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A Review on Oscillatory Problems in Francis Turbines

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1. Introduction

There is a worldwide requirement of hydropower turbines for peak load operation at increasing heads and for low head turbines operating with large head variations.

The reason for this requirement is the increasing demand for clean peak power production caused by the increasing number of wind mills and to reduce the peak power production from thermal power plants that is increasing the polluting production of CO₂.

In addition to this the weight of the turbines versus power has decreased by changing from cast steel to welded structures of high tensile strength steel plates. Further the flow velocity in the turbines has increased and thus increasing the danger of pressure pulsations, vibrations and noise in the power house. Fatigue problems caused by material defects also require a thoroughly inspection and regular maintenance of the machines.

This chapter is aimed towards three main targets:

- To avoid not acceptable noise and fatigue problems from high frequency pressure pulsations caused by blade passing through the guide vane wakes and vortex shedding from the trailing edges of vanes.
- To improve the design and analysis of steady oscillatory problems of hydro turbines during operation at off best point load.
- Present a reliable analysis of surge problems that may cause power oscillations in the grid.

2. Historical development of turbines after World War II

2.1 More powerful turbines and weight reduction

The traditional way of producing high head turbines was using steel castings in the pressure carrying part like spiral casings, head cover and bottom cover. Then the turbines became very heavy requiring a labor consuming production in foundries and workshops. Because of relatively small units this design was used up to around 1960 for the pressure carrying parts.

The reason for the possibility of a weight reduction to around 1/3 for the spiral casings, as illustrated in fig. 1, was the improved welding technology and the introduction of high tensile stress plates with yield point of 460 MPa.

The turbines illustrated in fig. 2 gives a clear indication of the weight reduction of high head Francis turbines. The main reason for the weight reduction was saved production cost caused by less man hours to produce the turbines. In addition the transport costs were reduced.

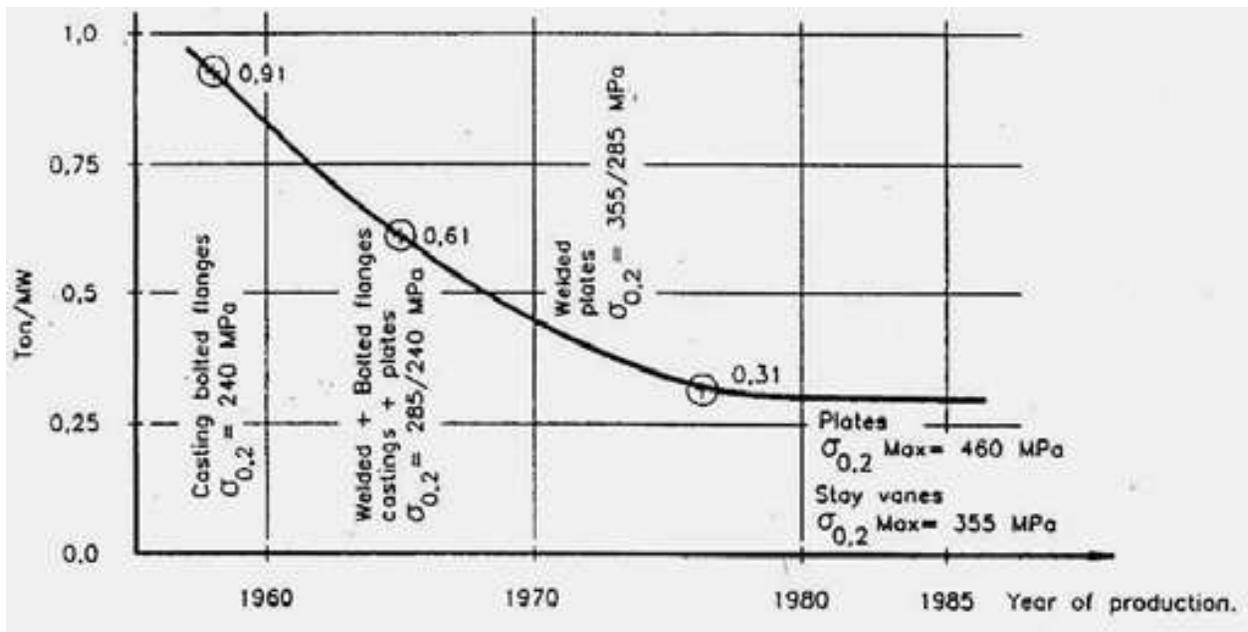


Fig. 1. Weight reduction of high head Francis turbines produced by KVAERNER from 1957 to 1985.

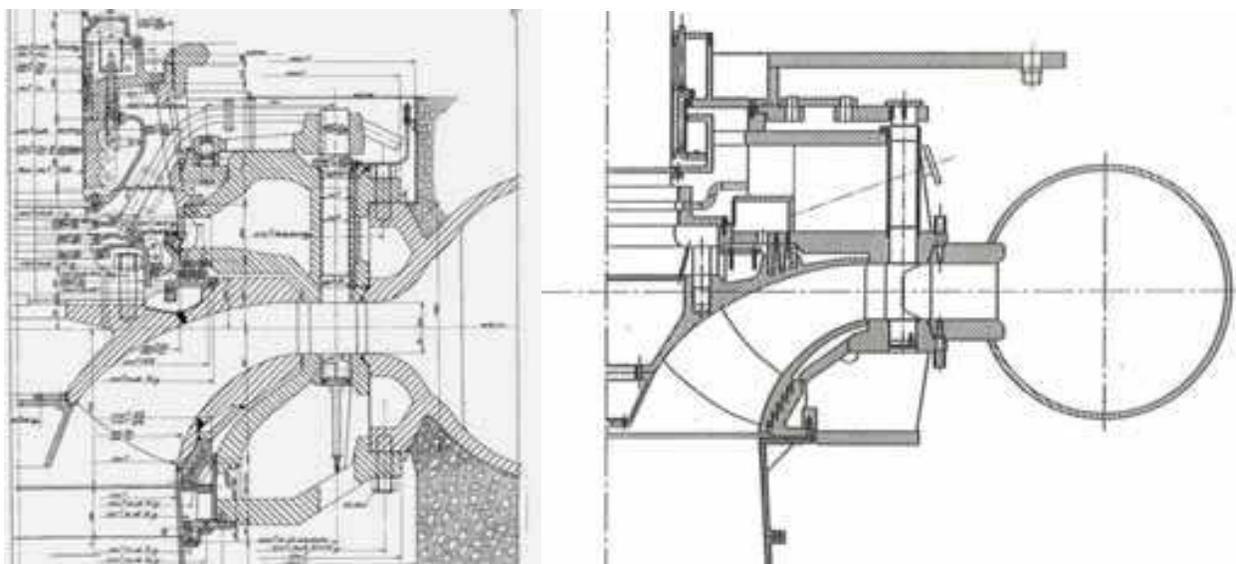


Fig. 2. A traditional high turbine for Tokke power plant made in 1959 (left) and a modern welded turbine of plate steel with yield point 460 MPa.

The draw back with the lighter turbines was that a lighter turbine gives less damping of pressure pulsations and the higher stresses required a more strict production control to reduce the size of material defects and geometrical inaccuracy. Also for Pelton turbines the design was changed from heavy casted bends and bifurcations to welded parts of rolled plates. In addition multi nozzle vertical units were introduced, mainly because the power for each unit increased from around 30 – 50 MW to 100- 315 MW in the period from 1957 to 1985. This development gave a weight reduction per kW similar to the high head Francis turbines.

However, this chapter is aiming at oscillatory problems in Francis turbines and a further description of the general trend in turbine design will not be included.

Another challenge for operation of the hydro turbines is the requirement for peak load operation from no load to overload in some cases. Further the growing demand for peak load support from the growing number of wind mills and the need for support to the coal fired power plants in order to run these power plants on planned more or less constant load setting in order to reduce the pollution from CO₂ production.

The next section in this chapter of the book will be aimed at dynamic high frequency problems in runners and adjacent components affected by high frequency pressure pulsations and draft tube surge problems at off best point operation.

Finally problems related to water hammer and pressure surges from draft tube voids acting as surge tanks will be described with examples of power oscillations with possible resonance from the electric system.

3. Dynamic problems of Francis turbines in the high frequency domain

3.1 The blade passing frequency

The dominating sound from a high head Francis turbine without cavitation problems is the blade passing frequency.

This sound is caused by the rotating pressure waves travelling on the outside of the runner in the water between head cover and bottom cover and through the guide vane cascade, stay vanes and spiral casing. These travelling pressure waves are caused by the passing of the runner blades through the wakes trailing from the guide vanes outlets. The frequency of these blade passings is the runner speed multiplied by the number of runner blades at the inlet i.e. including the splitter blades if the runner is furnished with splitter blades. However, inside the runner in the runner channel, the frequency of the pressure shocks will be the speed of the runner multiplied by the number of guide vanes.

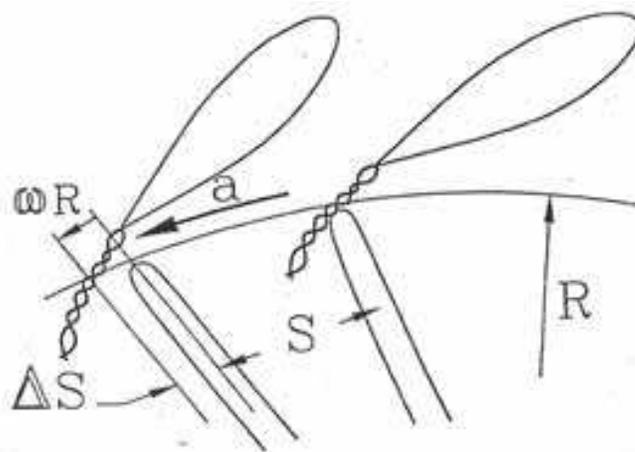


Fig. 1. Illustration of the blade passing of runner blades including splitter blades in a high head Francis turbine. (ω = rad/sec = angular speed)

The described "pressure shock" waves from the runner blades passing through the wakes from the guide vanes in a Francis turbine, may cause vibration and noise from the turbine above acceptable limits in special cases.

Several strain gauge measurements on runner blades in the field and pressure measurements on model turbine blades of high head Francis turbines have also proven the pressure pulsations in the blade channels.

In some cases these high frequency pressure pulsations have caused blade cracking after a short time in operation of a week or so depending on the speed of the turbine. The reason for this has normally been the shape and thickness of the blades and weld defects.

In special cases that will be described in the following, the reason has been very high pressure pulsations caused by a certain combination of runner blades and guide vanes numbers.

It should also be mentioned that resonance with the natural frequency or eigenvalues of the runner is not the reason for the problem of noise and or blade cracking of high head Francis turbines. This will be analyzed and explained later in this section. However, for low head Francis turbine there is normally not observed a similar dominating pressure pulsation of the blade passing frequency inside the runner.

The reason for this is a much larger distance from the guide vanes to the runner blade inlets at the crown compared to the small distance to the runner blades close to the band. The natural frequency of the low head runners may also in some cases be in resonance with the low frequency pressure surges in the draft tube with frequency around 1/3 of the runner speed.

However, by studying fig. 1, valid for high head runners we may find an interaction and amplification between the pressure shock wave from the regarded blade and the blade running in front of it, if the shock propagation speed in water reaches the blade in front of the regarded blade, simultaneously with the passing of this blade through the next guide vane wake.

Then an interaction and amplification of the high frequency pressure pulsations occurs as expressed by fulfillment of following equation and illustrated as in fig. 1:

$$R \frac{\omega}{\Delta S} = a / s \quad (1)$$

In fig.1 and eq.(1), S = distance between the runner blades along the runner rim at runner inlet, a = wave propagation speed in water and ωR = rim speed of the runner blade inlets (R = radius of runner blades at the inlets and ω = angular speed).

In fig. 2 is shown the high head Francis turbine at Hemsil I in Norway. The shock propagating speed in the water was calculated to be around $a = 900$ m/sec taking into account the flexibility of the head and bottom cover. The technical specification for the turbine was $H_n = 510$ m, $P_n = 35.7$ MW and speed $n = 750$ rpm.

The number of guide vanes in the turbine was 28 and the first runner had 30 blades at the inlet. (15 splitter blades and 15 full length blades) and then an interaction of the pressure waves could be proven by a serious noise problem with 120 dB in the air surrounding of the turbine. Because of the high noise level and blade cracking the pressure pulsations were measured between the band at the runner inlet and the bottom cover as illustrated in fig.2.

The pressure pulsations had amplitudes peak to peak of 45 m with a frequency of number of runner blades of 30 multiplied by the speed per sec. = 375 Hz.

The runner with 30 blades was exchanged with a new runner with similar geometry, but with $16+16 = 32$ blades and the noise level dropped to around 80-85 dB which is very good for a high head turbine operating at 510 m net head.

Later several measurements of stress amplitudes at the runner blade outlets of high head turbines have been made, mainly because of blade cracking caused by weld defects and residual stresses in the stainless composition of the weld deposits between blade and band or crown. Some blade cracking problems occurred around 1968 - 1970 because of unstable Austenite in the weld composite of the stainless 16% Cr 6% Ni alloy which was used. However, this problem was solved around 1970.

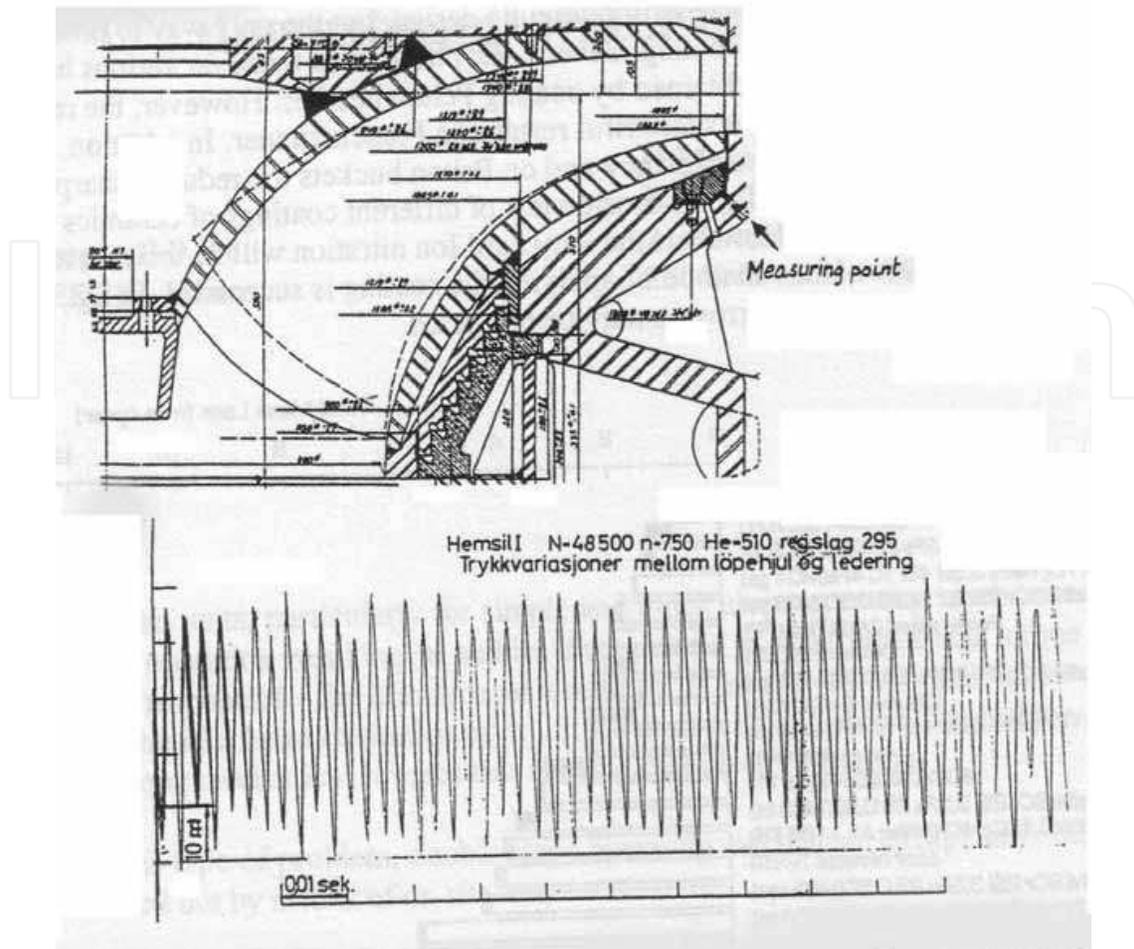


Fig. 2. Cross section of the runner at Hemsil I and the recorded pressure oscillations between runner and bottom cover..

However, the number of runner blades and guide vanes was carefully chosen to avoid interference as described for Hemsil I. An alternative to 32 runner blades and 28 guide vanes was 30 runner blades and 24 guide vanes which also gave a smooth running for the low specific speed Francis turbines operating at heads exceeding 300 m and sometimes exceeding 600 m.

In fig. 3 to the right the relative values of the measured static stress and stress amplitudes peak to peak at pressure side and suction side of the blade outlet of the runner at Tafjord is shown. In this turbine the number of guide vanes was 24 and number of runner blades was 30 and no interference with the runner blade passing the guide vane wakes, caused not acceptable noise. However, high residual stresses in the welds caused blade cracking also because of the pressure pulsations inside the runner in the blade channels even if these pulsations were moderate. The technical data for this turbine is: $P_n = 70$ MW, $H_n = 420$ m and speed $n = 500$ rpm.

To the right in fig. 3 the stress amplitudes from a similar measurement at one of the four turbines at Tonstad, is shown. The number of guide vanes and runner blades are the same as for Tafjord i.e. 24 guide vanes and 30 runner blades. The technical data for this turbine is: $P_n = 165$ MW, $H_n = 430$ m and speed $n = 375$ rpm. Note also the superimposed small stress amplitudes caused by the pressure and stress oscillations from the "wake passing" of the 5 blades in between the next guide vane passing of the regarded blade measured with strain gauges.

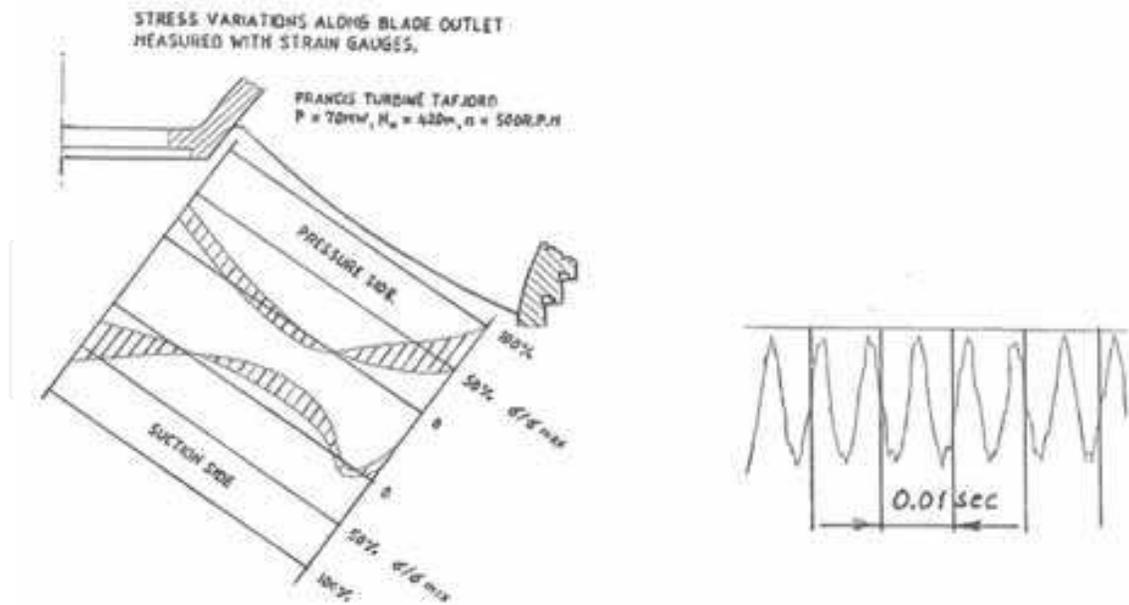


Fig. 3. Static stress and stress amplitudes measured along the outlet edge on pressure side and suction side of a runner blade of the turbine at Tafjord (left) and an illustration of the recorded stress amplitudes from a similar measurement at the blade outlet at one of the turbines at Tonstad. (Made in 1969.)

In fig. 4 is shown the result from a measurement of the stresses amplitudes at the blade outlet of a new runner installed in one of the turbines at Tonstad in 2004.

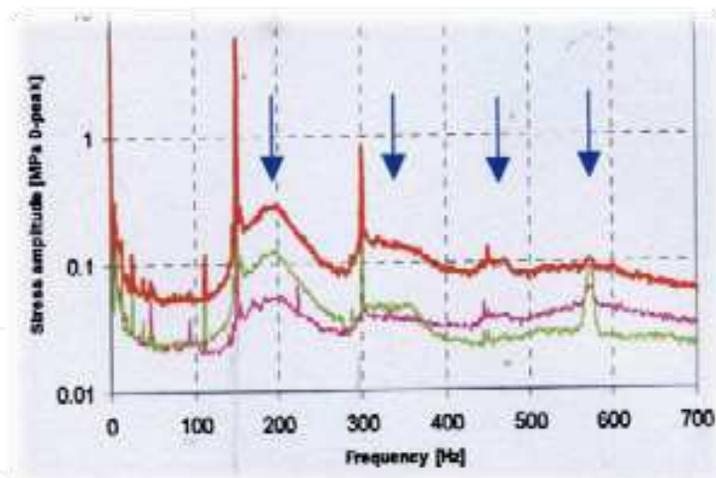


Fig. 4. Illustration of relative stress amplitudes measured at the blade outlets on a turbine at Tonstad compared with calculated natural frequencies marked with blue arrows. Number of guide vanes =24 and runner blades =30 (15 + 15) Speed =375 rpm.(REF. NORCONSULT, GE-KVAERNER)

The blade passing frequency of this turbine was $24 \cdot 375 / 60 = 150 \text{ Hz}$ in the runner channels creating the main stress amplitudes as illustrated as in fig. 4.

The natural frequency of the runner was also calculated and shown with blue arrows in fig. 4. This measurement proves the statement that the stress amplitudes in the blades of low specific speed turbines will get the frequency of the blade passing and with a negligible influence from the natural frequency (eigenvalues) of the structure.

3.2 Vortex shedding and noise from the trailing edge of vanes

Vortex sheading from stay vanes and guide vanes in a Francis turbine may create vibrations and not acceptable noise which may cause problems for the operation of the turbines which will be discussed in this section.

The result from a basic study of the so called Von Kármán’s vortex shedding, presented more than 50 years ago in Transaction of ASME in 1952 and Engineering of Power 1956 and 1960 /Ref. 1/, was used for reshaping the outlet of the fixed guide vanes in a small Francis turbine in Norway with noise problems shown in Fig. 6. The result of the research work presented in ASME /Ref. 1/, is illustrated in fig. 5.

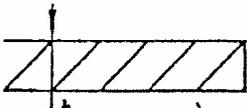
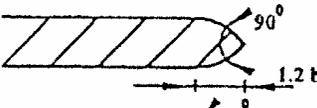
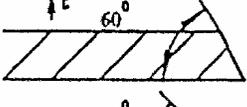
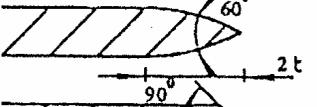
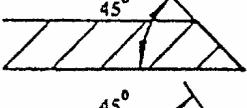
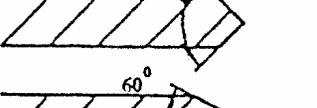
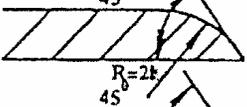
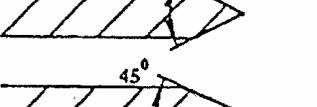
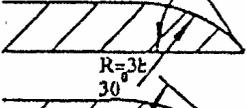
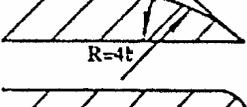
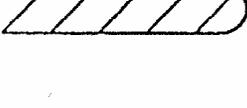
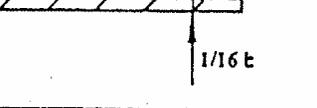
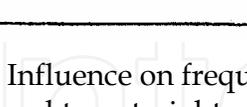
	A	B		A	B
	100 (100)	100		(0)	
	(48)			(0)	
	38 (20)	112		190 (230)	96
	3 (0)	131		380 (360)	93
	0	149		43	117
	0	181		0	159
	0			31 (43)	103
	(260)				

Fig. 5. Influence on frequency and amplitudes by the shape of the outlet edge of plates compared to a straight cut plate. A=relative amplitudes, B=relative frequency./ Ref.1/

From the experiments presented in fig. 1 and described in /Ref. 1/, the frequency of the vortex shedding can be calculated by following equation:

$$f = 190 * \frac{B}{100} * \frac{C}{t + 0.56} \text{ (Hz)} \tag{2}$$

The value = **B** in the equation can be taken from the table in **fig.5** and the relative amplitude = **A** can also be taken from **fig. 5**. Further: **C** = velocity of water (m/sec) at the outlet of the plate and **t** = thickness of the plate (mm). Following conclusion could be drawn by studying this work:

The outlet edge of all stay vane, guide vanes and runner blades in a turbine should have a skewed cut with an angle smaller or equal to 45 deg. measured relatively to the pressure side

of the vanes. However, a sharp corner should be avoided on the edge in contact with the suction side of the next guide vane in turbines with movable guide vanes, and a 1-3 mm wide skew cut should be made parallel to the suction side of the next guide vane side along the outlet edge for sealing in closed position. The width will depend on the size of the turbine. The vortex shedding problem of the previously mentioned small turbine in Norway shown in fig 6, was clearly demonstrated by the noise that was finally cured by reshaping the outlet of the fixed guide vanes which also corrected the power of the turbine.

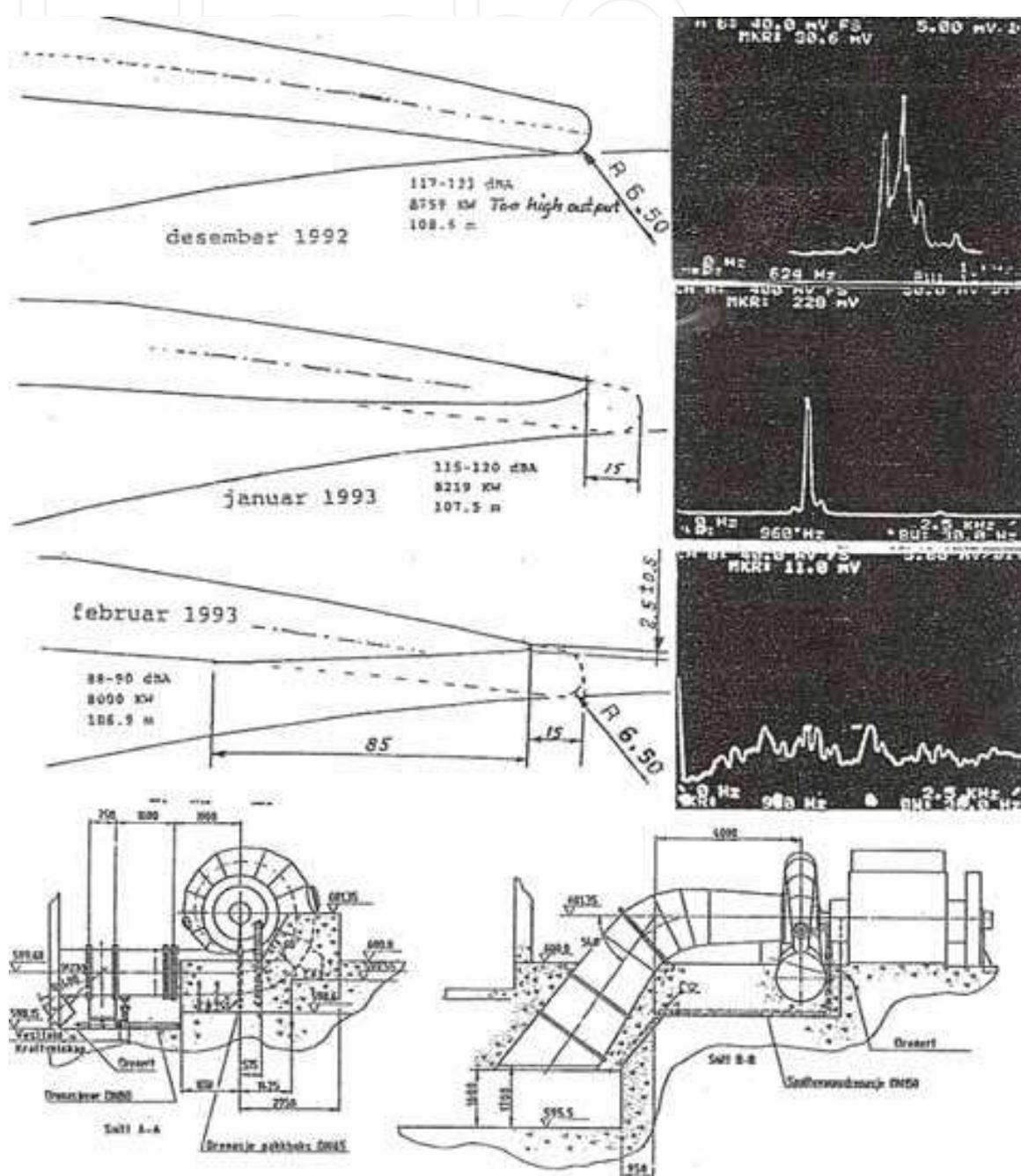


Fig. 6. Original shape and modified shape of stay vanes of a Francis turbine without movable guide vanes, but with stay vanes shaped for the design power with angle and outlet position similar to possible guide vanes. The reason for the illustrated modification was a very high noise and too high power of the turbine with the original vanes.

The original shape of the outlet edge of guide vanes was rounded which is the worst possible shape seen from the fluid mechanical side. However, from the structural side the rounded outlet edge of these guide vanes which also were part of the pressure carrying structure of the turbine seemed to be a reasonable choice. (It should also be mention that a careful stress analysis had to be made for approval of the modification of the outlet edges.)

The history for the improvement of noise reduction in this turbine was as follows. (see Fig. 6).

The original shape with semicircle outlet profile:

Noise: 117-123 dBA, Frequency $\omega=624$ Hz, Power: $P=8759$ kW, Net operational head: $H_n=108.5$ m.

After the first modification with skewed cross section with a smaller flow angle towards a decreasing radius ending with a tangen in 90 degrees to the pressure side of the vane:

Noise :115-120 dBA, Frequency $\omega=960$ Hz, Power: $P=8219$ kW, Net operationalhead: $H_n=107.5$ m.

After the third and final modification with a skewed profile in approximately 30 degree towards a sliightly rounded profile towards a stright cut of 2.5 mm thickness following improvement was obtained: (See fig 6. bottom)

Noise: 88-90 dBA, Frequency $\omega=980$ Hz, Power: $P=8000$ kW, Net operational test head: $H_n=106.9$

By this last modification the problem was solved also for the corrected power of 8000 kW.

4. Off best point operation of Francis turbines with low frequency draft tube pressure pulsations

4.1 Influence from the runner design on part load pressure surges.

The dynamic behaviour at part load has been a major problem for low head and medium head Francis turbines. The main reason for this is unstable swirl flow in the draft tube which is normally caused by the cross flow from crown towards the band on the pressure side of the runner blades. In fig. 7 the so called swirling rope at part load and the contra rotating centre void at full load and overload are illustrated.

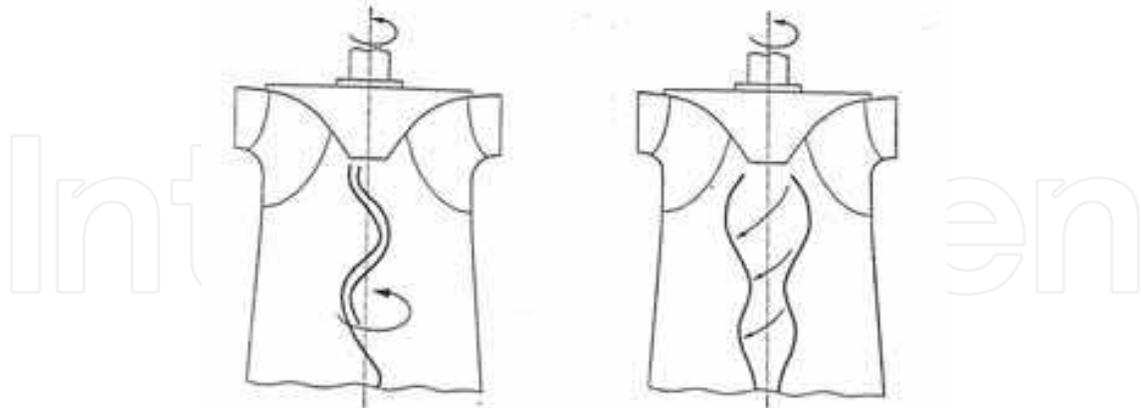


Fig. 7. Swirling part load draft tube rope that may cause unstable conditions and blade cracking (left) and rotating centre void at full load and overload (right).

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 cross flow by a negative blade lean at the inlet of the runner. Further the pressure distribution on the pressure side of the blades must be balanced all the way towards the blade outlet by adjusting the blade lean. This philosophy must be used during creation of a new runner.

The described method illustrated by the equations in fig. 8 bottom, is based on potential flow analysis which is valid for a runner with infinite number of blades. However, the

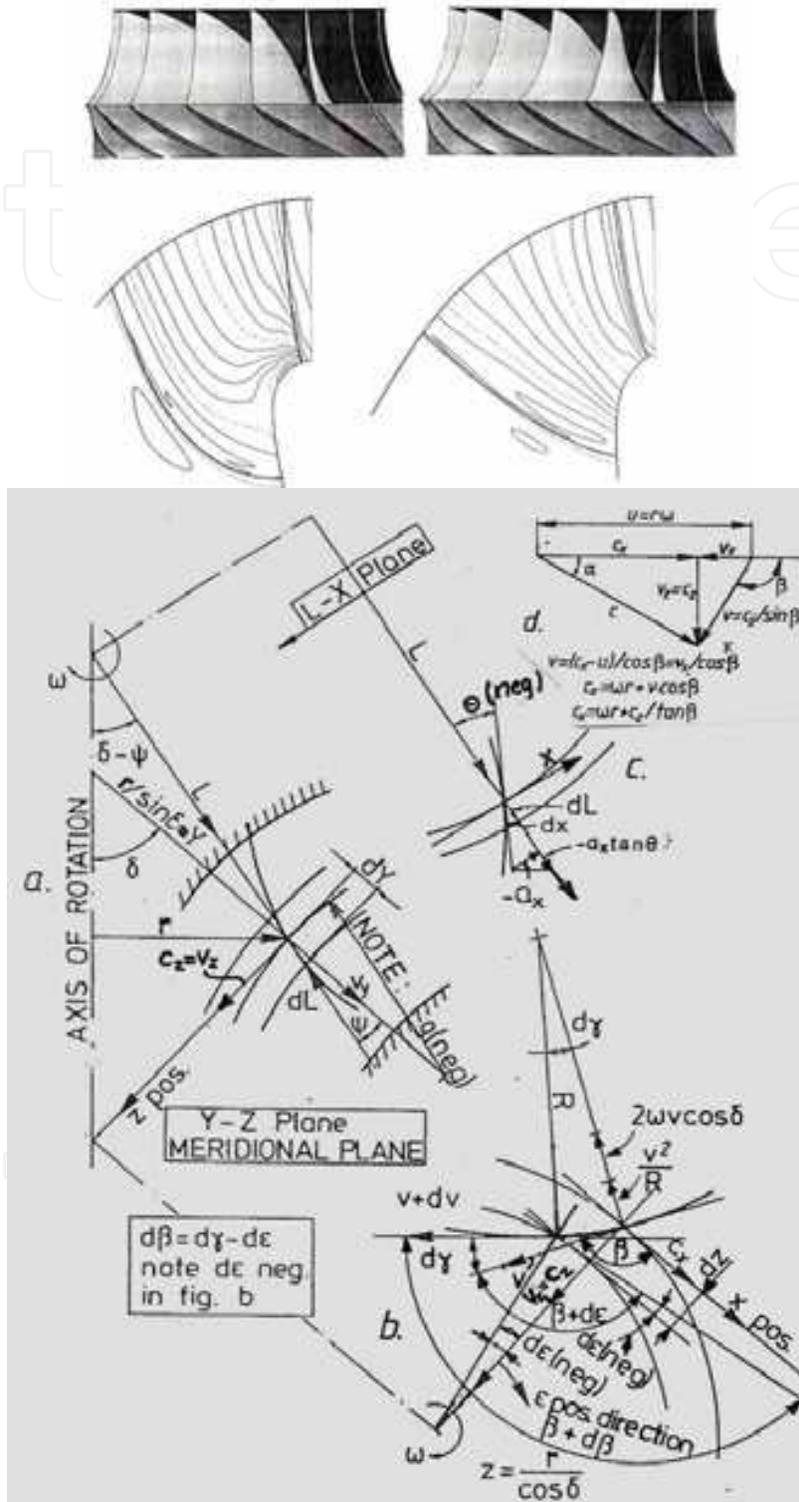


Fig. 8. Pressure distribution of a traditional low head runner (top left), an X-BLADE runner with the same specific speed (top right) and definition of geometry for a potential flow analysis for creation of the blade shape (bottom). Note the difference in pressure towards the band of the traditional runner compared with the X-Blade runner. (Top, left and right.)

following CFD analyses of runners designed by this method, has proven that the pre designed runner geometry gives an excellent starting point for creating a high efficiency runner with stable flow in the draft tube also at part load.

The most famous runners made by this method were the **X-Blade** runners for **Three Gorges Power Plant** in China. These runners operate from 61 m to 113 m net head with a non - restricted range of operation and no operating problems have occurred for the X-Blade runners according to reports. The blade lean angle at the inlet of a pressure balanced so called **X-Blade** runner, will be negative, but the blade must be twisted towards the outlet the other way to an angle similar to a traditional runner so the inlet edge and outlet edge forms an X when looking through the runner from inlet to outlet and this is the reason for the name X-Blade runner.

The blade lean angle for a runner blade is defined by the angle between the blade and a meridian plane perpendicular to the stream surface through the axis of rotation, denoted by Θ as illustrated in fig.8 (bottom) together with the other parameters used in the equations (3), (4) and (5). The dimensionless pressure gradient $d\bar{h}/dy = d(h/H_n)/dy$, perpendicular to the stream surfaces can be expressed by equation (3) 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.8 (bottom). The equation for the pressure gradient ($d\bar{h}/dy$) 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 as expressed in equation 3.

$$\begin{aligned} d\bar{h}/dy = 2[[(1/R - \cos\delta\cos^3\beta/r)(\underline{C}_z^2/\sin^3\beta) - \underline{C}_z(\delta\underline{C}_z/\delta z)/\tan\beta + 2\underline{\omega}\cos\delta\underline{C}_z]\tan\Theta \\ + (\sin\delta/(r\tan^2\beta) + 1/\rho)\underline{C}_z^2 + (2\underline{\omega}\sin\delta/\tan\beta)\underline{C}_z + \underline{\omega}^2r\sin\delta] \end{aligned} \quad (3)$$

For a pressure balanced blade $d\bar{h}/dy = 0$ when the blade lean angle Θ is adjusted correctly as illustrated by the X-blade runner in fig. 8 to the right on top.

Equation (3) is as described above based on the equilibrium of forces and is two dimensional, valid for a runner with infinite number of blades i.e. potential flow.

In addition to equation (3) **the ROTHALPY equation (4)**, expressing the hydraulic pressure along a stream line, must be established together with the **equation of continuity (5)**. The presented theory is as mentioned two dimensional (i.e. potential theory valid for an infinite number of blades). However, this theory 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. The final detailed shaping will be made by a full 3D viscous analysis. By making this simplified study, it is quite clear that a negative blade lean on the inlet of the runner was necessary in order 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 at turbine manufacturer KVAERNER where the author of this chapter was working part time as technical advisor./Ref.2/.

$$\underline{h} = \underline{\omega}^2 r^2 - \underline{C}_z^2 / \sin^2\beta + (1 - 2\underline{U}_1\underline{C}_{u1}) - J \quad (\text{J is estimated loss along a stream line.}) \quad (4)$$

In addition to eq. (3) and eq.(4) the equation of continuity in the channel between two stream surfaces is used to find the meridian velocity C_z . (**J = 0.01-0.02**)

$$\underline{C}_z = (Q/N)/(2\pi r b \Phi) \quad (5)$$

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

4.2 Part load problems of Francis Turbines

In Fig.9 to the right is shown the cross section of a traditionally designed runner installed in the turbine at the power plant Frøyntul in Norway and to the left the relative values of the measured stresses on the outlet of the runner blades is shown. The magnitudes of the oscillating stresses did not allow continuous operation below 65 % load for this turbine due to danger of blade cracking. After synchronizing, the opening of the guide vanes had to be fast to reach the minimum allowable opening. The technical data for Frøyntul is: $P=37.7$ MW, $H_n=54$, m, $n=214$ rpm.

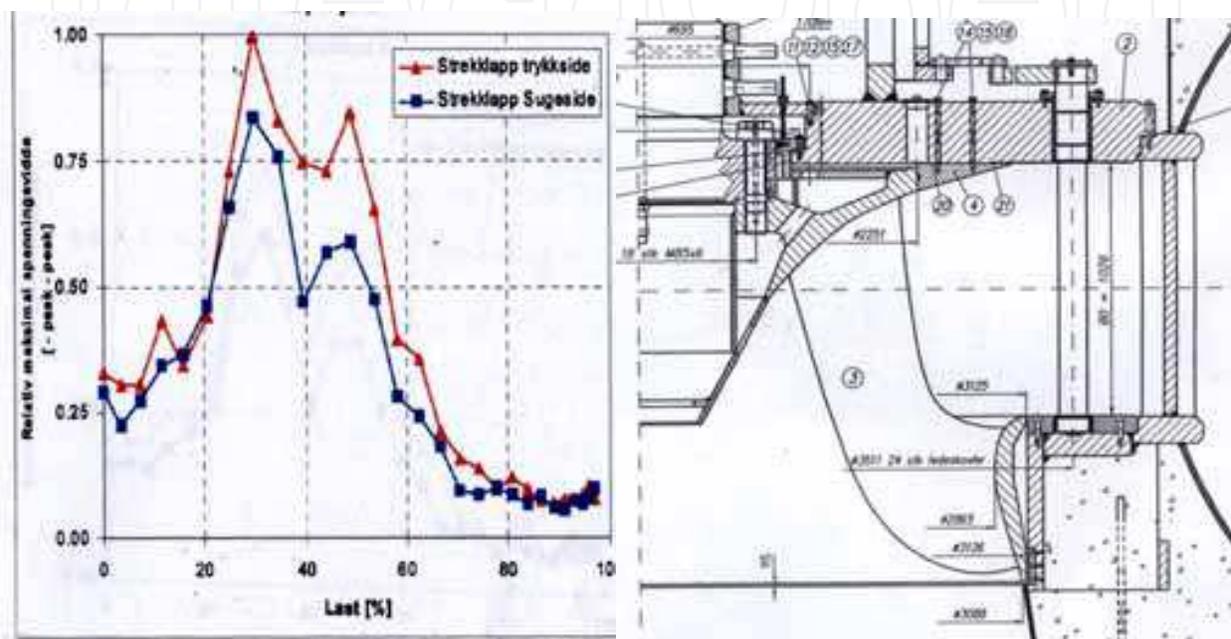


Fig. 9. Cross section of the low head turbine for Frøyntul power plant in Norway (right) and the relative values of stress amplitudes on the runner blade outlets (left). (Courtesy: KVAERNER, NORCONSULT)

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. It should be noted that the highest value of stress amplitudes measured on this first X-BLADE was below 25% compared to the same scale as illustrated in the diagram to the left in fig. 9. From 30% load and up to full load the stresses were below 15% with the same relative scale.

For comparison of geometry the cross section of the runner for Bratsberg is shown to the right in fig. 10. Even if the turbine at Bratsberg has a lower specific speed than the turbine for Frøyntul, a comparison is still valid.

The technical data for the turbine for Bratsberg are: $P= 60$ MW, $H_n= 130$ m and $n=300$ rpm.

The technical data for the turbine for Three Gorges proves a higher specific speed than Bratsberg and closer to Frøyntul, but still this turbine is running very smoothly as illustrated in fig. 11. The technical data are: $P_n = 710$ MW at $H_n = 80,6$ m and $n=75$ rpm and $P=852$ MW at $H_n = 92$ m/Ref. 4/.

Finally the measured pressure oscillations in the model turbine for Three Gorges is shown in fig.11./Ref.5/. However, the results should be similar for X-blade runners for high and medium specific speed if they are well designed.

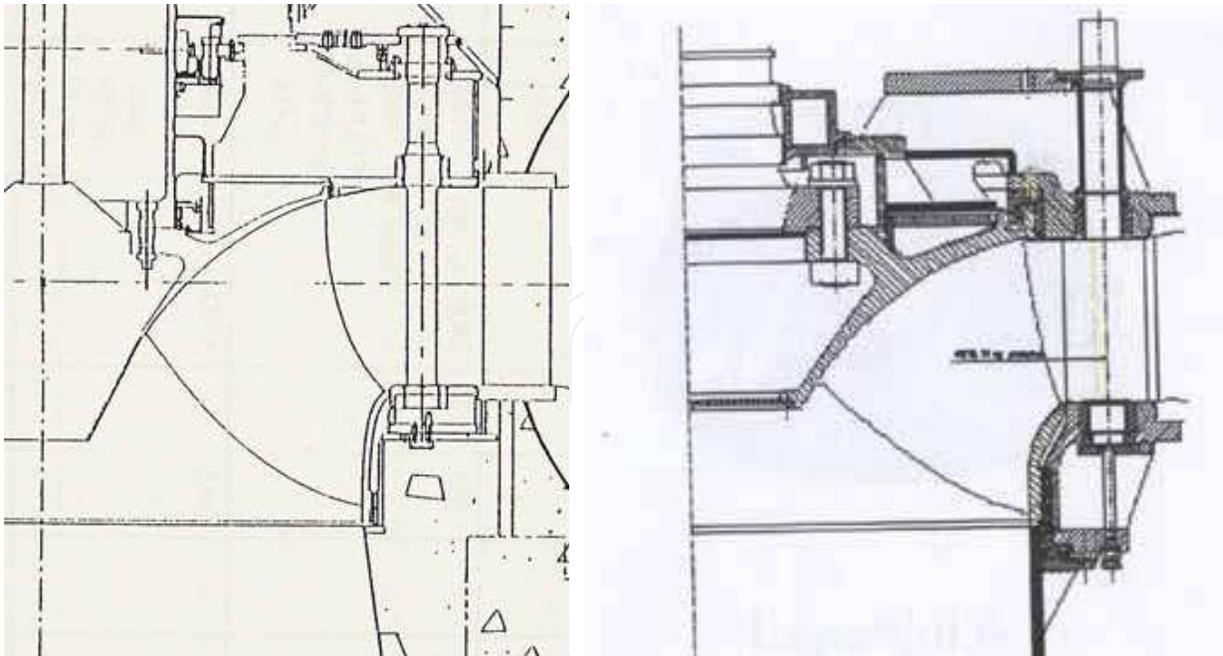


Fig. 10. Cross section of the Francis turbine for Bratsberg with the first X-BLADE runner in operation. (Right) $P_n = 60$. MW, $H_n = 130$ m, $n = 300$ rpm and the runner for Three Gorges with a higher specific speed (Left) with $P_n = 710$. MW at $H_n = 80.6$ m and $n = 75$ rpm.

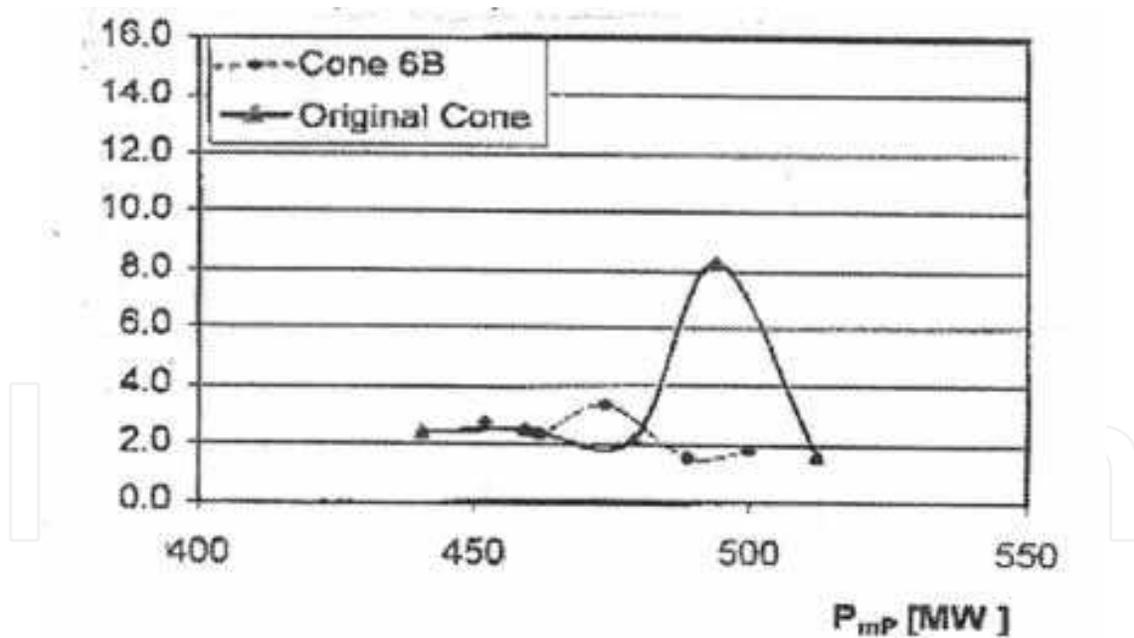


Fig. 11. Measured pressure oscillations peak to peak in % of nominal net head measured in the draft tube of the model turbine for Three Gorges before (full line) and after an optimized semi tapered centre piece was installed at the outlet of the hub (dotted line). Note the power refers to prototype. (Courtesy: GE Hydro (Kvaerner))/Ref.5/

During model testing of the X-blade runner for Three Gorges Power Plant a thoroughly research on pressure pulsation was made. Special attention was made on the influence of a tapered centre cone attached to the outlet cone of the runner.

$$f = \left(\frac{1}{2\pi} \right) \sqrt{\frac{gA}{LA_{eq}}} \quad (6)$$

In equation (6), g = the gravity constant (=9.81), A = middle cross section of draft tube, L = length of draft tube and A_{eq} = the equivalent cross section of a fictitious area of a water level in a surge shaft for a tunnel expressed by the air volume and the polytropic exponent κ of the saturated air in the void underneath the centre of the runner as expressed in equation (7), based on experience from air cushioned surge chambers in Norwegian power plants. (Note also that the area of the water level A_w in fig. 12 has been neglected due to smallness in the accumulator system illustrated.)

$$A_{eq} = V_o / (\kappa * H_o) \quad (7)$$

In equation 7, V_o = volume of the vapour and air filled void underneath the runner in the draft tube at full load, H_o = water pressure in the void underneath the runner, $\kappa = 1.3$ is the polytropic exponent for water saturated air in accumulators /Ref. 3/ and H_o = water pressure in draft tube underneath the runner.

In the system illustrated in fig. 12 attention should also be paid at the natural frequency of the penstock with length = L_p from the turbine to the water level in the surge shaft at the top of the penstock. For the two turbines with the shortest penstocks at the large high head power plant at Kvittdal in Norway large pressure pulsations and power oscillations occurred at full load partly caused by resonance with the penstock. The equation for the first, the third, and fifth harmonic of the reflected water hammer pressure waves yields:

$$f_p = a / 4L_p, 3a / 4L_p, 5a / 4L_p, \dots \quad (8)$$

In addition to these frequencies we know that the natural frequency of the speed of the generator rotor in the magnetic field of the stator will be between 1 and 2 Hz depending on the connection to the machines in the power house.

In the power house of Kvittdal 4 turbines of $P_n = 315$ MW, $H_n = 520$ m and $n = 333.3$ rpm each was installed connected to separate penstocks leading to a common air filled surge chamber.

By running the two turbines connected to the shortest penstocks, power oscillations and pressure surges occurred at full load with the frequency between 1.4 and 1.5 Hz. The power oscillations were recorded to be 60 MW even with locked guide vanes in fixed position. However, by switching off the voltage governor during testing the power oscillations stopped. Further, by again switching on the voltage governor the oscillations came back.

The possible natural frequencies of the penstock were according to eq. 8 also between 1.4 and 1.5 Hz.

By using data for the draft tube and assuming we had an air and vapour volume = $V_o = 0.5$ m³ underneath the runner based on observations from model tests and the absolute pressure $H_o = 5.0$ m WC and the average cross section of the draft tube $A = 5.0$ m² with known length of $L = 30$ m and setting the polytropic exponent for vapour saturated air $\kappa = 1.3$, we finally got the natural frequency of the draft tube system to be:

$f = 1.46$ Hz which was very close to frequency of the measured oscillations.

Further the natural frequency of the generator rotor was reported to be between 1.4 and 1.5 Hz and the first harmonic of the water hammer frequency of the penstock was observed during shut down tests and calculated to be $f_p = a / (4L_p) = 1.4$ Hz.

The reason for possible power oscillations of the two high head Francis turbines with the shortst penstocks at Kvilldal may than be explained as follows, based on the fact that the turbine characteristic diagram for a high head Francis turbine has strongly sloping characteristic curves where any increase in speed will reduce the flow and create pressure rises. /Ref. 4/

Following conclusion based on the full load surges at Kvilldal yields:

A rising pressure will increase the speed and then the flow will decrease because of the sloping turbine characteristic of the high head Francis turbines. We then get a positive feed back in the system because the reduced flow caused by the increased speed will in turn give a further increase in the pressure rise especially because the flow oscillations in the penstock and the pressure surges in the draft tube had the same natural frequency, as the generator rotor.

To solve the problem at Kvilldal a damping device was installed in the voltage governor which damped the rotor oscillations and then the connection to the turbine characteristic was broken and the no power oscillations occurred.

5. Concluding remarks

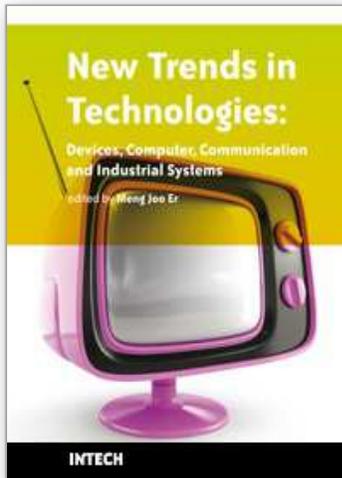
Besides the research on oscillatory problems to avoid not acceptable pressure pulsations and fatigue problems, the goal for the research of turbine design must be aimed at following important factors:

EFFICIENCY, SAFETY, MAINTENANCE AND LIFETIME.

The reason for this is the growing need for peak power and because hydro power is the most important non-polluting source for peak power and then safety, reliability and life time will be most important.

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The grandest accomplishments of engineering took place in the twentieth century. The widespread development and distribution of electricity and clean water, automobiles and airplanes, radio and television, spacecraft and lasers, antibiotics and medical imaging, computers and the Internet are just some of the highlights from a century in which engineering revolutionized and improved virtually every aspect of human life. In this book, the authors provide a glimpse of new trends in technologies pertaining to devices, computers, communications and industrial systems.

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