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Performance and safety of hydraulic turbines

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Performance and safety of hydraulic turbines

H Brekke[1]

[1]Department of Energy and Process Engineering. Norwegian University of Science and Technology, Priv. address: Litavegen 8, Trondheim, NO 7032, Norway, Phone: +47 72559915, Fax: +47 73593854,

E-mail: hermod.brekke@ntnu.no

Abstract. The first part of the paper contains the choice of small turbines for run of the river power plants. Then a discussion is given on the optimization of the performance of different types of large turbines. Finally a discussion on the safety and necessary maintenance of turbines is given with special attention to bolt connections.

1. The best economic choice of turbines

For large hydropower projects there will normally be an evaluation of efficiency versus price. Further the efficiency will be measured either of model turbines or at site of the full scale machines for high heads where the measuring accuracy is good. However, for small hydro the cheapest buy is often the case for the turbines, but on the contrary the best economic choice of diameter or cost of penstock is made versus the hydraulic head loss. The reason for this is often the lack of knowledge by local small consultants and the cost of efficiency measurement. In addition there are many small manufacturers making cheap turbines with low efficiency, but able to obtain the guaranteed maximum power. Then the customer is satisfied as long as there are no problems with brake down of turbine parts caused by fatigue problems with broken buckets or turbine blades or cavitation damage.

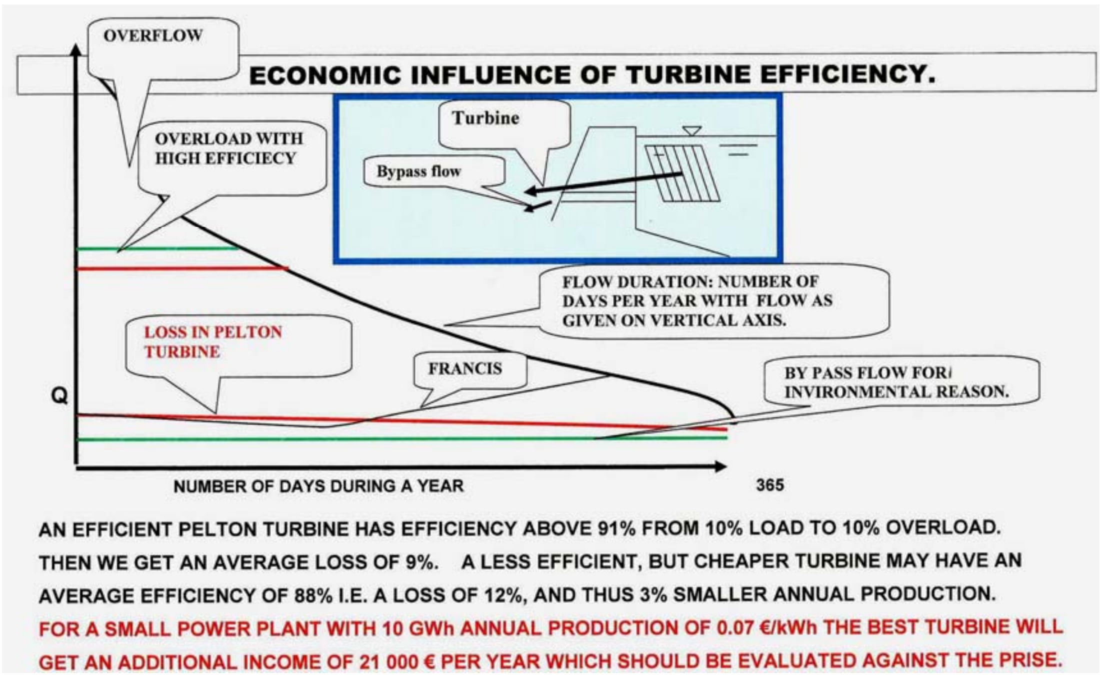

Fig. 1 Analysis of the flow utilization in a run of the river plant by multi nozzle Pelton turbines with a difference in efficiency of 3%.One Francis turbine would produce around 15% less.

However, when operating at part load in run of the river plants using all available water in the dry part of the year, the loss is depending on the turbine efficiency at part load which goes in favour of Pelton turbines. In Fig. 1

the flow duration curve is shown with illustration of the efficiency of a 6 jet Pelton turbine and a Francis turbine. Further by comparing the efficiency of two Pelton turbines with 3% difference in efficiency the difference in annual income would be € 21 000.

2. Turbine design for optimal performance

The dynamic behaviour at part load has been a major problem for low head and medium head Francis turbines. The main reason for this has been inter blade separation and unstable swirl flow in the draft tube. A theoretical study of the flow in the runner blade channels regarding the cross flow from hub to band, has proven that it is possible to reduce this unfavourable flow by a negative blade lean at the inlet and thus balancing the pressure all the way towards the outlet of the blades

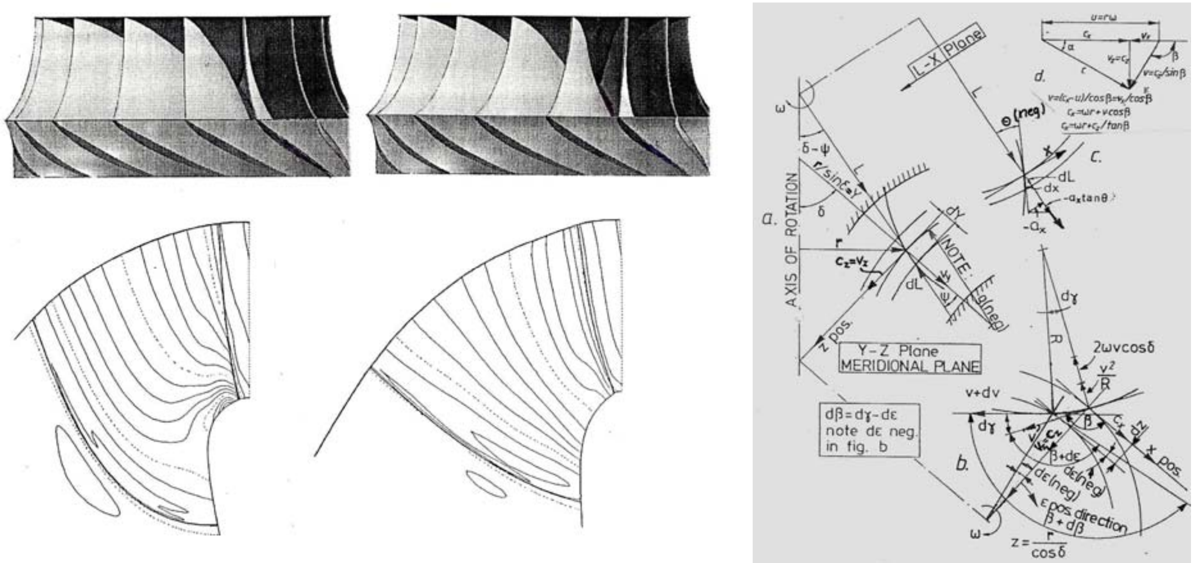

Fig. 2 Pressure distribution of a traditional runner (left), X-BLADE runner (middle) and definition of geometry for a potential flow analysis for creation of the blade shape before the final CFD analysis (right). Note the difference in pressure towards the band.

The blade lean angle for a runner blade is defined by the angle between the blade and a meridian plane measured perpendicular to the stream surface, denoted by Θ as illustrated in fig.2 (right) together with the other parameters used in the equations (1), (2) and (3). The dimensionless pressure gradient $d\underline{h}/dy=d(h/H_n)/dy$ perpendicular to the stream surfaces can be expressed by equation (1) along the blade as a function of the dimensionless meridian velocity component $\underline{c}_z = \underline{v}_z = c_z/(2gH_n)^{0.5}$ and the angular speed $\underline{\omega} = \omega/(2gH_n)^{0.5}$ (m^{-1}). The directions x, y and z are defined in fig.2 (right). The equation for the pressure gradient ($d\underline{h}/dy$) eq. (1) is valid along the blade surface from the runner crown to band and is a function of $\underline{c}_z$ and the blade lean angle Θ.

$$dh/dy=2[[(1/R-\cos\delta\cos^3\beta/r)(c_z^2/\sin^3\beta) - c_z(\delta c_z/\delta z)/\tan\beta + 2\omega\cos\delta c_z]\tan\Theta + (\sin\delta/(r\tan^2\beta)+1/\rho)c_z^2+(2\omega\sin\delta/\tan\beta)c_z+\omega^2 r\sin\delta] \quad (1)$$

For a pressure balanced dh/dy = 0 when Θ is adjusted correctly illustrated by the X-blade runner in fig. 2. Equation (1) is based on the equilibrium of forces and is valid for a runner with infinite number of blades i.e. potential flow. In addition to equation (1) the ROTALPY equation (2), must be established together with the equation of continuity (3). The presented theory is two dimensional (infinite number of blades), but is still useful in order to obtain a physical understanding of the quantitative influence of the blade lean during the basic design of a Francis runner followed by a full 3D viscous analysis. This simplified study made it quite clear that a negative blade lean on the inlet of the runner was necessary to reduce the cross flow on the pressure side of the blade and the X-BLADE runner was a result of this study in 1996.

$$\underline{h} = \underline{\omega}^2 r^2 - \underline{c}_z^2/\sin^2\beta + (1 - 2\underline{u}_1 c_{u1}) - J \quad \text{(J is an estimated loss along a stream line.)} \quad (2)$$

IOP Conf. Series: Earth and Environmental Science **12** (2010) 012061 doi:10.1088/1755-1315/12/1/012061

In addition to eq. (1) and eq.(2) the equation of continuity between each stream surfaces is used to find the meridian velocity c_m.

$$\underline{c}_z = (Q/N)/(2\pi\ r\ b\ \Phi) \tag{3}$$

(b= distance between the estimated and adjusted stream surfaces, N= number of blades, and Φ = influence from blade thickness.)

In Fig.3 to the right is shown the cross section of a traditionally designed runner installed in the turbine at the power plant Frøystul in Norway and to the left is shown a proposal for installing an improved runner of the X-shape. However, the first runner of the X-blade design was installed in the turbine at Bratsberg Power Plant in Norway proving a very stable operation over the whole range of operation.

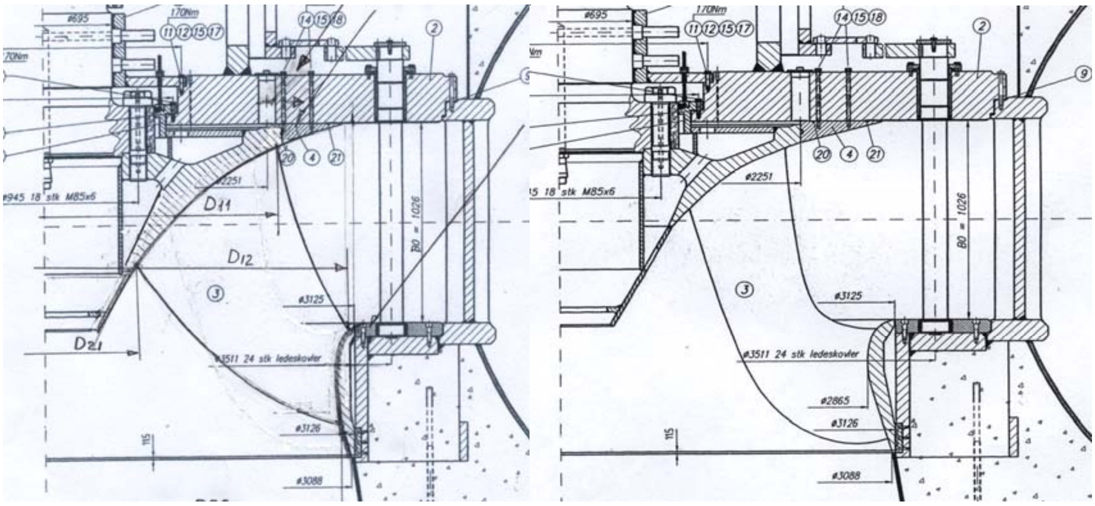

Fig. 3 The original runner for Frøystul Power Plant (right) and a proposal for an X-BLADE runner installed in the turbine (left).

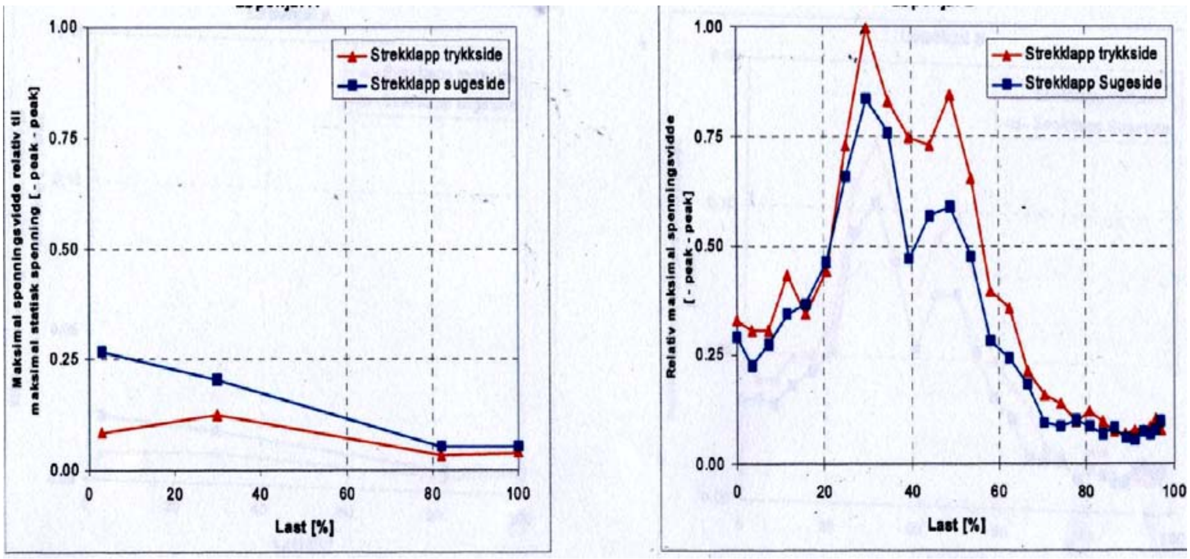

Fig. 4 Relative stress amplitudes measured on the runner blade outlets of the runner for Frøystul power plant (right) compared with a similar measurement on the X-BLADE runner at Bratsberg power plant (left). Both power plants are located in Norway.

The model proved a guaranteed limit of draft tube surges and the efficiency was close to 95%.after fine tuning. In Fig.4 (left) the measured relative stress amplitudes on the runner blade outlets on the first X-blade runner in operation in the field at Bratsberg Power Plant is shown and compared to the stress amplitudes measured at Frøystul Power Plant shown to the right. It should be noted that the specific speed of the turbine for Bratsberg was lower than for Frøystul, but an improvement would occur if an X-blade runner had been installed at Frøystul. The owner of Frøystul Power Plant received last year a proposal from the author to ask for a bid of a new runner because the original runner can not operate at part load without blade cracking problems.

For high head runners the introduction of splitter blades made it possible to operate high head turbines over the whole range of power without restrictions if the splitter blades were designed properly. This know how was invented just before World War II and was improved in Norway during the boom of hydropower installations from 7 000 MW around 1955 to 27 000 MW in 1990. The reason for this was the electric power needed for the electro chemical- and electro furnace industry with requirements of fast changes from no load to full load with operation often on isolated grid. In addition the power production was consumed for heating and light in cities. Also for splitter blade runners the blade lean angle and shape of hub and band is important in the same way as for the X-blade runners. A detailed description will not be possible in this condensed paper. However, for heads around 350 m efficiency exceeding 95% has been obtained with a smooth part load operation.

Fig. 5 A splitter blade runner during fabrication around 40 years ago.

For Pelton turbines an improvement has been made during the last 20 years and the efficiency to day has exceeded 92%. However, it should be noticed that the efficiency of model turbines that are normally operating at 100 m, is higher than the efficiency of the larger prototypes normally operating at heads above 700 m. The general change in bucket design towards modern turbines is illustrated in Fig. 6**.** However, also nozzles and inflow conditions have been optimized for large turbines, while small hydro turbines often suffer from general errors in the inflow geometry such as bends in two planes and bad design of nozzles.

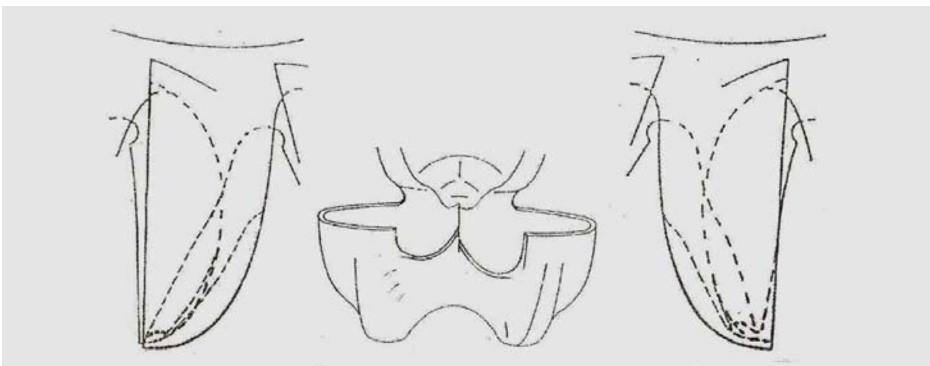

Fig. 6 Improvement of Pelton buckets by a higher splitter and adjustment of jet entrance portion and outlet angles.

3. Safety and life time

3.1 Rotating parts

For high head Francis turbines and Reversible Pump Turbines for high heads relatively strong pressure pulsations occur in the blade channels created by the blades passing the wakes from the guide vanes. The frequency will be the rotational speed per sec multiplied by the number of guide vanes. However, on the outside of the runner the frequency will be the number of runner blades multiplied by the rotational speed per sec. This has been proven by pressure measurements between lower cover and runner band at high head plants in Norway, [2]**.** In one case described in [2]**,** also a severe noise problem occurred because the shock wave from the blade passing through the wake reached the next guide vane wake simultaneously with the blade in front i.e. $\omega R=a(S+\Delta S)$. (See illustration to the left in Fig. 7)

The conclusion will then be as proven by measured stresses on the runner blades in several cases: The natural frequency will probably not have any influence on fatigue problems of low specific speed Francis runners. However, water hammer reflections in the blade channels should be studied. In Fig. 7 an illustration is given of the measured stress amplitudes on the runner blades compared to the calculated natural frequencies marked with arrows.

The main reason for the blade cracking of high head turbines is normally a non favorable blade shape which is causing too high stresses. Further or too large weld defects and/or high residual stresses from fabrication could be another reason, [3]**.** However, for low head plants with high specific speed turbines installed, the main reason for blade cracking are pressure pulsations in the draft tube at part load with frequencies that may find resonance with the natural frequency of the runner. Measured stress amplitudes on the blade outlets on the original high specific speed runner in the turbine at Frøystul, is shown in Fig.4 to the right. Note that the measuring results show no stress peaks with frequency of the blade passing frequency because of the long distance between guide vanes and blade inlets.

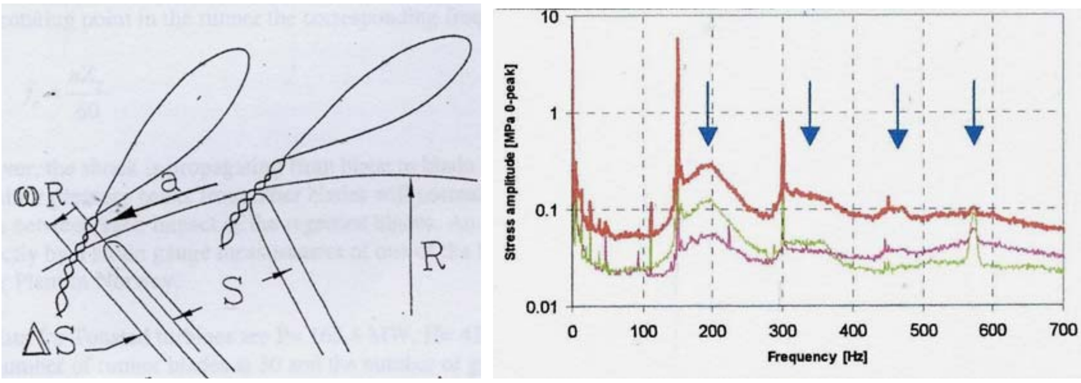

Fig. 7 Illustration of blade passing (left) and results from stress measurements on runner blade outlets at Tonstad Power Plant (NORCONSULT). (Speed n=375 rpm and number of guide vanes 24). Blue arrows indicate the calculation of possible natural frequencies of the runner.

The frequency of the draft tube pressure pulsations are normally around 1/3 of the runner speed. For comparison to the blade passing frequency of high head turbine we will with 24 guide vanes and n=375 rpm, get a frequency of 148.8 Hz and 10^8 cycles will be reached in 7.8 days or around 1.1 week while a low head runner with the same speed needs 1.1*24*3=79.2 weeks to reach 10^8 cycles.

From the Fracture Mechanic Theory we learn that the threshold value for a material defect exposed to high cycle stress amplitudes, where no crack propagation occurs is 10^8 cycles. Because of this a high head Francis runner should be inspected just before or at the latest at the time when 10^8 cycles is passing in order to be able to remove or repair possible growing cracks. This is because a total fracture where pieces will fall out of the blades at the runner outlet will occur during the following 10^8 cycles or 7.8 days for a high head runner.

The good news are that after 3 or 4 inspections of a runner operating without any crack growth, the runner will theoretically have an infinite life time. However, for a high specific speed runner operating at low head, the inspection period will be 79.2 weeks or around 1.5 year and the runner cannot be declared to have an infinite life until it have been in operation for 6 years mainly at part load without any repair**.**

The threshold value for no crack growth on materials exposed to cyclic loading yields according to [4]:

IOP Conf. Series: Earth and Environmental Science **12** (2010) 012061 doi:10.1088/1755-1315/12/1/012061

$$\Delta K_0 = \Delta\sigma\sqrt{a}\,\varphi \qquad (4)$$

where $\Delta\sigma$ = stress width or peak to peak amplitude, a = crack depth and φ = 1.26 which is a geometry factor for a semi elliptic crack with depth 0.5 times the length for a surface crack and depth/length ratio = 2a/2a=1/1 of a submerged crack, [3, 4].

According to material research made by Sulzer in Switzerland in 1983 and later confirmed by other tests referred to in [4,5,6], ΔK_0=72 (N $mm^{-3/2}$) for a value of $R=\sigma_{min}/\sigma_{max}=0.5$, where no crack propagation occurs. For practical reasons that leads to an acceptable crack size of around 1.3 mm depth and 2.6 mm length for stress amplitudes of around 50 N/mm^2.

For Pelton turbines the fatigue problems are caused by the jet impulses on the buckets and in some cases stress amplitudes from additional natural frequency of runners with weak disks which may give additional superimposed stress amplitudes and fatigue problems. The frequency for the pulsating hydraulic load will be the runner speed multiplied by the number of nozzles.

For casted runners and also to some extent for “micro cast” runners and forged runners or runner welded by forged parts, material defects will always be present.

In Fig.8 is shown an example of a broken Pelton bucket caused by a small, but incorrect weld repair without heat treatment of a small material defect from casting (MnS) during repair of an old runner. The total fracture occurred around one year after installation of the complete overhauled runner after many years in operation.

Fig. 8 Broken Pelton bucket caused by a not successful not heat treated small weld repair. Detail to the left.

3.2 Pressure loaded turbine parts

Based on Fracture Mechanic theory, the static working stresses in pressurized parts in turbines must be limited to a value which gives a critical crack size large enough to penetrate the plate i.e. to fulfil the criterion **“leakage before rupture”**. The parameters in question are the stress level and the Crack Tip Opening Displacement (CTOD) which is a material constant available from the steel mills. From the CTOD value the critical size of a crack where the unstable crack propagation occurs, can be calculated. An example of such unstable crack propagation is the penstock rupture that occurred in ultra high head power plant Biedron at Grand Dixence in Switzerland around New Year 2000. Because of high tensile stress in the plates and consequently high working stresses, the critical crack was small in this case and water hammer waves superimposing the static pressure, caused the crack growth and the ultimate rupture. Further an acceptable material defect in a new turbine must be so small that it does not grow to a size which leads to an unstable fracture within 10 000 - 50 000 loading cycles i.e. the number of start stops or depressurizing pressurizing cycles of the turbine. The stress level must also be limited so a not acceptable defect in a turbine is big enough so it will not be overlooked during inspection by Ultrasonic-, X-ray, Magnaflux or Penetrant liquid methods.

All stress concentration locations must also be documented by FEM analysis and the stress level must be taken into consideration when setting the criterion for acceptable material defects. Special attention should be paid to locations where cracks of any size will not give any leakage. Such locations are the stay vanes in the spiral casing of Francis Turbines, Reversible Pump Turbines or Kaplan Turbines and reinforcement ribs in bifurcations of Pelton turbines. In Fig. 9 to the right a detected crack in a stay vane of a spiral casing is shown and to the right the result from a Finite Element Method (FEM) analysis is shown where the stress concentrations can be found. Any growing cracks must also be repaired before it is too close to the critical size.

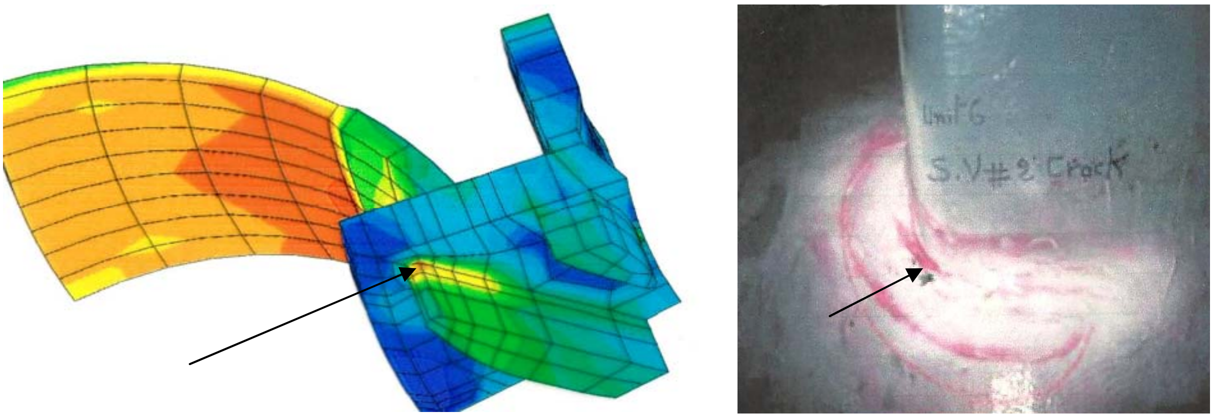

Fig. 9 Crack in stay vanes that will not give “leakage before rupture”. Left: FEM analysis showing the stress concentrations. Right: Crack detection during annual control indicating a growing crack starting from a weld defect.

3.3 Bolt connections - maintenance

Special attention should be paid to the bolt connection for joints exposed to cyclic loading. A very well known serious accident occurred at turbine number 2 in Shushenskaya power plant in Russia last year. This turbine had a very rough operation at part load with high pressure pulsations leading to blade cracking and cavitation damage on the runner. Maintenance of the bolts in the head cover flange had also been neglected, resulting in a total catastrophe ripping the whole head cover off the turbine, throwing the head cover and rotating parts of turbine and generator through the stator and drowning the whole power house.

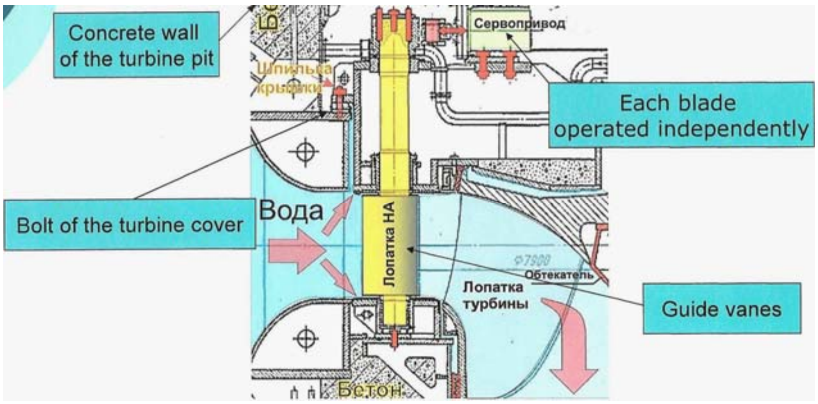

Fig. 10 Cross section of the 640 MW Francis turbine at Shuskenskaya Power Plant in Russia.

In Fig.10 a cross section of the turbine is shown and in Fig.11 is presented the Russian report on the status of the head cover bolts before the accident.

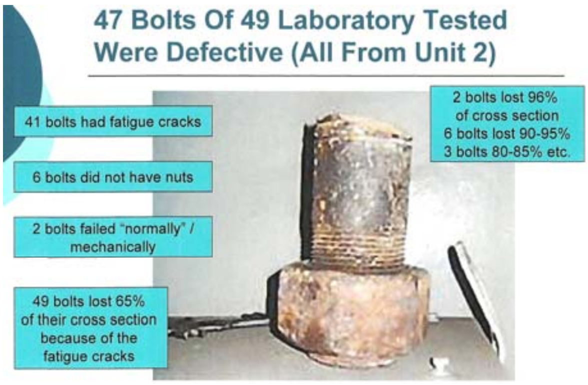

Fig. 11 The Russian report from the examination of the head cover bolts at Shusenskaya power plant.

The report from the catastrophe also described the rough operation of the turbine and that the planed maintenance of turbine no.2 had been postponed, At the time of the accident the recoded vibrations of the turbine had exceed the allowable value so the turbine should have been stopped. However, the automatic stopping system was not in operation according at the time of the accident according to the Russian report.

However, other examples of bolt failures have been observed caused by neglected maintenance. In Fig. 12 to the left the flange on the runner chamber of a large Kaplan turbine in Norway is shown.

The flange connection of the runner chamber of a Kaplan turbine is exposed to a pulsating hydraulic pressure caused by the difference on pressure side and suction side of the rotating blades. (See Fig. 12.)

In this case corroded bolts were broken at the last thread into the nuts or at the threads in the embedded flange in a similar way as for the head cover bolts at Shusenskaya power plant in Russia.

In the this case the hydraulic pressure was low and no great catastrophe would occur, but the flange had opened up to 2 mm on one side and the runner blades had touched the chamber wall. It should also be noted that the generator stator was out of roundness with eccentricity above normal standard, pulling the rotor and shaft to one side causing eccentricity of the turbine shaft and hub.

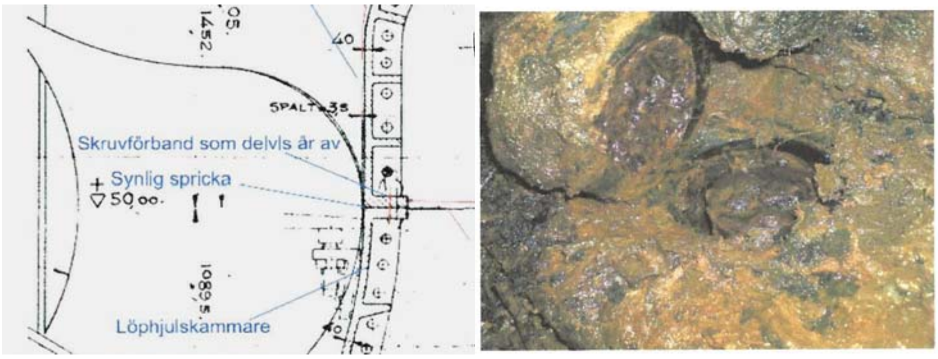

Fig. 12 Runner chamber and one of the broken bolts in the runner chamber flanch at Kykkelsrud power plant in Norway.

The last example of bolt fractures caused by dynamic load deals with relatively small stainless bolts where rust does not play any important role as in the two other examples described in this paper.

The bolt connection illustrated in Fig. 13 deals with a bolted plate covering the balancing weight for the runner of a vertical reversible pump turbine at Hyatt power plant in California.

The reason for the dynamic load in this case was the drainage pipe leading the leakage water from the upper labyrinth seal to the top of the draft tube of the pump turbine. Then a regarded point of the cover plate passed a low pressure zone each time the point passed through the drainage pipe inlet.

In addition the drainage pipe was connected to the top of the draft tube where relatively large pressure pulsations with frequency around 1/3 of the rotating speed, occurred at certain points of operation. These draft tube pressure pulsations occur normally at part load in turbine mode, but pressure pulsations may also occur in pumping mode at extreme variations in head. In addition water hammer pressure pulsations were created in the draft tube during start and stop of the turbine. After approximately 5 years in operation all stainless bolts were broken in this case and the cover plate loosened caused by the cyclic load.

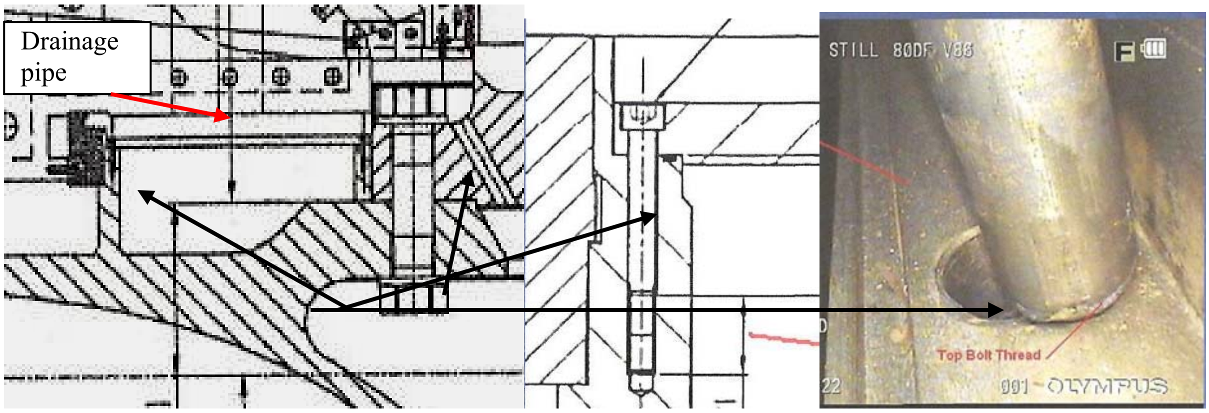

Fig. 13. Cross section of the runner at one of the three Reversible Pump Turbines installed in Hyatt Power Plant (left) and a close up drawing of one of the bolts (middle) and a close up photo of a broken bolt (right)

The dimensions and quality of the bolts were 1/2" diameter threads with quality 18/8 CrNi steel according to ASTMA197 Grade B85 with (ultimate/yield) stress 95/50 ksi = 655/345 MPa. Note the European standard ISO 3506 (8.75) has strength (800/600) MPa. Because of this the tightening of so small bolts as 1/2" might have caused stresses exceeding yield stress in the last thread into the treaded hole. Then even a small superimposed stress amplitudes might create fatigue in the grain structure even without any material defect if the stress is close to the ultimate stress for the steel in question.

Of interest is also that the time to reach 10^8 number of pressure cycles with frequency equal to the speed of the turbine with n=189 RPM will be $10^8/(189*60*24)=367.4$ days or one year if the turbine is operating continuously. A longer initiating time is expected for a possible microstructure fracture in materials without initiated defects, but with very high static stresses superimposed. Then $5*10^8$ cycles or 5 years before rupture seems to be reasonable in this case. A study of material fatigue without initiated start cracks has been discussed thoroughly in [5].

In spite of the large difference in size of the bolts and material quality used in the three described cases, it is an obvious similarity that the fatigue fracture starts in the last thread towards the nut or threaded hole, and that a total fracture occurs after a certain number of cycles. Then inspection and maintenance is necessary for all important bolted joints to avoid fatal ruptures.

It should also be mentioned that none of the bolts in discussion in this paper had reduced diameter between the threaded parts or between the threaded part and the bolt head for the bolts in shown in fig 13.

High stress bolts exposed to fatigue should, in the opinion of the author, have reduced diameter between the threaded parts in order to reduce the stresses on the last thread. Such bolts are always used in car engines and gas turbines etc. and also for hydraulic machinery such bolts should be considered to be used, even if the cost will be increased.

4. Conclusion

Efficiency, reliability, and maintenance of hydraulic machinery are most important for the economy and safety of hydropower. However, basic knowledge of design and maintenance is required both by manufacturers, consultants and owner of hydropower plants to select the best equipment for a safe operation with highest possible production.

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