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POLISH ACADEMY OF SCIENCES

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

115

WYDAWNICTWO IMP PAN

GDAŃSK 2004

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ISSN 0079-3205

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 115, 2004, 113-124

ADAM ADAMKOWSKI, LESZEK KWAPISZ*

Strength analysis of penstock bifurcations in hydropower plants

The Szewalski Institute of the Fluid-Flow Machinery, Polish Academy of Sciences, 80-952 Gdańsk, ul J. Fiszera 14, PO Box 621, Poland

Abstract

The strength analysis of penstock bifurcation in hydropower plants is discussed in the present paper. The analysis consists of determining the maximal internal pressure, (e.g. during turbine load rejection), stress analysis of the pipeline shell for assumed loading and assumed or determined material properties. The investigation results can be helpful when determining the proper rate of the flow cut-off and recommending the strengthening precautions to be applied in places of maximum stress concentration in order to prevent future penstock ruptures.

Keywords: Water turbines; Hydraulic transients; Stress concentrations; Reinforcement of penstock bifurcation

1 Introduction

The stress magnitude in a pipeline bifurcation is usually 3-9 times greater than in regular pipeline shells [7-10]. For this reason special reinforcements are provided in order to decrease the stress concentration in crucial spots [7, 9].

The penstocks of hydropower plants built in the first half of the twentieth century are rarely equipped in such kind of reinforcement. The lack of reinforcement can result penstock failure, especially under sudden pressure rise conditions. The failure of the penstock in Lapino hydropower plant (Poland) can be a good example of the related strength problems [1]. The penstock rupture took place at the connection of the penstock with the turbine inlet pipe during turbine load rejection.

*Corresponding author. E-mail address: kwapi@imp.gda.pl

The method of strength analysis of a hydropower pipeline bifurcation is presented in this paper. The analysis consists of the following parts:

- determination of the maximal internal pressure, e.g. during turbine load rejection,
- determination of the mechanical properties of the pipeline shell material and rivet or weld junction,
- stress analysis of the pipeline shell for assumed loading and material properties.

The pressure loading is determined theoretically or experimentally. In the first case, a computer code developed in the Institute of Fluid-Flow Machinery of the Polish Academy of Sciences (IMP PAN) for prediction of water hammer in pipeline systems of hydraulic machines is used. The code has been validated using numerous experimental results.

Mechanical properties of the material are obtained by means of the standard tensile strength tests, provided that sufficient material samples can be taken from the penstock shell. In other cases, the mechanical properties are estimated by means of chemical tests and metallographic investigation. The complex stresses distribution in the analysed penstock shell is calculated by means of commercial codes (like ADINA, ABAQUS , NASTRAN) utilising the finite-element method (FEM). Additionally, mainly for the verification of the numerical results, strain gauge measurements are applied in the selected crucial points of bifurcations. Some results of investigations conducted in one of the Polish hydropower plants and stress analysis of the penstock shells are presented in this paper. Good coincidence of experimental and numerical results is confirmed.

It is proved that in some cases the maximal pressure during emergency transients states can be reduced by suitable change in the control turbine systems. It is also shown that the calculated stress concentration factors, confirmed by strain gage measurements, vary from 4 to 8 in the penstock bifurcations and that the high concentration stresses can be effectively reduced using collar reinforcements. The technical recommendations relevant to increasing safety of the hydropower plants are prepared based on the obtained results.

2 Determination of external loading condition

The pressure loading can be determined numerically using the HYDTRA (HYDraulic TRAnsients) computer code developed in the IMP PAN [2] for prediction of water hammer in the pipeline systems of hydraulic machines. The program is based on the method of characteristics, commonly applied for solving equations governing the unsteady liquid flow in the pipes. The program has been validated on several occasions using numerous experimental results [2, 3]. The discrepancy between calculation and experimental data is usually below few percent.

The pressure loading can be also determined experimentally. Simultaneously with the pressure measurements, the strain measurements in the selected places of the penstock shell are usually carried out. The strain measurements are described in the subsequent sections.

2.1 Examples of the measurement and calculation results

Figure 1 shows selected results of measurements conducted in one of Polish hydropower plants of 40 m rated head and equipped with three Francis turbines of 6 MW total output. The turbines are fed from the common steel penstock. The pressure inside the penstock and relevant stresses in selected points on penstock shell (see Fig. 9) were measured during load rejection of one of the turbines. As the measured stress magnitude was relatively high, numerical simulation of the simultaneous load rejection of all 3 turbines had to be applied instead of a field test (Fig. 3) (in order to avoid an overload of the penstock). The calculation was carried out under assumption that wicket gates closure law remained identical in each of three turbines.

The simultaneous load rejection of 3 turbines with the normal speed of wickets gates closure, causes an 80% pressure increase over the steady-state level in the penstock. The closure period of doubled duration could decrease the maximum pressure rise down to 36% of the steady-state pressure level as it is shown in Fig. 3. However, this measure would increase the runner overspeed from 24% to about 36% of the rated speed value. Fortunately, this overspeed is still in the allowed range which allows to recommend such a procedure.

As it is generally known slowing down the wicket gates movement process reduces the maximum pressure level in the penstock which is especially important in the old hydropower installations.

The reliability of the HYDTRA calculation was checked out by comparing the calculated and measured curves of pressure in the penstock (p_t) and runner speed (n) during a load rejection of one of the turbines (see Fig. 2).

3 Stress analysis in the penstock shell

3.1 Numerical calculation

Numerical calculation of stress distribution in the penstock shell was based on the FEM technique. A quadrilateral 8-node iso-parametric thin shell element was selected for this kind of calculation. No significant differences between results

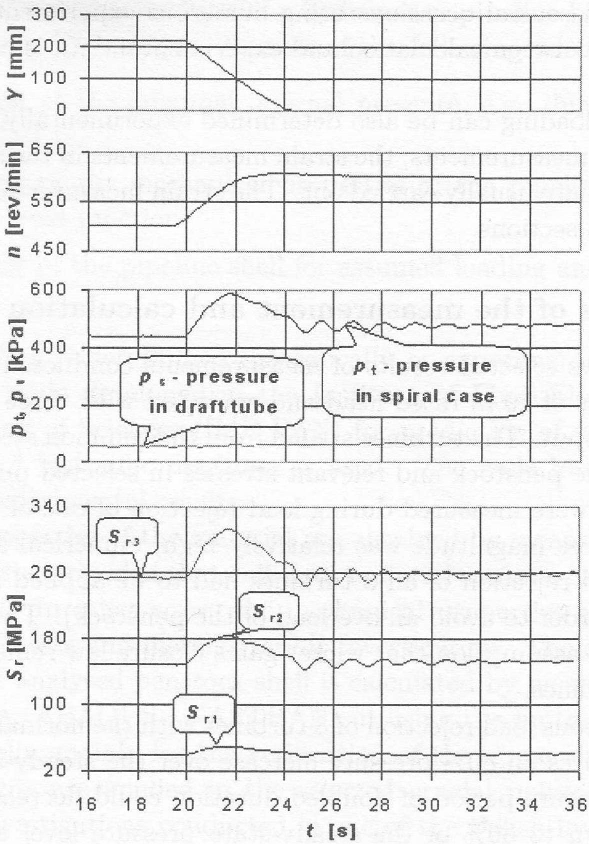

Figure 1. Quantities measured during load rejection of 1 turbine; Y – position of the servomotor piston, n – rotational speed, p_t – pressure in the spiral case, p_s – pressure in the draft tube, S_{r1}, S_{r2}, S_{r3} – equivalent stresses in the different shell places (see Fig. 4)

obtained by means of available ADINA, ABAQUS and NASTRAN commercial computer codes were stated. The calculation drew on measured geometry of the penstock segment and strength properties of the material found from the material tests. Components of the calculated stress field were substituted with the equivalent stresses according to the Huber/Henkey/von Mises (HMH) theory.

The obtained results show very unfavourable distribution of stresses in the analysed segment of the penstock shell. The connection of the inlet pipe with the main branch of the penstock is featured by significant concentration of stresses. The stress concentration coefficient, a ratio of the maximum stress value to the stress prevailing in the uniform conical segment, is about 8. This result coincides with the values often quoted in the literature [7].

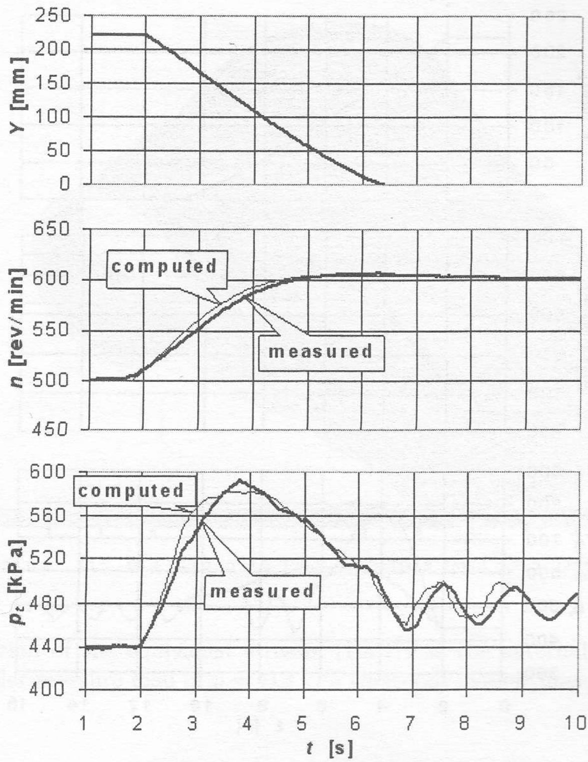

Figure 2. The comparison between the recorded and calculated curves of pressure in the penstock (p_t) and runner speed (n) during a load rejection.

The calculation was carried out for the following cases:

- penstock bifurcation with reinforcements of fin shape situated perpendicularly to the line of penstock – inlet pipe junction. (Figs. 4 and 6)
- penstock bifurcation without existing reinforcements, (Fig. 5)
- penstock bifurcation with recommended collar reinforcements and existing reinforcements. (Fig. 7,8)

The comparison between results of calculations carried out for existing fin shape reinforcements, Fig. 4, and for case without reinforcements, Fig. 5, shows that this kind of reinforcements is ineffective. The main concentration of stresses has been only slightly reduced (from 253 MPa to 240 MPa on external side of the shell) and, additionally, the fin shape reinforcement introduces stress concentration at the fin ends (Fig. 5) of about 180 MPa – fortunately, only on the external side of the shell.

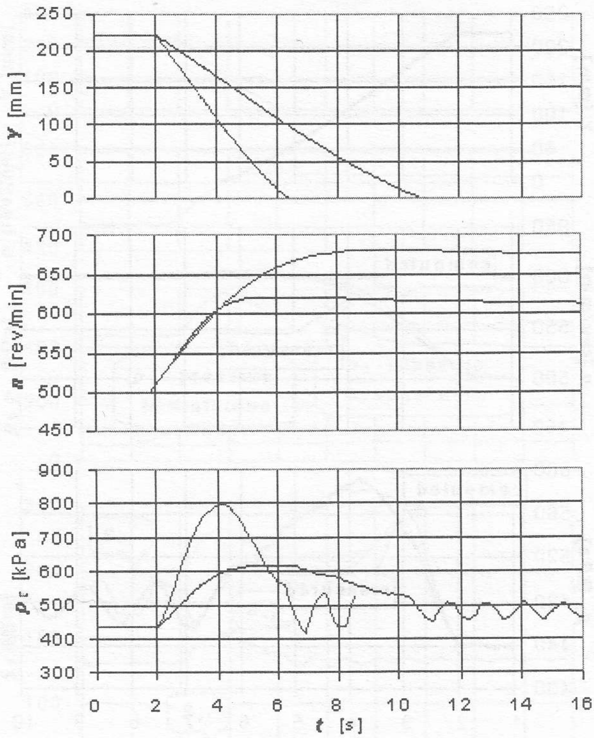

Figure 3. The computational results for the simultaneous power load rejection of all the turbines. The influence of slowing down the closure rate on pressure in the penstock (p_t) and runner rotational speed (n) is shown.

In case of 725 kPa external pressure loading (during simultaneous load rejection of all the turbines), the elastic-plastic behaviour of the shell material with its hardening characteristics has been assumed in order to make the numerical simulation more realistic. The uniform strain hardening characteristics has been based on assumption following from the material characteristics, that is yield stress $R_e = 235$ MPa, tensile strength $R_m = 375$ MPa and maximum strain $\varepsilon = 26\%$.

Results obtained by non-linear calculation with elastic-plastic behaviour of material show lower equivalent stresses, and more homogenous distribution of stresses in comparison to the linear prediction. However, the maximum obtained stress is still too high according to the safety requirements. For that reason the collar reinforcement shown in Figs. 7 and 8 has been recommended.

After a series of calculations the collar dimensions (width 28 mm, height 400 mm), have been specified so, as to reduce twice the maximal existing stresses under the steady-state rated loading and to reduce these stresses below the yield

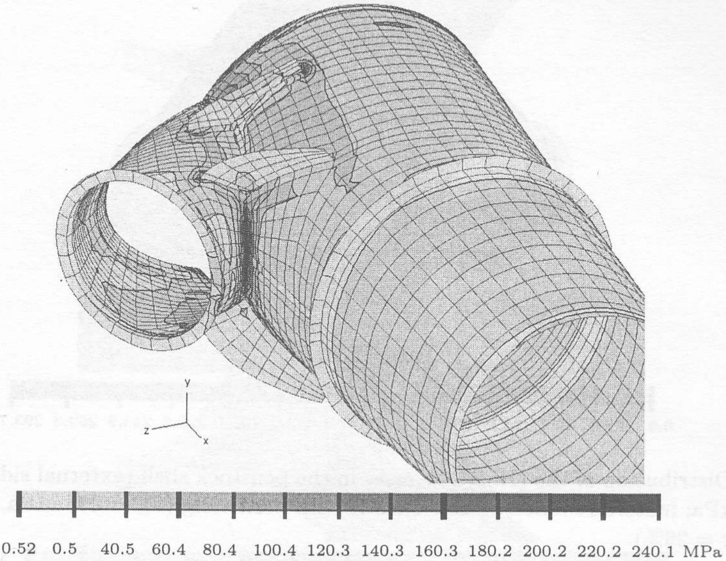

Figure 4. Distribution of the equivalent stresses (HMH) at the external side of the penstock shell under pressure load of $p = 415$ kPa (material model: linear elastic).

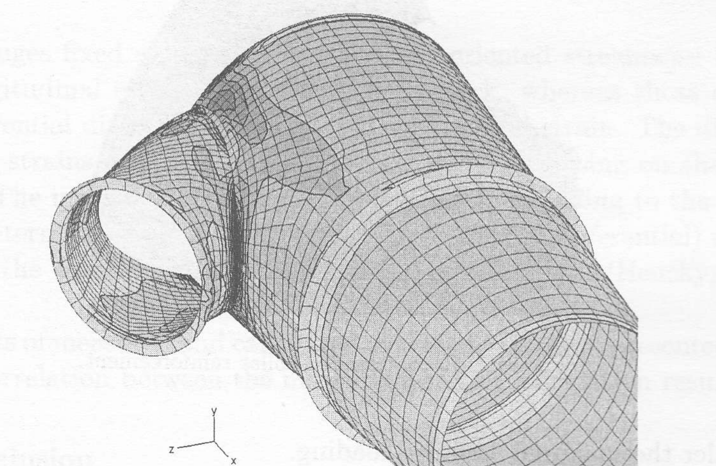

Figure 5. Distribution of the equivalent stresses (HMH) at the external side of the penstock shell under $p = 415$ kPa pressure load (material: linear elastic). Bifurcation without fin reinforcement.

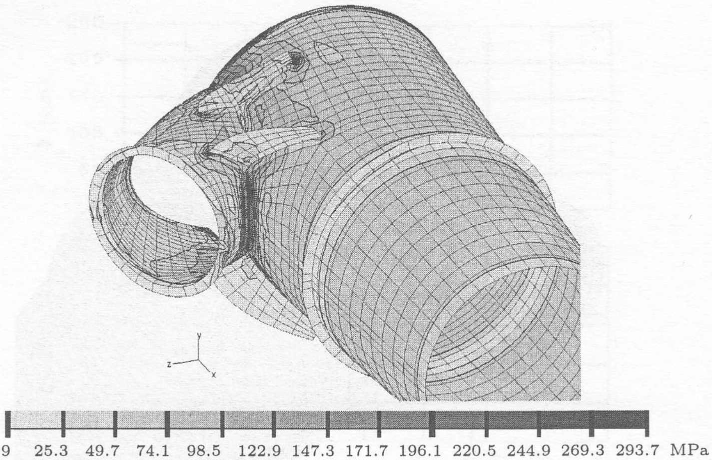

Figure 6. Distribution of the (HMH) stresses in the penstock shell (external side) under $p = 725$ kPa; material model: elastic- plastic with hardening ($R_e = 235$ MPa, $R_m = 375$ MPa, $\varepsilon = 26\%$).

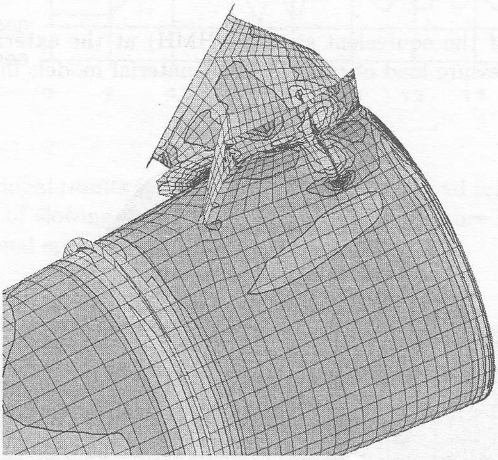

Figure 7. Recommended collar reinforcement.

point under the maximal assumed loading.

3.2 The stress measurements

The stresses in the shell of the penstock were evaluated by means of strain gauge measurements. In the few selected points on the shell (see Fig. 9) strain values were measured in two perpendicular directions, using fixed (glued) strain gauges.

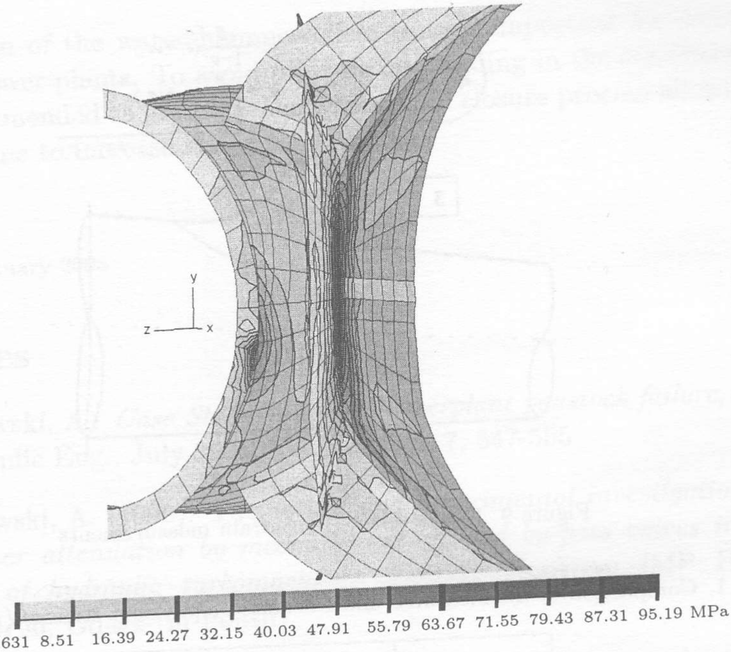

Figure 8. Concentration of the equivalent stresses (HMH) in penstock shell under $p = 415$ kPa pressure load (external penstock surface, material model: linear elastic). Recommended collar reinforcement width=28 mm high 400 mm.

Strain gauges fixed at point 1 (Fig. 9) were oriented streamwise in order to measure longitudinal strain of the conical penstock, whereas those oriented in the circumferential directions measured circumferential strain. *The directions of the principal strains in the other points were assumed basing on the results of calculation.* The measured strains were recalculated according to the Hooke law in order to determine the principal (longitudinal and circumferential) stress components and the equivalent stresses according to the Huber/Hencky/von Mises hypothesis.

The results of measured and calculated equivalent stresses, presented in Tab. 1, show good correlation between the measurement and calculation results.

3.2.1 *Conclusion*

- *The method of strength analysis of a hydropower pipeline bifurcation has been presented.* The analysis consists of:

  a. determination of the maximal internal pressure, e.g. during turbine load rejection,

  b. stress analysis of the pipeline shell for assumed loading and assumed

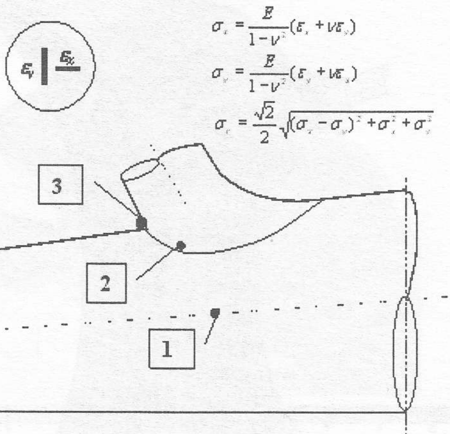

Figure 9. Places selected for strain measurements

Table 1. Comparison of the measured and calculated equivalent stresses (ADINA code)

	Equivalent stresses under the steady-state rated loading of 415 kPa		
	S_{1r}	S_{2r}	S_{3r}
	MPa	MPa	MPa
Experiment	39.6	149.1	247.5
Calculation	37.5	141.4	240.0

or determined material and rivet or weld junction properties. Good coincidence of experimental and numerical results has been confirmed.

- The presented results show highly non-uniform stress distribution in the analysed penstock bifurcation. The maximum of the reduced stresses is typically located at the connection of the penstock with the turbine inlet pipe. It has been also shown that the calculated stress concentration factors in the penstock bifurcations do coincide with those established basing on the strain gauge measurement (about 8 points) and that the high concentration stresses can be effectively reduced using collar reinforcements. This should be taken into account especially in case of old hydropower plants, lacking proper bifurcation reinforcements and subjected to quality decrease of the structural material applied and the weld or rivet junctions.

- Besides the shell reinforcements in the bifurcation area of the penstock, the reduction of the water hammer effect is very important for safety of the hydropower plants. To avoid pressure overloading in the considered case it is recommended to graduate the wicket gate closure process allowing in the same time to increase the runner speed.

Received 20 January 2004

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