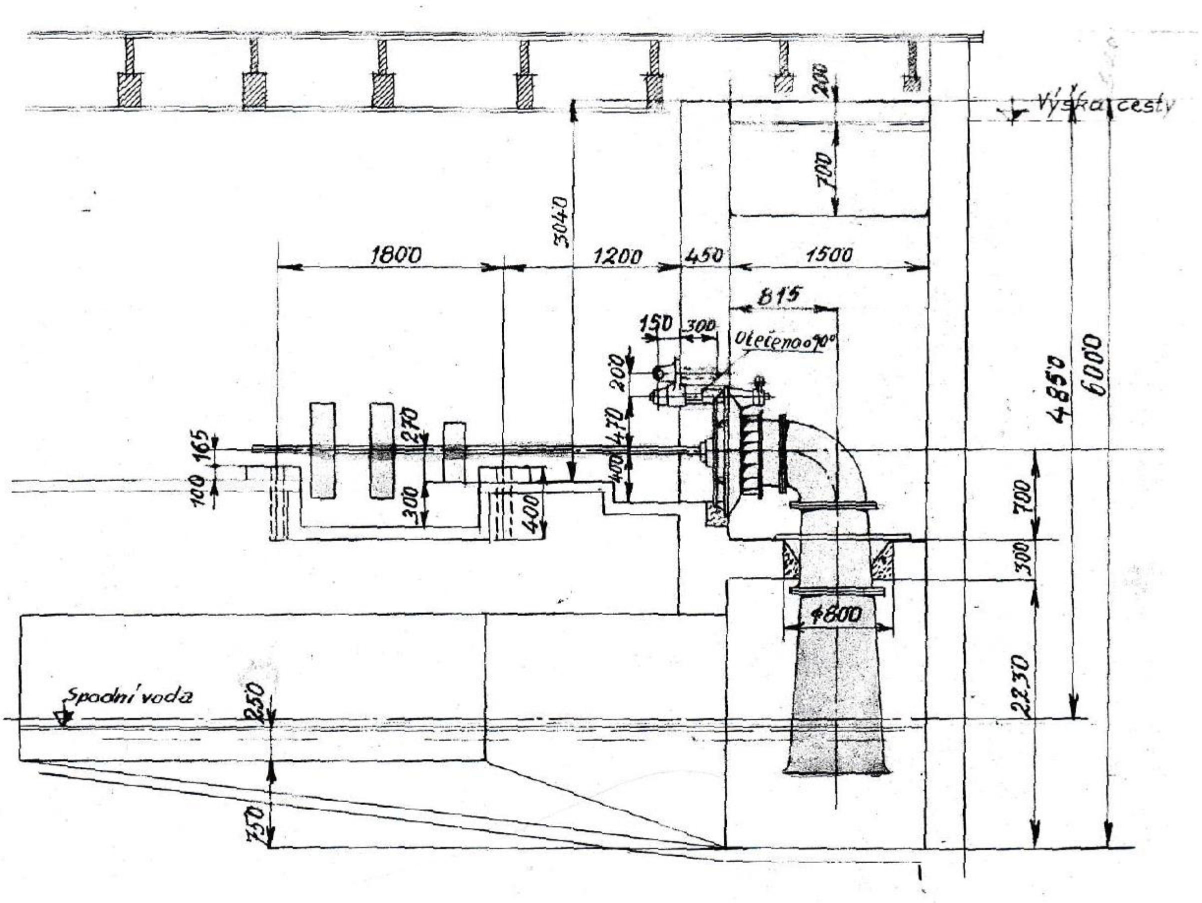


Fig. 6 Original drawing cut-out [2].

Table 5. Virtual prototype parameters.

Runner diameter.	350 mm
Number of runner blades old runner / new runner.	10 / 11
Number of guide vanes.	10
Maximum opening of guide vanes.	a_0 = 82 mm
Guide vane opening range.	a_0 = 29 - 77
Gross head.	4.8 m
Net head.	~ 4.6 m
Suction head .	2.2 m
Runner speed. Old / new runner	400 / 470 /min.
Flow range.	0.1 – 0.38 m^3/s

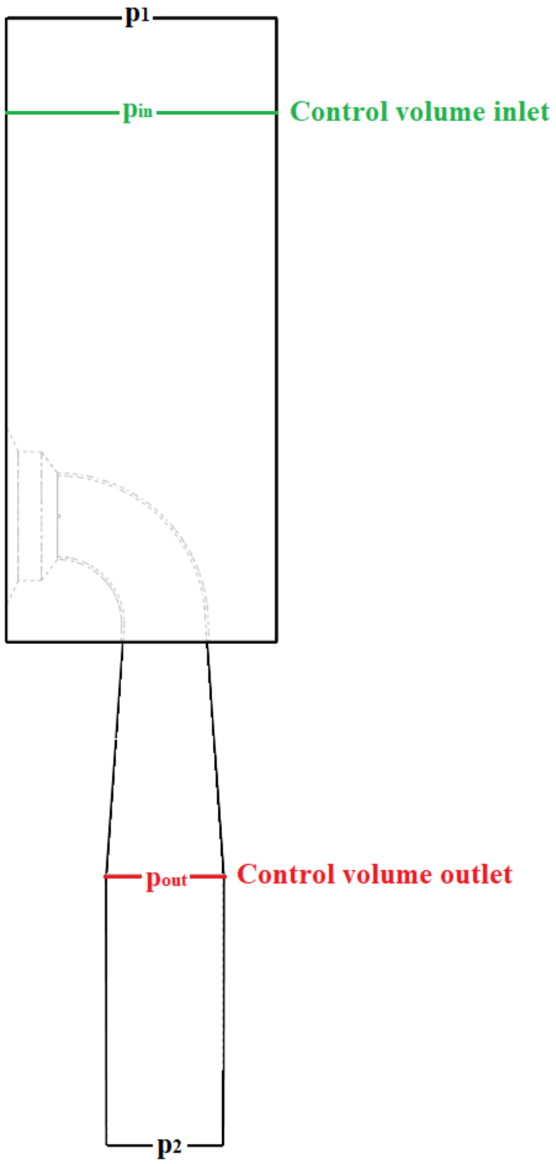


Fig. 7. Scheme of the virtual prototype with boundary conditions.

Well known technique of cyclic symmetry has been used to model guide vanes and runner. Here Autogrid by Numeca proved to be powerful. It is enough to have blade surface and meridian curves to get quality cyclically symmetric sector. See also fig. 4.

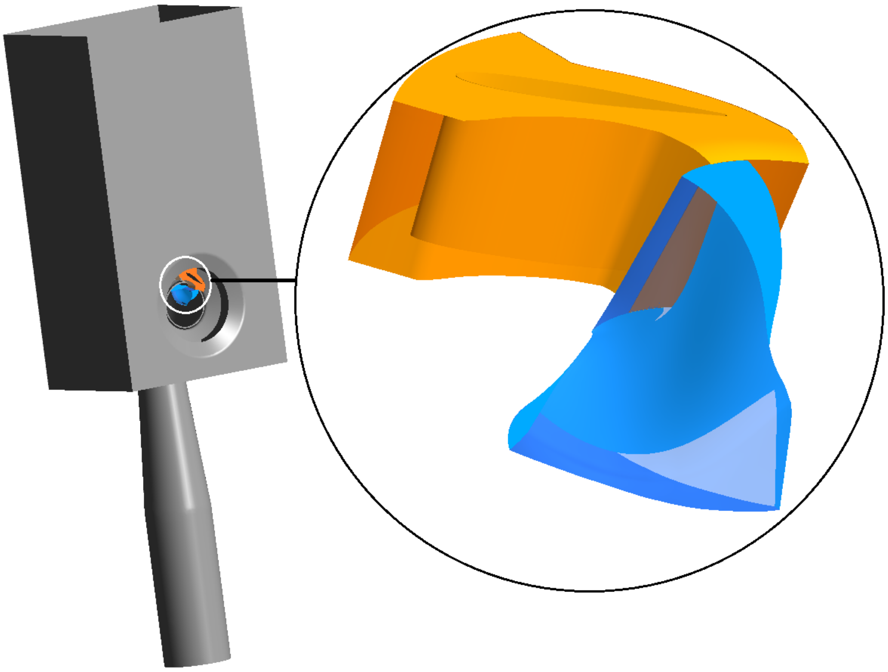
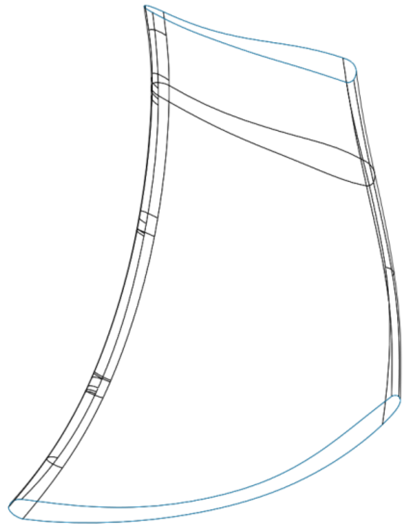
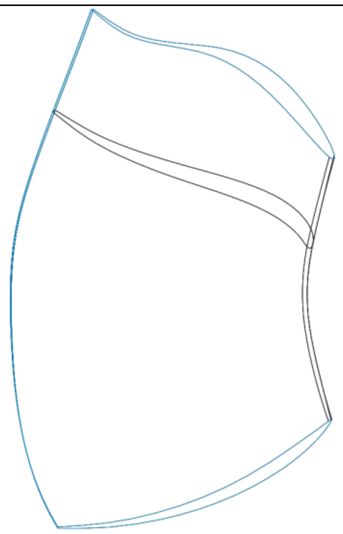


Fig. 8 Virtual prototype with detail of cyclic symmetry for guide vane and runner.

5.2 Method

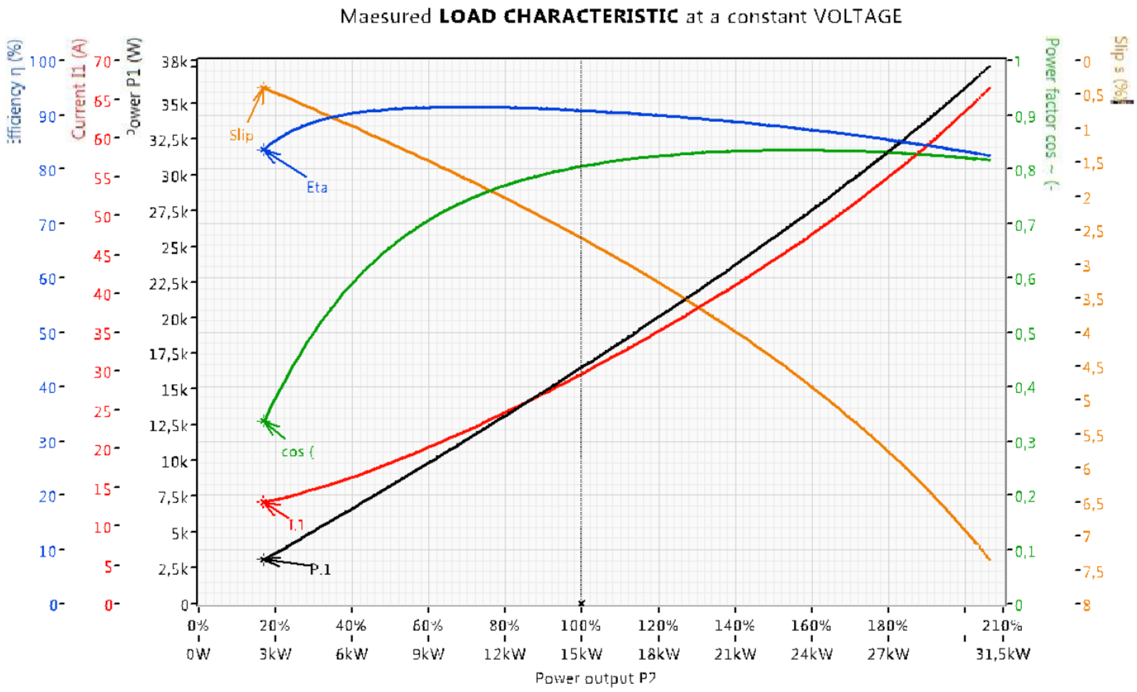
To get better result it is quite clear at first sight that more sophisticated wing shape can be used for runner blades. Following figures compare shapes of the old and new runner blade respectively. New blade is also thinner and has bigger surface area. Pressure side surface area of the old blade is 2.1 dm^2 and 2.5 dm^2 of the new one. There is even one more blade in new runner corresponding hub and shroud meridian. In this way it is possible to get higher torque at runner shaft axis with same hydraulic conditions which means higher efficiency and power output. Shape of the blade must be incorporated into the virtual prototype and verified by CFD. Then sequential iterations give desired result.

Table 6. Blade shapes with plane sections.

	
Old blade.	New blade.

6 Results and discussion

It has been mentioned that the goal of the refurbishment was to get generator power output of 13 kW. The generator is Siemens with maximum output of 15 kW. Efficiency dependence on load given by Siemens is according to following graph.



Graph 1 Blue curve is of the interest.

Generator is driven by a belt transmission with efficiency of 91 % [4] .

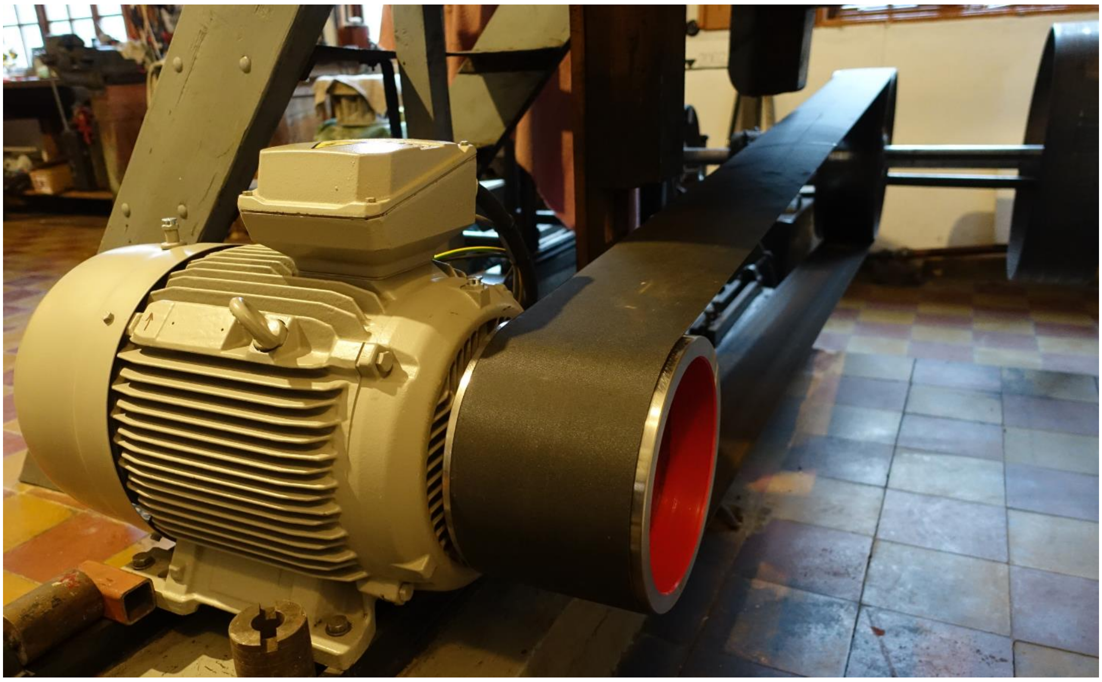


Fig. 9 Belt transmission from turbine to generator shaft.

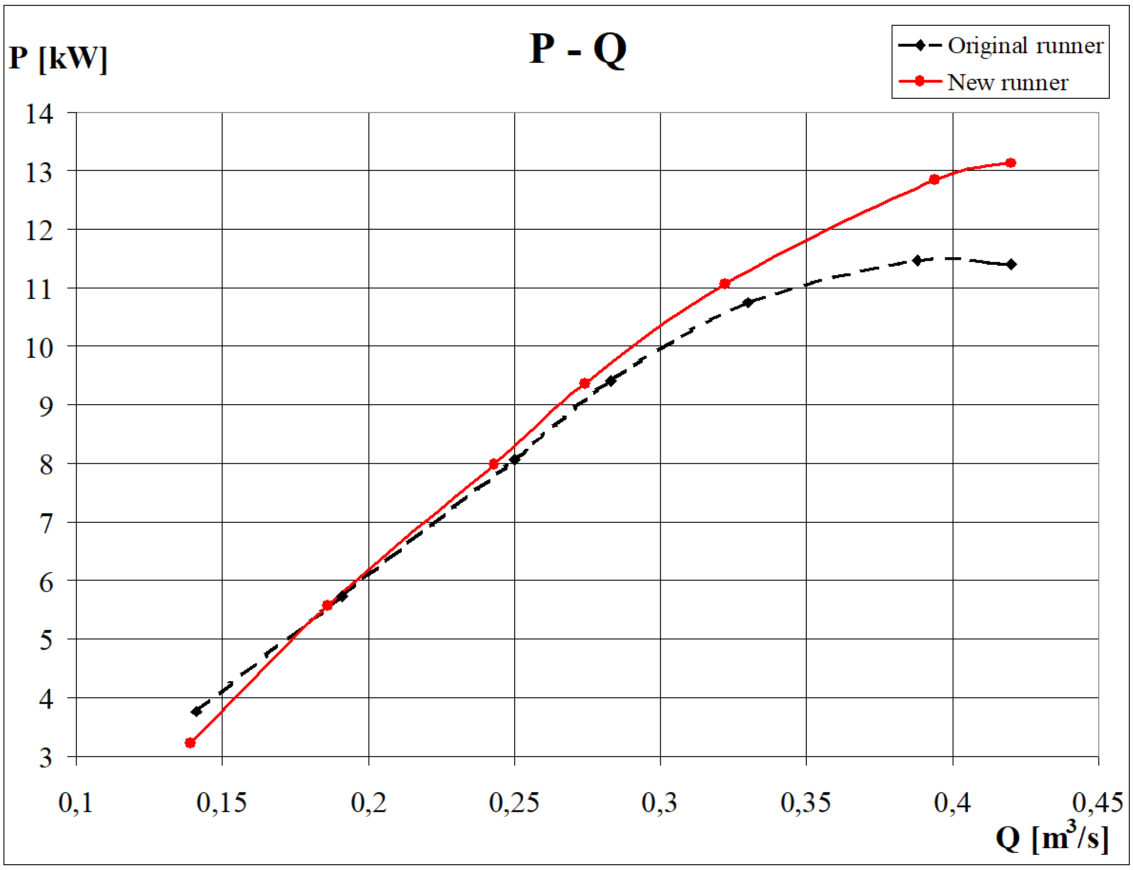
Table 7. Computed data of original turbo generator.

Q **[m³/s]**	**a₀** **[mm]**	**Guide vane opening [%]**	**ηH** **[%]**	**ηG** **[%]**	**η** **[%]**	**P** **[kW]**
0.141	29	35	74.9	87.0	59.3	3,8
0.191	38	46	81.3	90.0	66.6	5,7
0.250	48	59	86.4	91.0	71.5	8,1
0.283	53	65	88.1	92.0	73.8	9,4
0.330	62	76	86.3	92.0	72.3	10,7
0.388	75	91	78.3	92.0	65.6	11,5
0.420	82	100	72.0	91.9	60.2	11,4

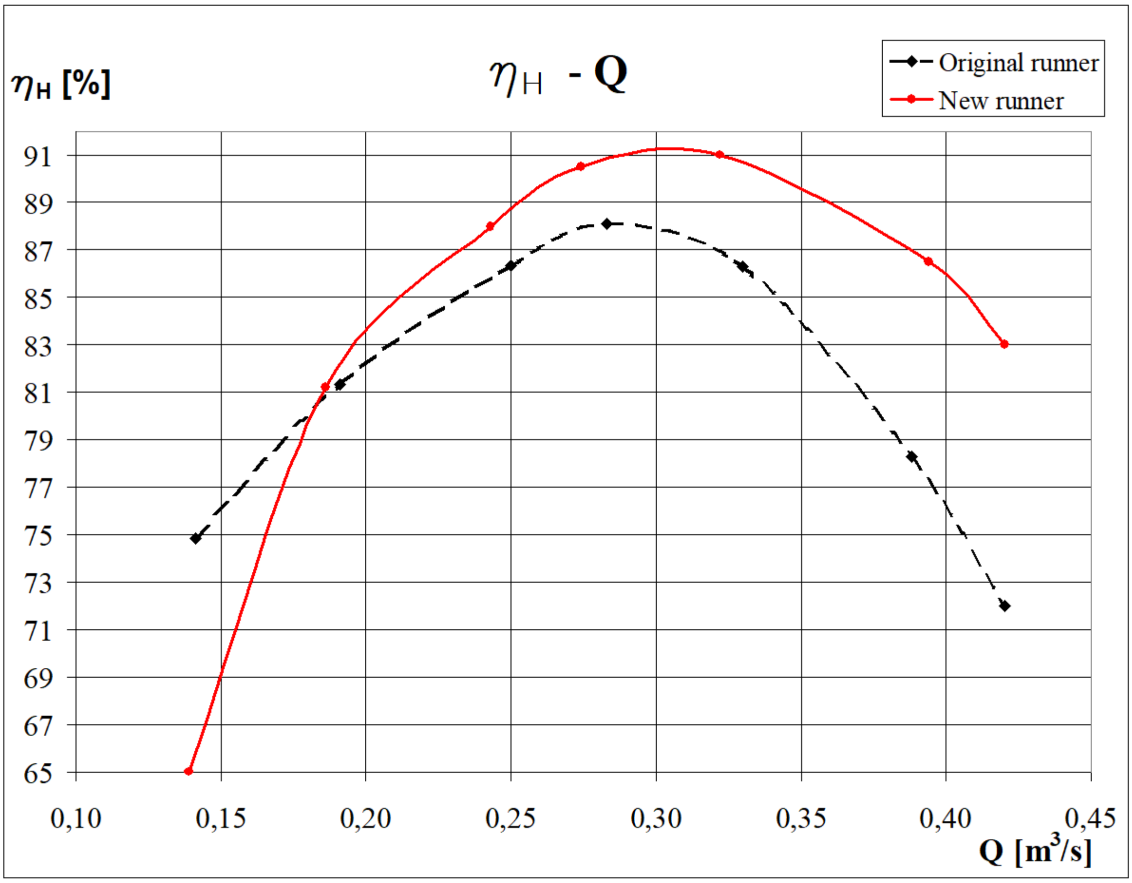
Table 8. Computed data of refurbished turbo generator.

Q **[m³/s]**	**a₀** **[mm]**	**Guide vane opening [%]**	**ηH** **[%]**	**ηG** **[%]**	**η** **[%]**	**P** **[kW]**
0.139	29	35	65.0	87.0	51.5	3,2
0.186	38	46	81.2	90.0	66.5	5,6
0.243	48	59	88.0	91.0	72.9	8.0
0.274	53	65	90.5	92.0	75.8	9,4
0.322	62	76	91.0	92.0	76.2	11,1
0.394	75	91	86.5	91.9	72.3	12,8
0.420	82	100	83.0	91.8	69.3	13,1

Tables above expressed in graphs show better the difference between old and new design.



Graph 2. Computed data.

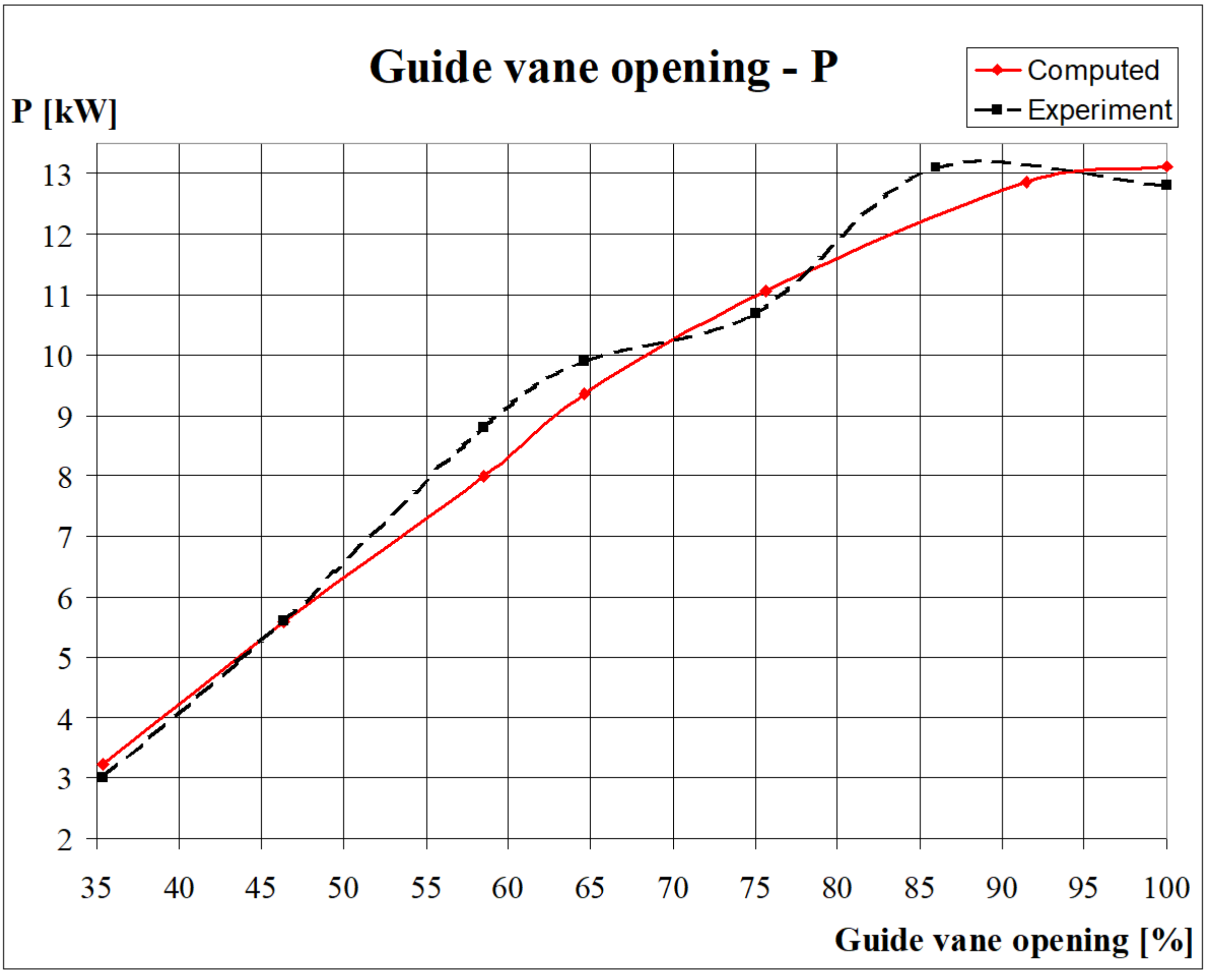


Graph 3. Computed data.

The only comparison of real and computed data can be done for dependency between guide vane opening and power output at given head. Reason is that no guarantee measurement takes place for such small turbines. See following graph. 100 % opening means a_0 = 82 mm as given in table 5.

To find out the reason for differences in efficiency of designs it is helpful to analyse streamlines going out of the runner into the draft tube. Ideal result is if they are not curly or even swirl. This should of course happen for operation point of the highest efficiency. Following comparison shows not only better shape of flow lines in the draft tube but also more uniform velocity which is desirable.

The other reason is the runner blade itself. See chapter 5.2 . Concerning this it is worth noticing the importance of having more blades with sophisticated wing shape. Table 10 shows the difference between streamlines both in contour and velocity near blade. Again for optimum operation point. For this reason an offset surface by 0.1 mm from hub, shroud and blade has been constructed. Significantly worse streamlines and bigger range of velocity on the old runner give rise to also worse flow into the draft tube.



Graph 4. Experimental vs. computed data.

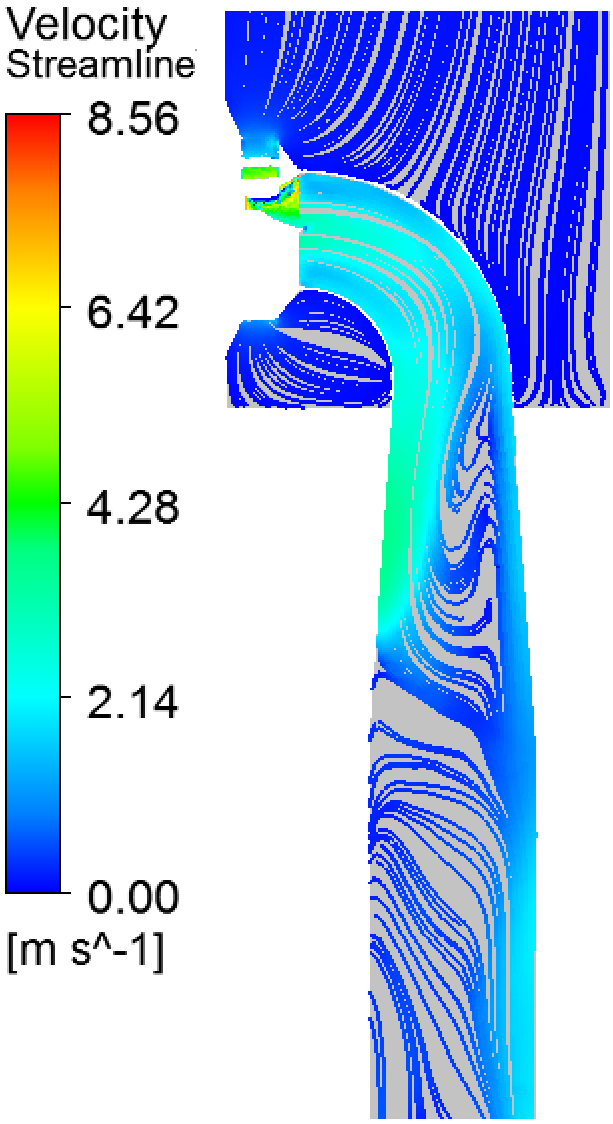
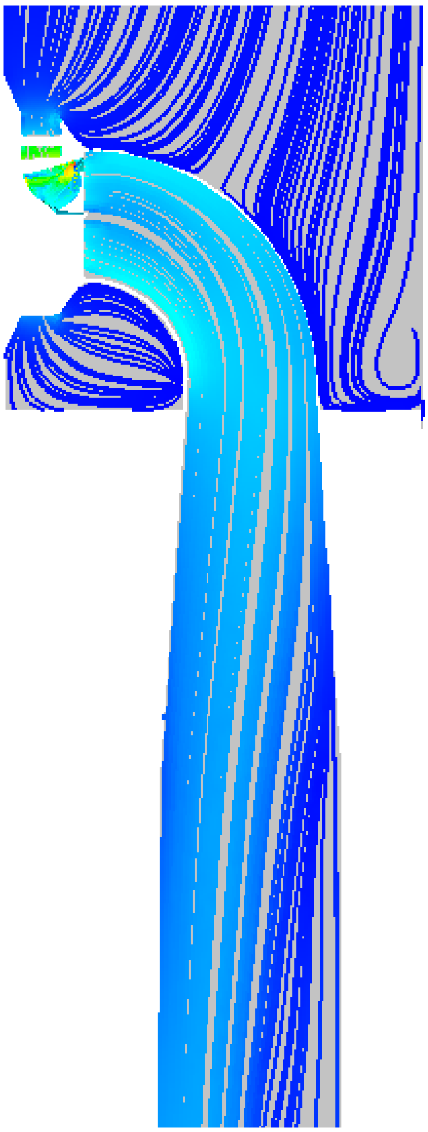
7 Conclusions

No guarantee measurement was done after the refurbishment. It is not performed in small power plants since it is expensive. Following figure 10 shows that the required electric power output value has been reached.

Fig. 10 Screen with the top power output.

More sophisticated shape of runner blades resulted in better transformation of pressure into the power output together with less loss. This altogether enables to get quality power output of the turbo generator. That is to say that it is partially also due to better efficiency of new generator but author has no data about efficiency of the old one.

Table 9. Stream lines in midplane for top efficiency.

	Old runner	**New runner**
		
Q	0.283	0.322
a_0	53	62
η_H	0.881	0.91

7.1 Method error estimation

Precise values of flow rate and net head were unknown. Error estimation comes from graph 4 and is taken for power output. Total power output P_{exp} read in the screen is taken as a gauge.

P_{comp} and P_{exp} stand for computed and experimental power output respectively.

$$\Delta P = P_{comp} - P_{exp} \quad [kW]$$

$$Error = \frac{\Delta P}{P_{comp}} 100 \quad [\%]$$

Table 10. Streamlines on an offset surface. Pressure and suction side respectively.

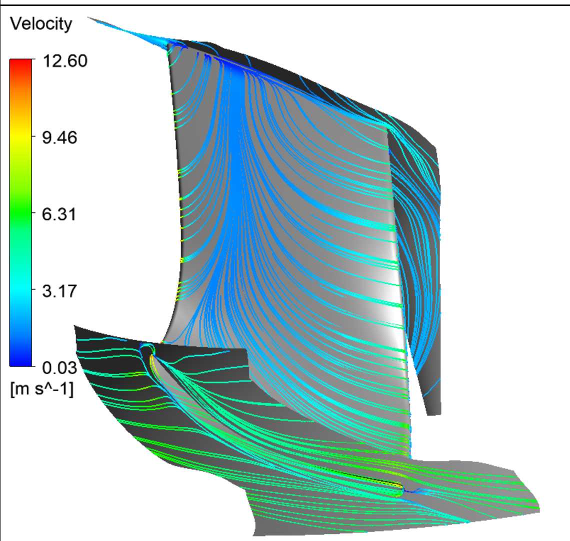
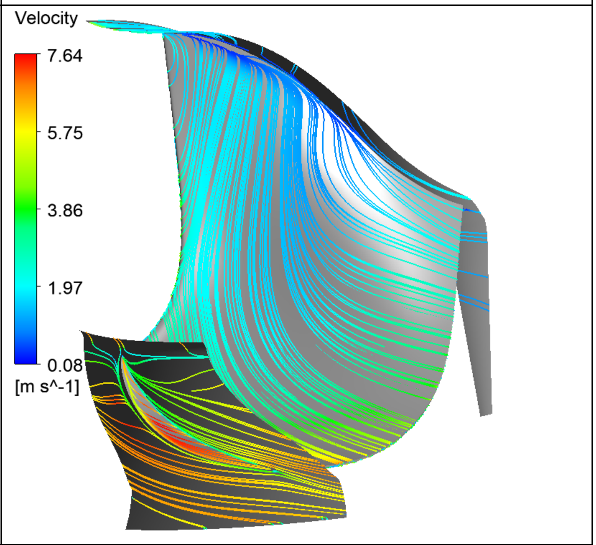
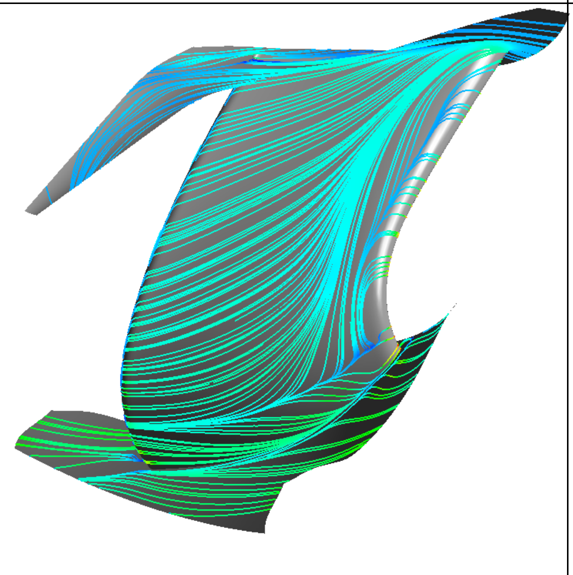
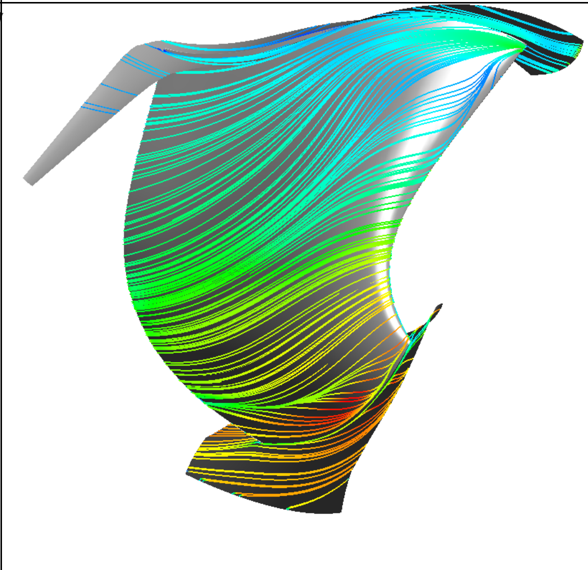
	Old runner	New runner
		
		
Q	0.283	0.322
a_0	53	62
η_H	0.88	0.91

Table 11.

Guide vane opening [%]	P_{comp} [kW]	P_{exp} [kW]	ΔP [kW]	Error+ [%]	Error- [%]	Error zero [%]
35	3.2	2.8	0.4	13		
46	5.6	5.6	0.0			0
59	8.0	8.8	-0.9		-10	
65	9.4	9.9	-0.5		-5	
75	11.0	10.7	0.3	3		
91	12.8	13.1	-0.3		-2	
100	13.1	12.8	0.3	2		

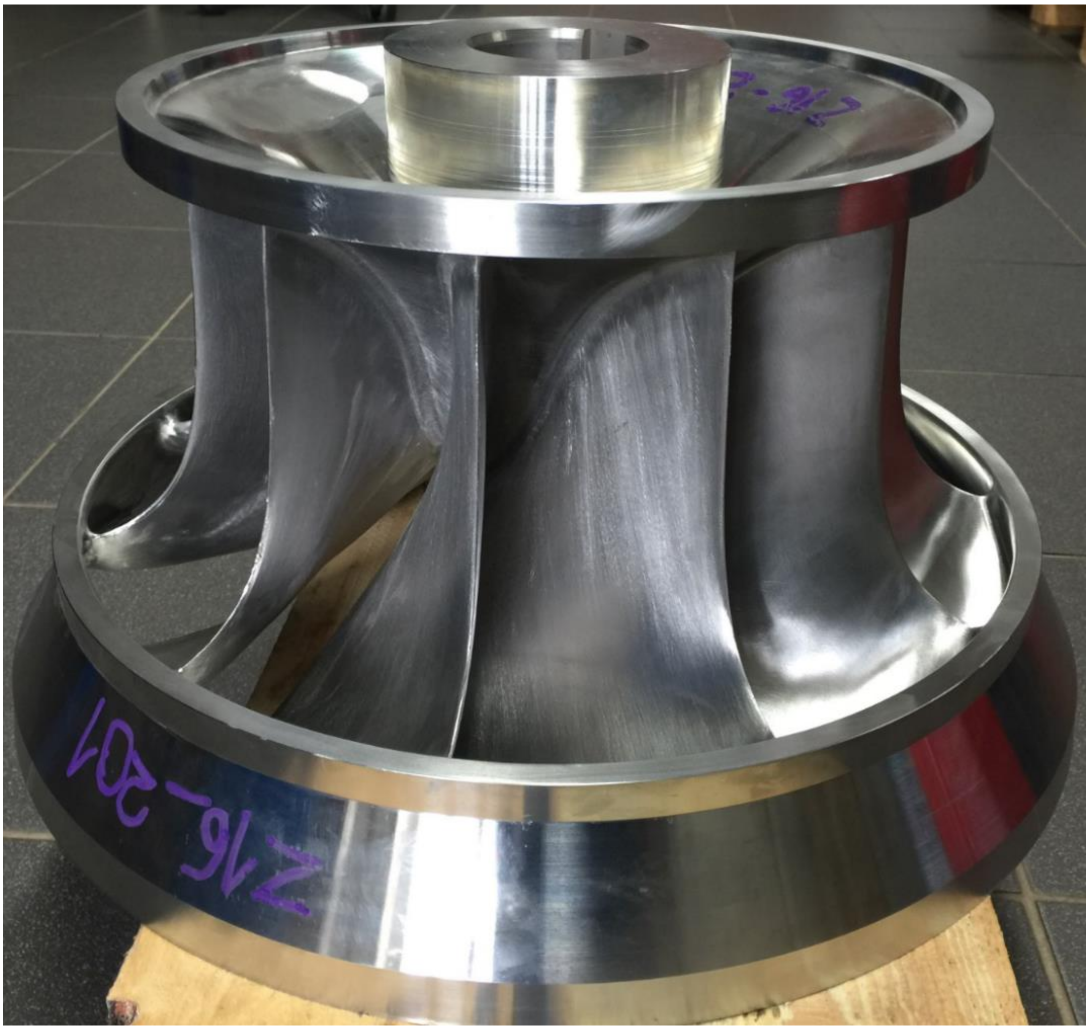


Fig. 11 New runner in the workshop after being machined.

Fig. 12 New runner mounted into the turbine.

$$\text{Average Error+} = \frac{\sum \text{Error} +}{4} = 4.4\ \%$$

$$\text{Average Error-} = \frac{\sum \text{Error} -}{4} = -4.4\ \%$$

Since usual error range in guarantee measurement is ± 2% [6] the result is acceptable.

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I am grateful to Mr. Čížek the owner of the small power plant for providing me with original documentation, contemporary and original data and photographs.

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