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STRENGTH CALCULATIONS PERFORMED ON THE SPIRAL CASING OF A FRANCIS TURBINE OPERATING IN SECONDARY CONTROL REGIME

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The operation of a hydro unit in control services, especially in secondary control or power control, leads to additional stresses on the Francis hydraulic turbine compared to the normal operating mode. These additional stresses result from the frequent closing and opening of the wicket gates, from the pressure pulsations that appear in the spiral casing and draft tube, from the frequent change of the vibration level in the bearings of the hydro unit, from minimum power to maximum power, etc. The paper presents the strength calculation of the spiral casing of the Francis turbine that equips the two hydro units (HA) of 76.5 MW, from the Ruieni hydroelectric power plant.

Keywords: turbine, spiral casing, Francis, stay vane, SolidWorks Simulation

1. Introduction

The spiral casing is the entrance component in a hydraulic turbine, that assures the uniform fluid flow for the wicket gate. The spiral casing has a spiral shape because the streamlines are logarithmic spirals. For medium and large heads, the frictional losses in the spiral chamber require special attention due to the high speeds and the long distance travelled by the fluid. There are many researches regarding the design, the flow and stress analyses of the spiral casing, aiming to reduce the frictional losses and mechanical stress. Muntean and Resiga use FLUENT to investigate the 3D flow in Kaplan turbine spiral casing and distributor [1]. Two stay vanes configurations are considered. The authors investigate the loading of the stay vanes for each configuration, as well as the circumferential non-uniformities of the velocity field at the runner inlet.

Balint and collaborators investigate the swirling flow optimization in the spiral casing and distributor of Kaplan hydraulic turbines [2]. The main goal was to reduce the unsteady blade loadings for the runners of Kaplan hydraulic turbines

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by using a technique of flow optimization in the inlet circuit of the turbine: spiral casing, stay and guide vanes. The methodology presented in the paper is applied for the case of Kaplan hydraulic turbines of Iron Gates I Power Plant with a runner diameter of 9.5 m and a runner speed of 71.5 rpm. The paper proposes a procedure of modifying the stay vane that is the most deprecated placed in the flow field by increasing the chord length for a constant curvature shape of the blade. The main advantage is that this solution inquires low costs and the technologic effort (by welding a bent metal sheet) is less expensive comparing to fabrication a new blade for the stay vane.

Dahal and collaborators [3] present the results from the design of spiral casing of Francis Turbine for Micro Hydro Applications. The main objective of the paper is to design the best configuration of cross section of spiral casing (trapezoidal, square and circular cross sections) which has a minimal pressure loss and can provide required inlet flow condition for stay vanes. The conclusion is that for micro Francis turbine where spiral casing dimension becomes big due to high flow and low head condition, trapezoidal spiral casing with free vortex configuration could be a better option, due to the ease of manufacturing.

Desai and collaborators use ANSYS CFX, ANSYS Design modeler, ANSYS ICEM CFD to validate the hydraulic design of spiral casing and stay vanes of Francis turbine [4]. From the investigation of the radial velocity distribution and comparison of various parameters for each configuration, conclusion can be made that the overall hydraulic behaviour of the elliptical configuration obtained from CFD analysis is found better than that of conventional circular configuration.

Panda and collaborators use ANSYS to analyse the stress around spiral casing of Francis turbine of a Hydel powerhouse by finite element method [5]. The stress patterns obtained by FEM analysis for a uniformly distributed load through a circular ring of spiral case are found generally comparable to the stress patterns found in the literature. Comparing with photo-elastic experimental method, FEM is a more convenient and less time-consuming tool for the analysis of complex structure like spiral casing.

Wu and collaborators perform a structural analysis of the embedded spiral casing in the three hydropower station to study the reliability of the spiral casing and the safety of the turbine unit under various working conditions [6].

Centre of Research in Hydraulics, Automation and Thermal Processes (CCHAPT), from the Babeș-Bolyai University, Romania, has so far performed tests and strength and fatigue calculations on hydro units with Kaplan turbines and Francis turbines operating in secondary control. From these tests and calculations and from the damages that took place at the hydraulic turbines in Romania, it was found an intensification of the fatigue phenomenon to which the regulating parts of the turbine and the spiral casing are subjected, due to the increase of mechanical stresses and especially of the fatigue cycles. Under these conditions, it is necessary to carry out a complex program of tests and strength and fatigue calculations for all hydropower units equipped with single-adjustment (Francis) and double-adjustment (Kaplan and bulb) turbines, before introducing them into a long-term in secondary control regime.

HPP Ruieni is also in this situation. The Ruieni hydroelectric power plant is of underground type with a high head. It is equipped with two hydro units with Francis turbines type FVM 78 – 326. The main technical characteristics of the Francis FVM 78 – 326 turbine are:

- Inlet runner diameter D = 2.6 m
- Nominal speed n = 428.6 rpm
- Maximum net head Hmax = 350 m
- Rated net head Hr = 326 m
- Minimum net head Hmin = 250 m
- Maximum power Pmax = 76.5 MW

The pressures for which the calculations were performed in the paper resulted from design parameters, governor guarantees, and tests performed on a hydro unit in transient regimes, as follows [7]:

- the most frequent pressure p_3 = 3.36 MPa, resulting from the rated head ensured by the hydroelectric power plant [7];
- the test pressure p_1 = 7.06 MPa, to which the spiral casing is subjected in accordance with the specific design rules;
- the maximum working pressure p_2 = 4.71 MPa, resulting from the governor guarantees. The governor over-pressure guarantee, at transient regimes, is 40% of the calculation pressure;
- the working pressure 1: p_4 = 3.724 MPa, resulting from HA 1 load rejection, at the maximum active power Pa = 76.5 MW and the head H = 339.39 m, with HA 2 stopped. Figure 1 shows the evolution of the pressure in the penstock, at the HA 1 load rejection from HPP Ruieni, at the above parameters ~~[7]~~;
- working pressure 2: p_5 = 3.9 MPa, resulting from the simultaneous load rejection of HA 1 and HA 2, from the active power P_a = 70 MW and the head H = 300 m. Overpressure in the penstock, at HA 1 is 4.5 bar, at HA 2 it is 4.0 bar. It can be estimated that at the maximum power P_a = 76.5 MW and the head H = 339.39 m, the overpressure is about 5.0 bar. Under these conditions, the calculation pressure for the spiral casing is 39.0 bar [8].

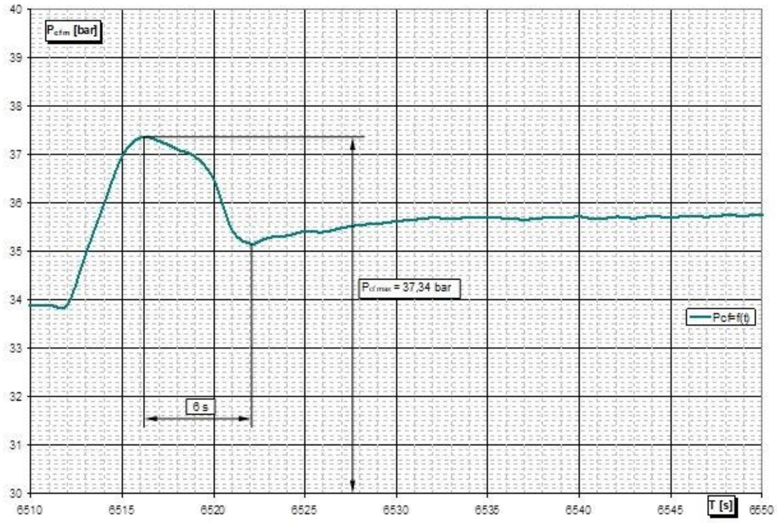

Fig. 1. The evolution of the pressure in the penstock, at the HA 1 load rejection

2. The spiral casing and stay vane geometry

The geometry of the spiral casing and the stay vane are shown in figure 2. The characteristic dimensions R0, AC and RM respectively depending on the φ angle of the spiral casing are presented in table 1 and figure 3.

The 3D geometry was modeled in SolidWorks by solids, resulting in a mass of 72,667 kilograms.

The spiral casing and the stay vane are defined by the following main parameters:

- the inner inlet diameter is ϕ1800 mm;
- the distance from the inlet section to section no. I is 2300 mm;
- the number of the spiral casing plates is 22;
- the angular extension of the spiral casing plates is 15^o;
- the spiral casing plates thickness is between g = 45 ÷ 30 mm;
- the stay vane has 12 columns, with a profile defined by coordinates;
- the height of the stay vane columns is variable in the range 265 ÷ 930 mm;
- the maximum thickness of the stay vane columns is 120 mm;
- the angular extension of the stay vane columns is 42^o.

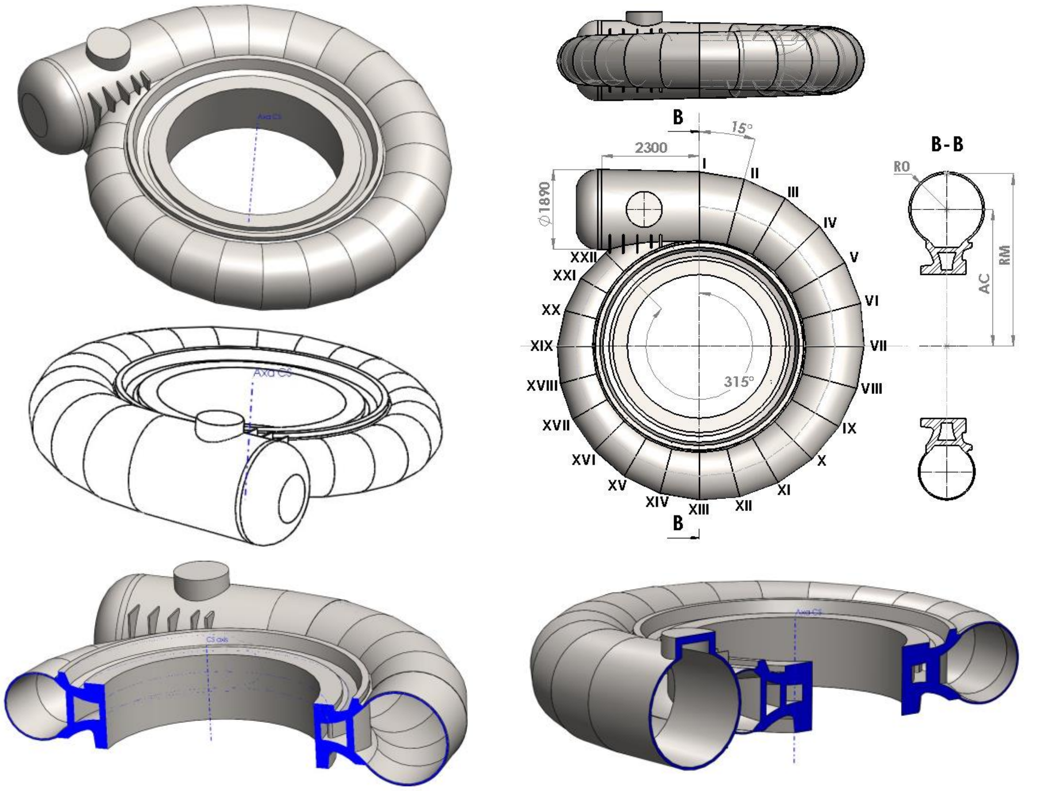

Fig. 2. The spiral casing and stay vane geometry

Table 1

The characteristic dimensions R0, AC and RM

Sect.	**U/M**	**I**	**II**	**III**	**IV**	**V**	**VI**	**VII**	**VIII**	**IX**	**X**	**XI**
φ	deg	0	15	30	45	60	75	90	105	120	135	150
RO	mm	845	830	810	795	775	760	740	725	705	690	670
AC	mm	3240	3222	3198	3179	3154	3136	3110	3091	3064	3044	3017
RM	mm	4085	4052	4008	3974	3929	3896	3850	3816	3769	3734	3687
g	mm	45	45 / 40	40	40	40	40 / 35	35/30	30	30	30	30
Sect.	**U/M**	**XII**	**XIII**	**XIV**	**XV**	**XVI**	**XVII**	**XVIII**	**XIX**	**XX**	**XXI**	**XXII**
φ	deg	165	180	195	210	225	240	255	270	285	300	315
RO	mm	655	635	620	600	580	565	545	530	510	495	475
AC	mm	2996	2967	2945	2914	2881	2855	2819	2789	2744	2704	2631
RM	mm	3651	3602	3565	3514	3461	3420	3364	3319	3254	3199	3106
g	mm	30	30	30	30	30	30	30/25	25	25	25	30

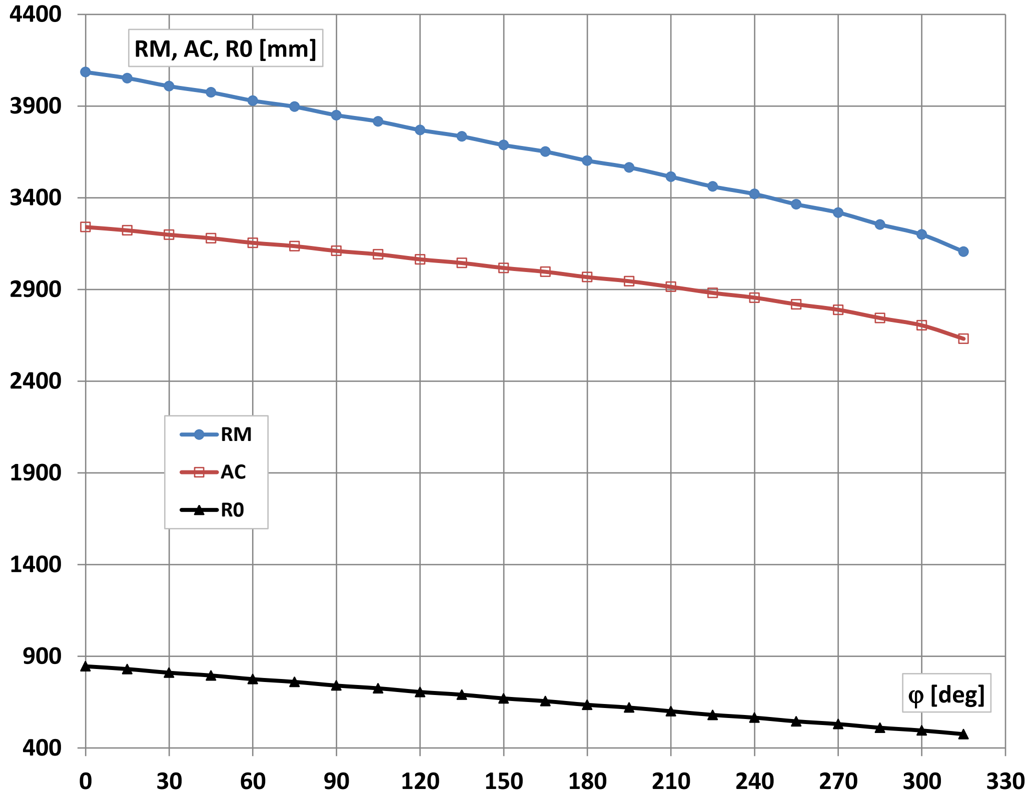

Fig. 3. The spiral casing and stay vane geometry

3. The boundary conditions

The numerical simulation aims to test the spiral casing and the stay vane in terms of resistance. The calculations were performed through the Simulation module integrated in the SolidWorks software, for the following five values of the internal pressure: maximum test pressure p_1=7.06 MPa, maximum working pressure p_2=4.71MPa, nominal pressure p_3=3.36MPa, working pressure 1 p_4=3.724 MPa and working pressure 2 p_5=3.9 MPa. For the structural analysis the following conditions are imposed, figure 4:

○ the spiral casing will be fixed on the lower ring surface of the stay vane; **Fixed Geometry** constraints will be applied in this area, which imposes the 0 value of translations for the selected area.

○ the entrance to the spiral casing will be closed with an elliptical inlet cover, with inner diameter ϕ1800 mm and thickness 50 mm;

○ the exit from the stay vane will be closed with a cylindrical sealing ring, with 110 mm thickness, placed on the periphery of the exit area.

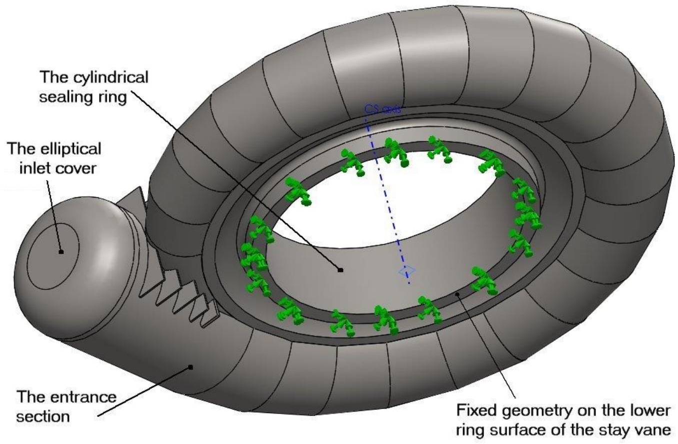

Fig. 4. The spiral casing and stay vane geometry

The load imposed for the resistance calculation will be the internal water pressure, for the previously specified pressure values. Consequently, in accordance with the standard procedures for spiral casing calculating resistance, the constant pressure will be applied on all the wet areas inside the spiral casing and stay vane, respectively, figure 5:

- inside the spiral casing plates and the internal areas of the stay vane.
- on the inside surfaces of the elliptical inlet cover and the cylindrical sealing ring.

The mesh of the geometry is presented in figure 6, consisting of 76,760 nodes and 41,031 finite elements.

The material used in the calculations is AISI 1020 steel, selected from the SolidWorks library, with the characteristics presented in table 2. Compared to the value of the **Yield Strength** σ_c = 351.57 MPa, we will consider the value of the admissible resistance $\sigma_a = 0.66 \cdot \sigma_c = 0.66 \cdot 351.57 = 232$ MPa.

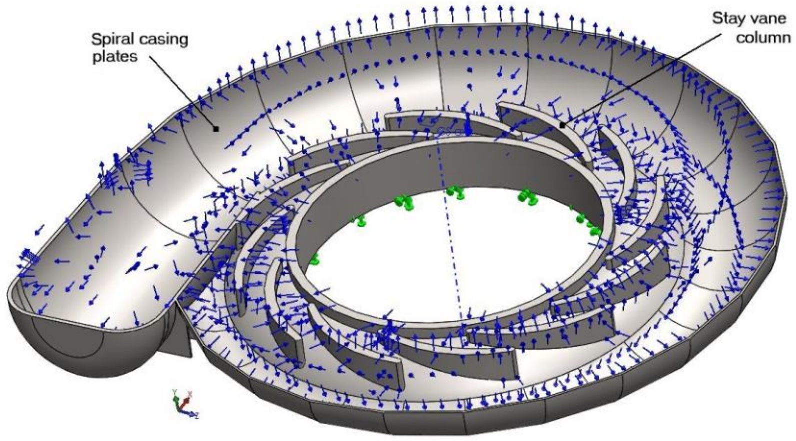

Fig. 5. The internal water pressure applied inside the spiral casing and stay vane (half section)

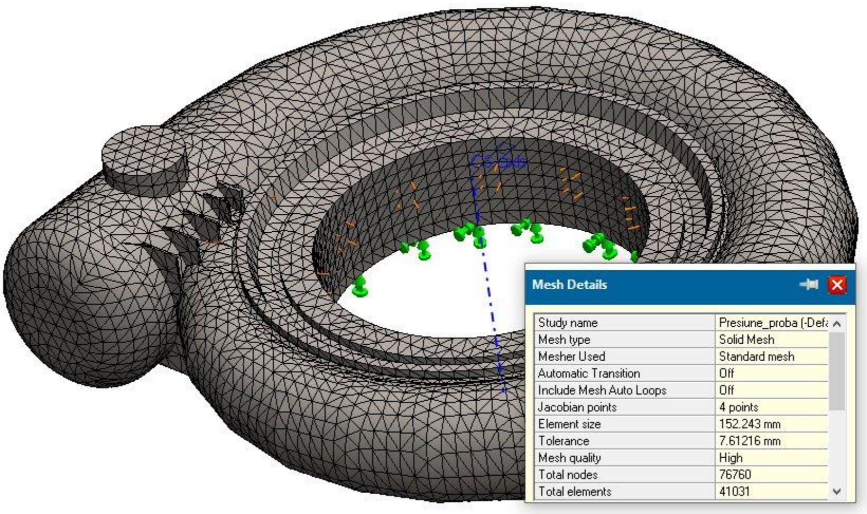

Fig. 6. The mesh of the geometry

Table 2

The characteristic of the AISI 1020 steel

Elastic Modulus	$2 \cdot 10^5$	MPa
Poisson's Ratio	0.29	N/A
Shear Modulus	$0.77x10^5$	MPa
Mass Density	7900	kg/m^3
Tensile Strength	420.5	MPa
Yield Strength	351.57	MPa

4. The numerical results

The numerical results are presented graphically only in the figures 6 ÷ 9, as color maps corresponding to the von Mises stress $\sigma_{VM\ max}$ respectively to the maximum displacements δ_{max} are summarized in table 3. It is observed that the maximum stresses and displacements are located in the area of the elliptical inlet cover, figure 4, area without interest, because this cover is used only for the maximum test pressure performed on site for a short time, after the final assembly and only once during existence of the spiral casing. For the area of the entrance section in the spiral casing, figure 4, the region with the highest effective values inside the spiral casing, the maximum von Mises stress $\sigma_{VM\ max}$ = 308.99 MPa for the maximum test pressure, performed only once during the existence of the spiral casing, is greater than the admissible resistance σ_a = 232 MPa, but below the **Yield Strength** σ_c = 351.57 MPa. Given the short duration of this test compared to the operating time of the turbine as well as the fact that the maximum pressure test is performed only once on site - after the final assembly, we consider that the value resulting from the simulation is not dangerous. For the studies (p_1, p_2, p_3 and p_4) the maximum von Mises stresses are below the admissible resistance σ_a, at the values 205.54, 146.57 MPa, 212.42 MPa and 222.46 MPa, resulting the factor of safety coefficients 1, 13, 1.58, 1.09 and 1.04, respectively. Also, the displacement values are small, in the range 1.702 ÷ 2.1844 mm.

Table 3

The summarized numerical results

Pressure type	Parameter	Symbol	U/M	Parameter location (figure 4)	
				The elliptical inlet cover	The entrance section
Maximum test pressure	Internal pressure	p_1	MPa	7.06	7.06
	Von Mises stress	$\sigma_{VM\ max}$	MPa	657.09 (figure 6)	308.99 (figure 7)
	Displacement	δ_{max}	mm	9.056 (figure 8)	2.557 (figure 9)
Maximum working pressure	Internal pressure	p_2	MPa	4.71	4.71
	Von Mises stress	$\sigma_{VM\ max}$	MPa	440.129	205.54
	Displacement	δ_{max}	mm	6.06	1.702
Nominal pressure	Internal pressure	p_3	MPa	3.36	3.36
	Von Mises stress	$\sigma_{VM\ max}$	MPa	314.0	146.57
	Displacement	δ_{max}	mm	4.323	1.2142
Working pressure 1	Internal pressure	p_4	MPa	3.724	3.724
	Von Mises stress	$\sigma_{VM\ max}$	MPa	346.693	212.42
	Displacement	δ_{max}	mm	4.777	2.0858
Working pressure 2	Internal pressure	p_5	MPa	3.9	3.9
	Von Mises stress	$\sigma_{VM\ max}$	MPa	363.072	222.46
	Displacement	δ_{max}	mm	5.003	2.1844

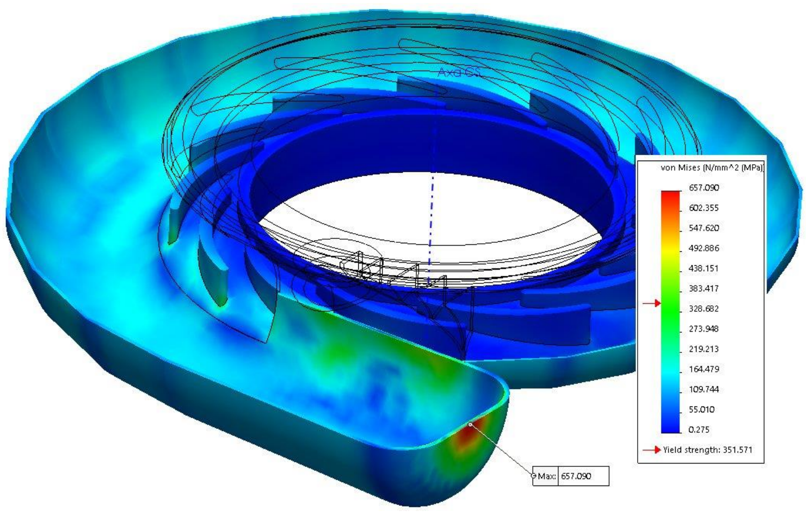

Fig. 6. The von Mises stress for maximum test pressure p_1 = 7.06 MPa The elliptical inlet cover location - $\sigma_{VM\ max}$= 657.09 MPa

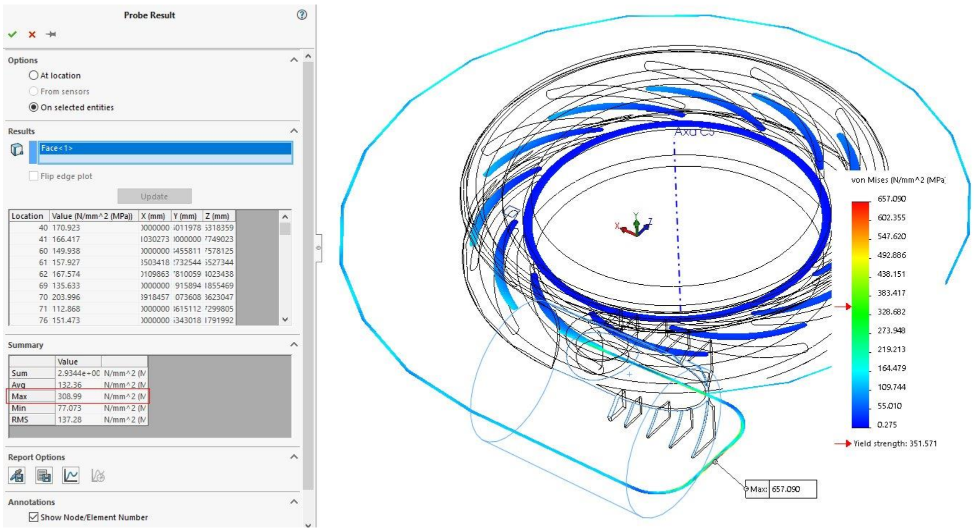

Fig. 7. The von Mises stress for maximum test pressure p_1 = 7.06 MPa
The entrance section location - $\sigma_{VM\ max}$= 308.99 MPa

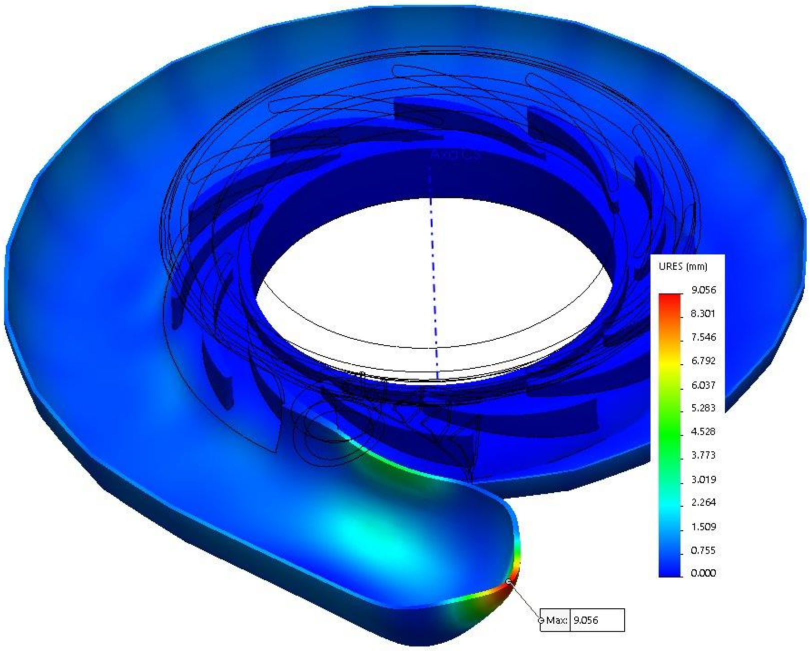

Fig. 8. The displacement for maximum test pressure p_1 = 7.06 MPa
The elliptical inlet cover location - δ_{max} = 9.056 mm

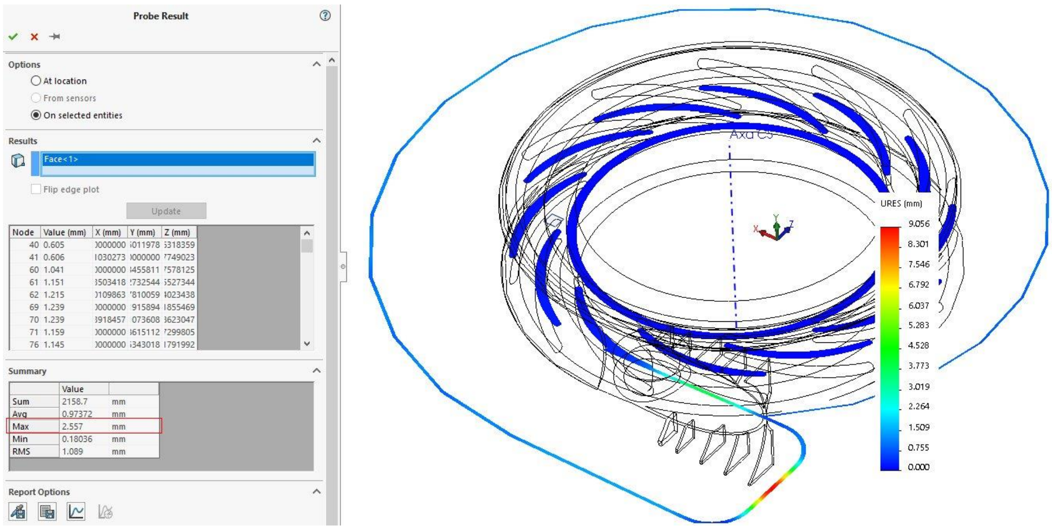

Fig. 9. The displacement for maximum test pressure p_1 = 7.06 MPa
The entrance section location - δ_{max} = 2.557 mm

6. Conclusions

- The secondary control introduces additional stresses on the spiral casing of a hydraulic turbine, which require strength calculations for all pressures that occur in normal operating regimes and transient regimes.
- For the hydraulic turbine from HPP Ruieni, the calculation pressures of the spiral casing were established, from the HPP parameters, the test pressure, the governor guarantees and from the tests performed in load rejection transient regimes.
- The equivalent von Mises stresses, for all calculation regimes related to the allowable stress, give safety coefficients higher than 1 or by reference to the Yield strength of the material of the spiral casing, the safety coefficients are over 1.5.
- The fatigue limit (σ-1) of the material from which the spiral casing is made is about 220 MPa. For this fatigue limit (σ-1), fatigue safety coefficients are lower than 1.5 [9]. For the components of the turbine of major importance, such as the spiral casing, the new design rules [9] recommend safety coefficients at fatigue greater than 1.5. Under these conditions for the analysed spiral casing, measurements on the hydro unit to determine the fatigue cycles and fatigue calculations and crack propagation are required.

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