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THE ROLE OF EXPERIMENTATION IN HYDRO-POWER PLANT BUILDING

The experiences and perspectives of development, production and field service in Litostroj

Vortex core in turbine draft tube

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1. INTRODUCTION

Experimental work and related measurements are one of the most important segment in research and development, construction and installation of a turbine. In Turbine Factory Litostroj a lot of experiences has been gained in this field in the last decades. A huge number of experimental results on turbine prototype all over the word is an important data base for analysis of these measurements, for calculation and development of new physical and mathematical models, etc. In this article a short snap shot through the experimental work in Litostroj will be presented.

Measurements in hydroelectric power plants can be roughly divided in two ways: by measurements of functional performances and measurement of physical properties - parameters, that characterize the hydro-power plant.

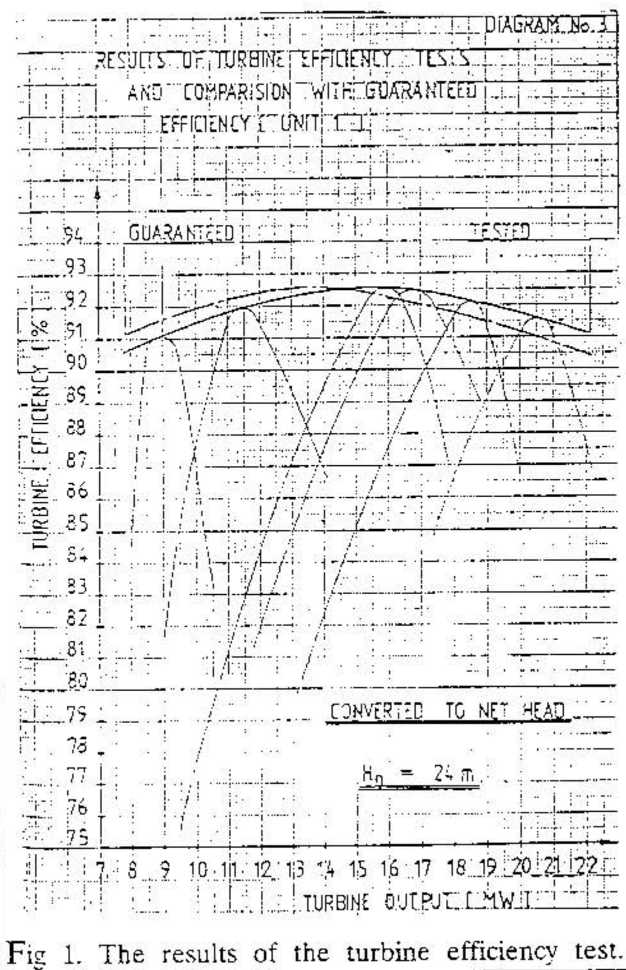

Fig 1. The results of the turbine efficiency test.

A. Measurements of functional performance :

1. *Standard measurements:*
 - development of the hydraulic shape of the runner and testing the turbine model

 - prototype performance test (by IEC 41 and other standards) - an example is shown on Fig. 1, and

 - special prototype measurements (particularly define by customers)

2. *Measurements of unexpected incidents* on hydro-power plants, diagnosis and solutions of problems.

3. Measurements of *research and development* - testing of the theoretical (mathematical) models of a turbine and its principal building blocks.

B. Measurements of physical parameters:

1. Oil hydraulic parameters (measurements of thickness of the oil film, measurements of parameters for turbine controller)
2. Hydrodynamic measurements (measurements of discharge, pressure in water system, velocity measurements)
3. Elastomechanic measurements (loads and displacements of the steel constructions - static and dynamic measurements).
4. Noise measurements.

2. SOME EXPERIENCES

2.1 Standard measurements

2.1.1 Model tests

Model test measurements are the most important part of the turbine hydraulic development, specially measurements of hydraulic shape of the runner blades. This part of development is performed in close co-operation with institute for hydraulic research Turboinstitut in Ljubljana.

Model test rigs are also important in a case, where at the prototype there is no opportunity to make an efficiency test according to ICE 41.

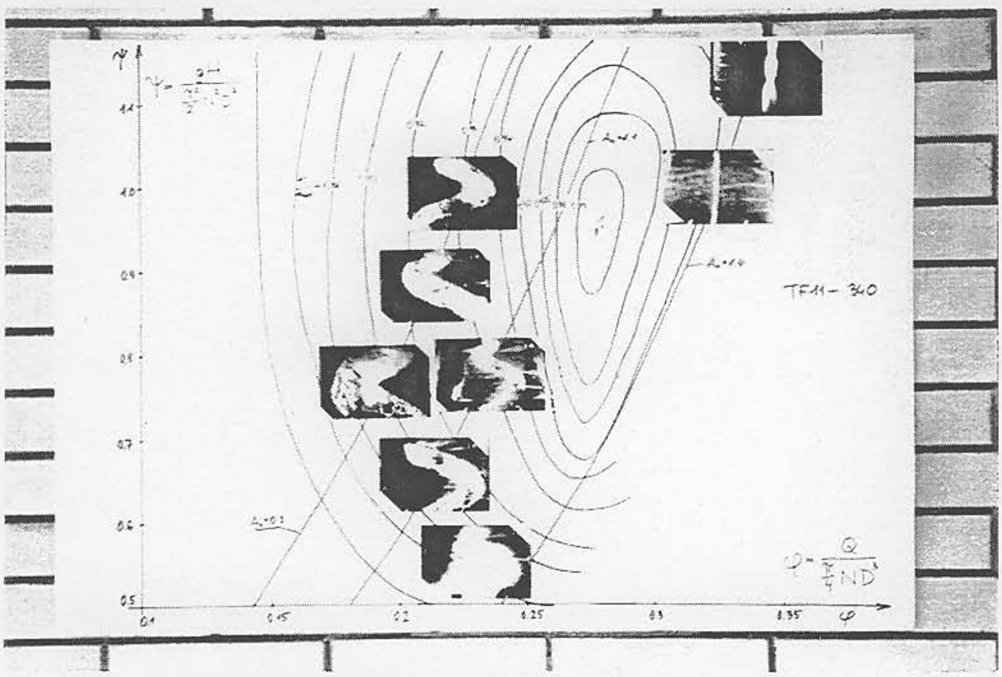

Fig 2. Shape of the vortex core motion in a draft tube

2.1.2 Prototype measurements

The most important measurements in the starting phase of operations of a hydro-power plant are the ones that allow the settlement of a time dependent guide vane opening as well as the settlement of runner blades and also the time constants of the speed and level governor are to be adjusted according to prototype measurements. To determine the cam relationship with one of the Index methods the efficiency has to be measured. Here, Winter-Kennedy is one of the most frequently methods of relative measurement of the efficiency (Fig 1). Our experiences are, that runout of the runner shaft and vibration of bearings has to be measured in the frequency range up to 1/4 of the runner speed. Results of these vibrations can show us, for example, that a vortex core appears in turbine draft tube (Fig 2).

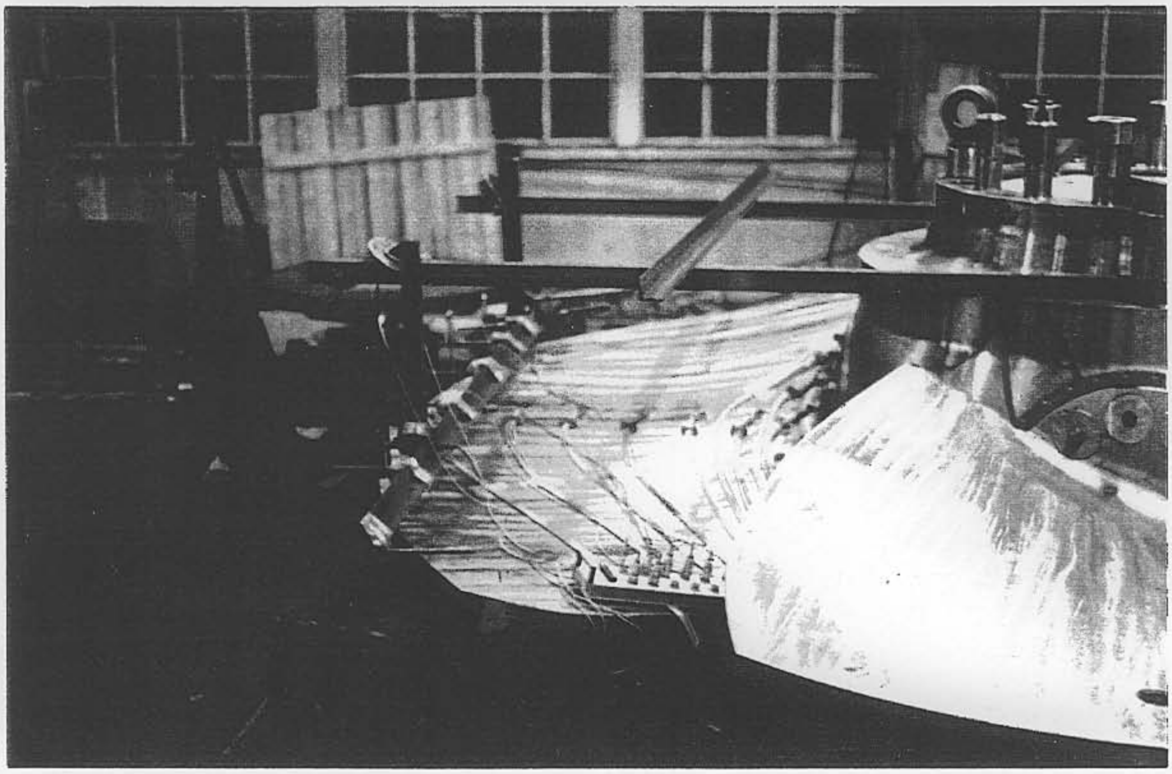

Fig 3. Static deformation of the runner blade for HPP Fala - Slovenia

2.1.3 Special measurements

Second category of the turbine prototype efficiency measurements is the one where special requirements are defined in a contract between both parties. An example is the measurement for deformation check of the runner blades of HPP Fala (Fig. 3). In another example an absolute measurement of discharge with current meters is presented. In this case an output of the turbine is measured on electric generator. Such measurement is very sophisticated: all current meter has to be calibrated according to ISO standard requirements. Generator efficiency has to be tested separately, etc.

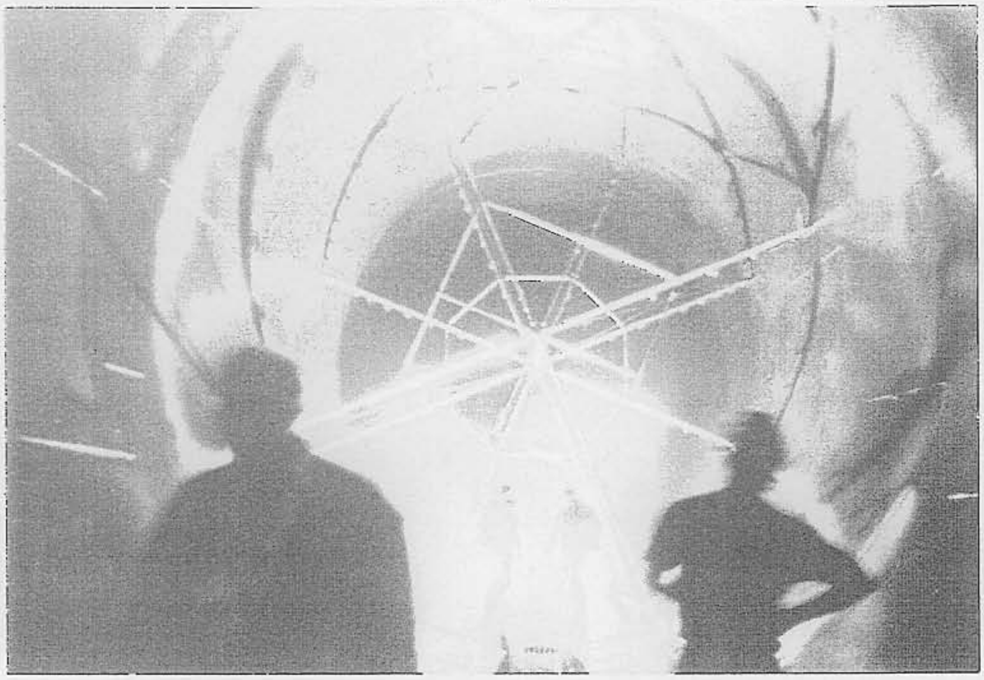

Fig. 4 HPP Stratos - cross section of the current meters

Special attention while measuring a huge discharge and high velocities has to be put into vibration problem of the current meter cross section. As example on HPP Stratos with highest velocities up to 7 mps and discharge about 250 m^3/s the construction of the cross section has been calculated with Ansys software and modified according to the Karman vortex (Fig. 5). In these measurements 8 strain gauges has been mounted on the cross arms and arm vibration has been continuously monitoring while measuring the velocity field.

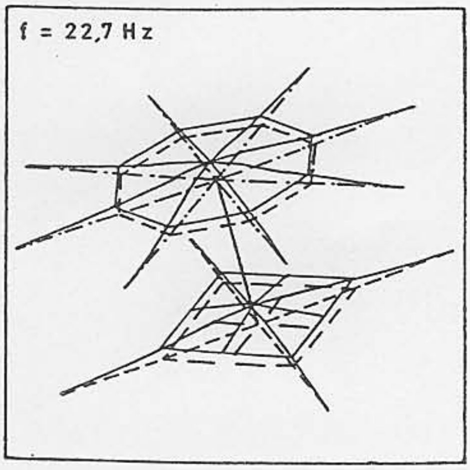

Fig. 5 Calculation of the cross section

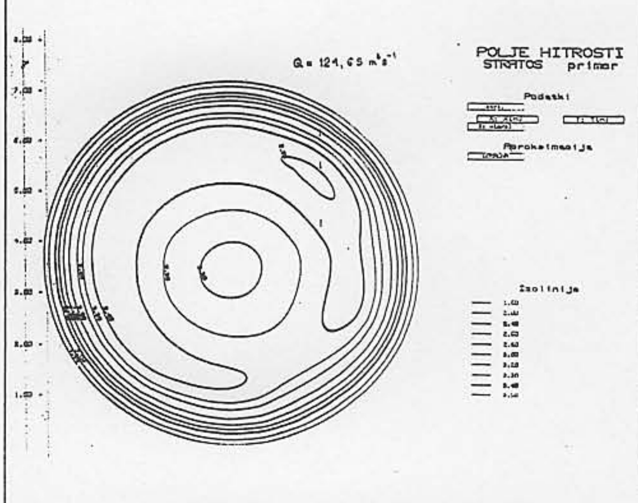

Fig. 6 HPP stratos - velocity field

2.2 Measurements of unexpected incidents

Beside turbine prototype measurements a large number of measurements are needed for diagnosis and repair of unexpected incidents on hydro-power plants. The most frequent problems of this type are cracks on the Pelton and Francis runners as well as on the stay vanes. All the results are reported and saved in files for later improvement studies.

2.3 Research and development measurements

In last few years beside continuos work on improvement of the turbine efficiency also a considerable number of theoretical research works have been done and published in reviews and presented on conferences. The research has been concentrated on water hammer in pumping system (Ref. 5,6) and hydro-power plants (Ref. 7), hydraulics auto-oscillations (Ref. 8,9) and draft tube surge (Ref. 10). Joint research on transient cavitating flow in pipe lines between Litostroj and the University of Adelaide has been performed (Ref. 11,12,13,14). There are several other research works carried out regularly, e.g. calculations of the critical speed including gyroscope effects (Ref. 1,2) and the vortex core in

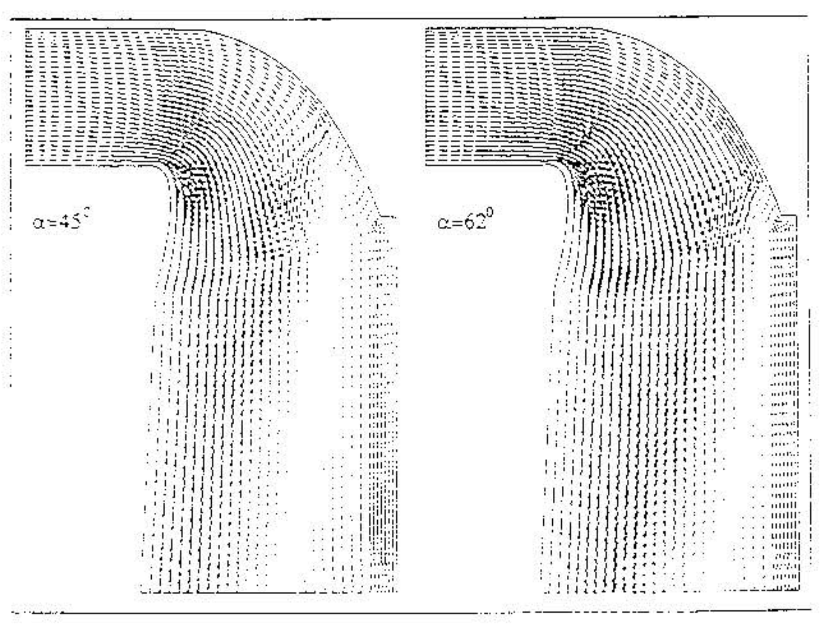

Fig. 7 Calculation of the velocity field in Francis turbine draft tube for two guide vane opening - a zone of reverse flow is presented.

draft tube of a Francis turbine (Ref. 3,15,16,17) as well as theoretical calculations of the Pelton and Francis runner oscillations.

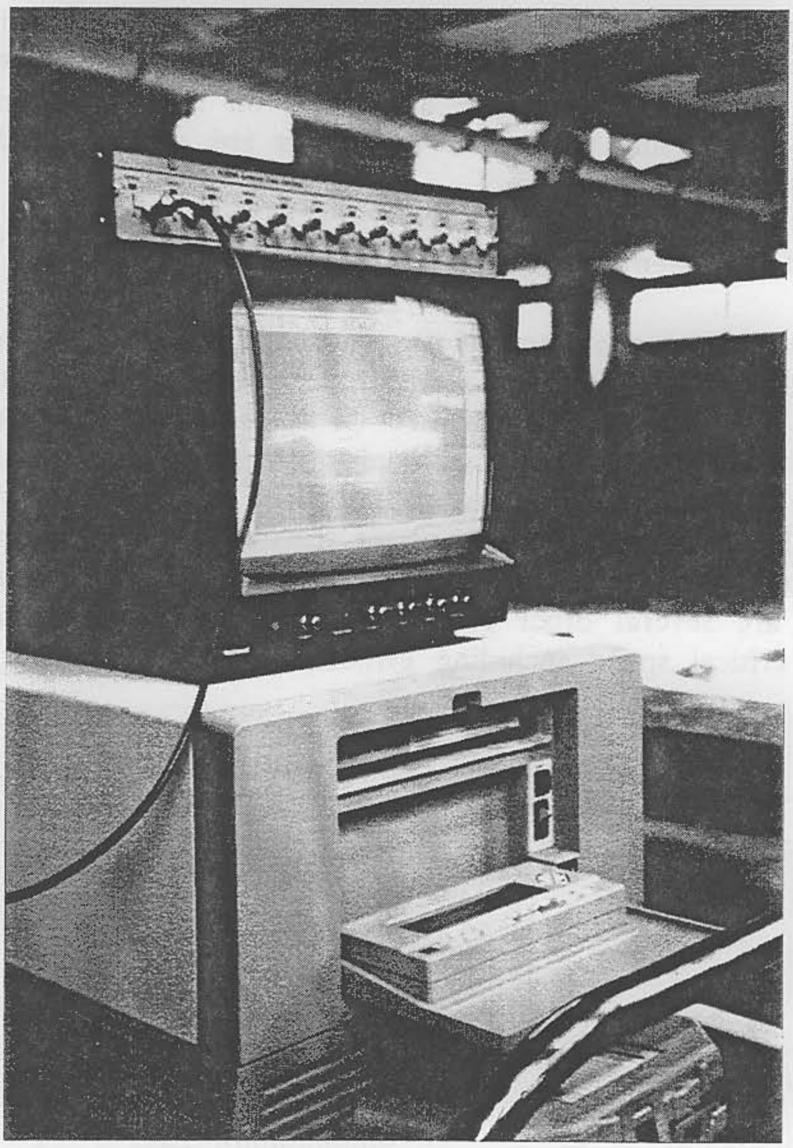

Fig. 8 Flow visualization of column separation in the pipeline carried out at the University of Adelaide (Ref. 13,14)

3. CASE STUDY: The control of spherical valve at the inlet of the turbine.

3.1 Brief description

In high pressure turbine power plant, spherical valve (Fig. 9) is used as the inlet turbine valve. Usually each valve has a parallel mounted small valve with function of equalizing the pressure before and after the valve. After some years of operation very often appears, that this small valve cannot equalize the pressure any more. In such a case, a pressure oscillation occurs while opening and closing the large spherical valve. A large amplitude of this oscillations occurs especially at small opening of the valve.

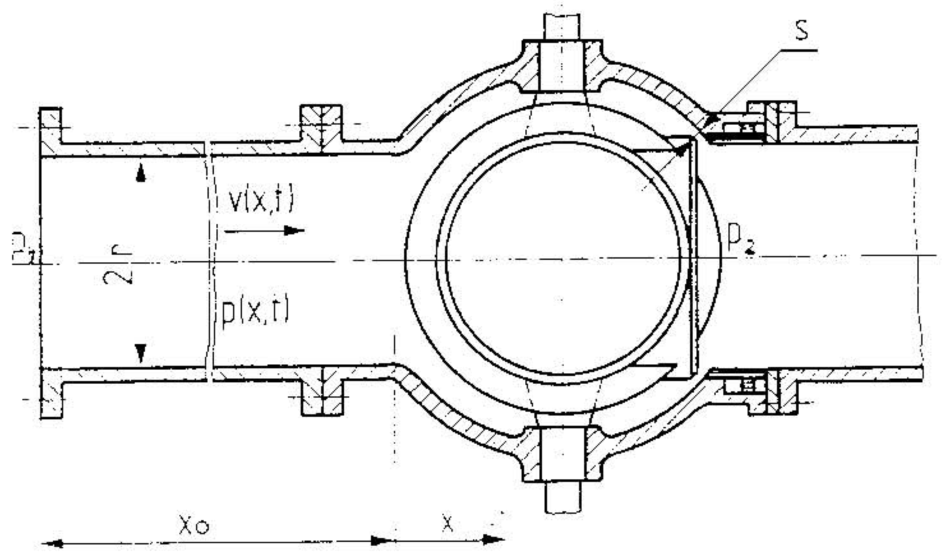

Fig. 9 Spherical valve

On the bases of the above mentioned fact a mathematical model and a solution to prevent the occurrence the vibrations was developed.

3.2 Nomenclature

c	sound speed in water	[m/s]
d	diameter of sealing ring	[m]
h_o	sealing ring of gap weight of unloaded spherical valve gate	[m]
k	stiffness coefficient of valve	[N/m]
m	valve mass	[kg]
p_1	pressure at the inlet of valve	[Pa]
p_2	pressure at the outlet of valve	[Pa]
Δp	pressure difference	[-]
Δp_o	static part of pressure difference	[Pa]
$\Delta_d p(t)$	dynamic part of pressure difference	[Pa]
S	cross section of valve	[m^2]
S_r	cross section of gap of the valve	[m^2]
t	time	[s]
$\Delta_1 t$	$\frac{2.(x_2 - x_1)}{c}$	[s]

$\Delta_2 t$	$\frac{2.(x_3 - x_2)}{c}$	[s]
u	displacement of a valve	[m]
$\dot{u}$	speed of spherical valve gate	[m/s]
$v(x_2,t)$	average velocity at x_2 in time t	[m/s]
$v_d(t)$	dynamic part of the average velocity at x_2 in time t	[m/s]
v_o	static part of an average velocity x_2	[m/s]
α	reflection factor at the end of tube	[-]
ρ	water density	[kgm^3]

3.3 Theoretical model

In Fig. 10, a model is shown for which a following system of equation is written:

$x_1 < x_2 < x_3$

$$\Delta p(t) = p_1\left(x_2, t - \frac{2(x_2 - x_1)}{c}\right) - p_2\left(x_2, t - \frac{2(x_3 - x_2)}{c}\right) - \rho c\left[2v(x_2,t) - v\left(x_2, t - \frac{2(x_2 - x_1)}{c}\right) - v\left(x_2, t - \frac{2(x_3 - x_2)}{c}\right)\right] \qquad 1.$$

$$m.\ddot{u} + k.u = S.\Delta p \qquad 2.$$

$$S_r = \pi.d.(h_o - u) \qquad 3.$$

$$v(x_2,t).S_o = S_r\sqrt{\frac{2\Delta p(t)}{\rho} + v^2(x_2,t)} + \vec{S}\dot{u} + 2\alpha\vec{S}_r.\left[2\vec{v}(x_2,t) - \vec{v}\left(x_2, t - \frac{2(x_2 - x_1)}{c}\right) - \vec{v}\left(x_2, t - \frac{2(x_3 - x_2)}{c}\right)\right] -$$
$$-2\vec{S}_r.\frac{S}{S_o}.\alpha^2.\left[\vec{v}\left(x_2, t - \frac{2(x_2 - x_1)}{c}\right) - \vec{v}\left(x_2, t - \frac{4(x_2 - x_1)}{c}\right) + \vec{v}\left(x_2, t - \frac{2(x_3 - x_2)}{c}\right) - \vec{v}\left(x_2, t - \frac{4(x_3 - x_2)}{c}\right)\right] \qquad 4.$$

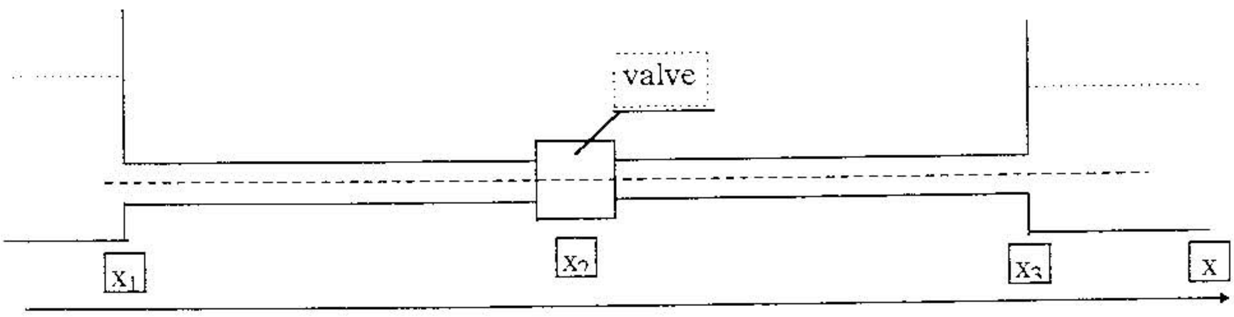

Fig. 10 Simulation model of a spherical valve

Equation 1 presents a pressure difference of the valve, equation 2 represents the elastic movement of the spherical valve gate, while equation 3 presents us a cross section of the gape in close position. Equation 4 presents a movement of the water close to valve. First part presents a Bernoulli equation, second an interation between water and valve, while other parts of the equation presents a reflection of water speed disturbance at the beginning and the end of the tube.

This system of equations can be solved only numerically. For discussion and practical use we linearise the equation 4 in the case that spherical valve gate is in closed position and sealing ring is open. Also we can say, that resonance frequency of the valve is higher than lowest frequency of the tube. In this case we can rewrite the above equations in the following form:

$$\Delta p(t) = \Delta p_o - \rho . c . [2v(t) - v(t - \Delta_1 t) - v(t - \Delta_2 t)] \quad 1.a$$

$$k . u = S . \Delta p \quad 2.a$$

$$S_r = \pi . d . (h_o - u) \quad 3.a$$

$$v(t) . S_o = S_r . \sqrt{\frac{2 . \Delta p}{\rho}} + S . \dot{u} \quad 4.a$$

where is:

$$\Delta_1 t = \frac{2(x_2 - x_1)}{c}$$

$$\Delta_2 t = \frac{2(x_3 - x_2)}{c}$$

Factor (h_o - u) in equation 3.a appears in a case that sealing of the valve is in the outlet part of the valve. If the sealing is in the inlet part, this factor is $h_o + u$.

Differentiating the equation 2.a, we get:

$$\dot{u} = \frac{S}{k} . \Delta\dot{p} \quad 5.$$

than the equation 4.a get the following form:

$$v(t) = \frac{\pi . d}{S_o} . \left(h_o - \frac{S}{k} . \Delta p \right) . \sqrt{\frac{2 . \Delta p}{\rho}} + \frac{S^2}{S_o k} . \Delta\dot{p} \quad 6.$$

In case that time dependent part of the pressure differences is small when comparing with the static part, $\sqrt{\Delta p}$ and $\sqrt{\Delta^3 p}$ can be developed into Taylor sum around Δp_o; only the first two parts are taken into consideration, so that:

$$\sqrt{\Delta p} = \sqrt{\Delta p_o}\cdot\left(1+\frac{\Delta_d \cdot p(t)}{2.\Delta p_o}\right)$$

$$\sqrt{\Delta p^3} = \sqrt{\Delta p_o^3}\cdot\left(1+\frac{3.\Delta_d \cdot p(t)}{2.\Delta p_o}\right)$$

With the assumptions above and in case S = S_0, the equation 6 can be rewritten as follows:

$$v(t) = \sqrt{\frac{2}{\rho}}.\pi.d.\left[\frac{h_o.\sqrt{\Delta p_o}}{S_o}.\left(1+\frac{\Delta_d \cdot p(t)}{2.\Delta p_o}\right) - \frac{\sqrt{\Delta p_o^3}}{k}.\left(1+\frac{3.\Delta_d \cdot p(t)}{2.\Delta p_o}\right)\right] + \frac{S.\Delta\dot{p}(t)}{k} \qquad 7.$$

As in stationary condition the dynamic part of the pressure is:

$$\Delta_d p(t) = -\rho.c.[2.v(t) - v(t-\Delta_1 t) - v(t-\Delta_2 t)], \qquad 8.$$

the equation 7 can be rewritten in the following form:

$$v(t) = v_o + v_d(t) = \sqrt{\frac{2}{\rho}}.\pi.d.\left(\frac{h_o.\sqrt{\Delta p_o}}{S_o} - \frac{\sqrt{\Delta p_o^3}}{k}\right) -$$

$$-\sqrt{\frac{\rho}{2}}\frac{\pi.d.c}{.\Delta p_o}.\left(\frac{h_o.\sqrt{\Delta p_o}}{S_o} - \frac{3.\sqrt{\Delta p_o^3}}{k}\right).[2.v_d(t) - v_d(t-\Delta_1 t) - v_d(t-\Delta_2 t)] - \qquad 9.$$

$$-\frac{\rho.c.S}{k}.[2.\dot{v}_d(t) - \dot{v}_d(t-\Delta_1 t) - \dot{v}_d(t-\Delta_2 t)]$$

From this equation, it follows:

$$v_o = \pi.d.\sqrt{\frac{2}{\rho}}.\left(\frac{h_o\sqrt{\Delta p_o}}{S_o} - \frac{\sqrt{\Delta p_o^3}}{k}\right) \qquad 10.$$

$$v_d(t) = -\sqrt{\frac{\rho}{2}}\frac{\pi.d.c}{\Delta p_o}.\left(\frac{h_o.\sqrt{\Delta p_o}}{S_o} - \frac{3.\sqrt{\Delta p_o^3}}{k}\right).[2.v_d(t) - v_d(t-\Delta_1 t) - v_d(t-\Delta_2 t)] - \qquad 11.$$

$$-\frac{\rho.c.S}{k}.[2.\dot{v}_d(t) - \dot{v}_d(t-\Delta_1 t) - \dot{v}_d(t-\Delta_2 t)]$$

Defining the following values:

$$A = \sqrt{\frac{\rho}{2}}.\frac{\pi.d.c}{\Delta p_o}.\left(\frac{h_o.\sqrt{\Delta p_o}}{S_o} - \frac{3.\sqrt{\Delta p_o^3}}{k}\right) \quad 12.$$

$$B = \frac{\rho.c.S}{k} \quad 13.$$

and substituting the equation 11, we get:

$$(1+2.A).v_d(t) = A.[v_d(t-\Delta_1 t)+v_d(t-\Delta_2 t)] - 2.B.\dot{v}_d(t) + B.[\dot{v}_d(t-\Delta_1 t)+\dot{v}_d(t-\Delta_2 t)] \quad 14.$$

As we are examining only stability, we will assume, that at the point x_2 there is some motion at stationary condition. Developing the motion in time intervals $2.\Delta_1 t$ and $2.\Delta_2 t$ into Fourier series and with assumption, that at the end of the tube there is

$$v_d(t-\Delta_1 t) = -v_d(t) \text{ in } v_d(t-\Delta_2 t) = -v_d(t)$$
$$v(t-2.\Delta t = v(t) \quad \text{in} \quad v(t-2.\Delta t) = v(t) \quad 15.$$

we finally get the following differential equation:

$$4.\ B.\ \dot{v}_d(t) = -(1 + A.4).\ v_d(t) \ . \quad 16.$$

Solution can be written as:

$$v_d(t) = C.e^{-\frac{1-4.A}{4.B}.t} \quad 17.$$

This equation for dynamic part of the pressure shows us, that stability of this pressure is reached at:

$$1+4.A > 0 \text{ and } A > -0{,}25 \quad 18.$$

As the valve should be stable in every position of sealing ring, also when $h_0=0$, where:

$$\boxed{A_{/h=0} = -\frac{3.\ \pi.\ d.\ c}{2.\ k}.\sqrt{2.\ \rho.\ \Delta p_o} > -0{,}25} \quad 19.$$

From this equation we can see, that the spherical valve is stable only under the following condition:

$$k > 6.\,\pi.\,d.\,c.\,\sqrt{2.\,\rho.\,\Delta p_o} \quad (20.)$$

This nonequality shows, that opening or closing of spherical valve is stable, if the valve is rigid enough to satisfy the above nonequation. In a case that the valve sealing surface is in the inlet part, then the two parts within the parenthesis in equation 3. and equation 12. are added, so that the condition in equation 19 is always fulfilled.
We can conclude that such spherical valves, at which by increasing of pressure also the gap increases, are stable. On the other hand, such spherical valves, at which the gap decreases because of pressure rise, are only stable if this decrease in excessive.

3.4 Discussion

We have made simulation for HPP Toro. We found out that the valve is rigid enough, therefore the condition I=0.048 << 0.25 is fulfilled. We expect, that there will not be any oscillations. All of the results will be tested in the hydro-power plant. The mathematical model is developed with assumption, that other parts of the valve are rigid.

4. CONCLUSION

In this article, an overall view of experimental and theoretical research in Litostroj is presented. Some typical results are shown. Special attention has been given to research and development. In the case study a mathematical model for control of the spherical valve is presented. Experimental verification of the model will be performed in the near future.

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