Design, Performance and Maintenance of Francis Turbines

By Hermod Brekke

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The structural design has moved from castings and riveted plates to fully fabricated structures of high tensile strength steel in the stationary parts and stainless 13/4 Cr/Ni or 16/5 (17/4) Cr/Ni steel have substituted the 13/1 Cr/Ni in runners. The paper also includes an ancient runner design with plate steel blades moulded in cast steel at crown and band. Such high head runners, put in operation in 1950, have been in operation in good condition Norway until about 15 years ago.

A discussion on stress analyses and fatigue problems of pressure loaded parts and high frequency fatigue in runners, will be presented.

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I. Introduction

The paper is based on the authors experience in Norway where the hydropower installation was increased from 7000 MW in 1959 to 30 000 MW with an annual production of 126 Two during his work at KVARNER HYDRO from1959 to1987 and associated work as Technical consultant until 2003.

From 2003 up to present the author has worked as Technical consultant in Norway and abroad.

The description of hydraulic and structural design of high head Francis turbines and Pelton turbines has been based mainly on the experience from Norway.

Experience from design and performance of both high head and low head turbines is given with aspecial attention to the pressure balanced X-Blade runner that was developed for Three Gorges in China.

a) The Design and Development of Turbines in Norway

i. A brief history of turbine production in Norway

KVAERNER BRUG Ltd. was a major turbine manufacturer in Norway until 1997 when it was sold to GE.

The company was founded in 1853 as a foundry and machine shop and the first produced turbine had an output of 170 kW designed for a head of 11.3 m.

Up to 1890 Kvaerner Brug produced 100 water turbines mainly for paper mills.

The first water turbine made in Norway for electricity supply was delivered in 1890 based on old turbine design.

In 1895 the first Francis turbine was produced and in 1898 the first Pelton turbine was made in Norway. Owing to our cold climate and dark winter nights, the demand for electricity supply for civil purposes was growing together with a growing electromechanical industry in the 19th century. As a result of this demand, the turbine production for electricity supply was fast growing especially after World War I.

In 1911 the Norwegian University of Science and Technology was founded and the Water Power Laboratory at the University was finished in 1914, putting the Norwegian research for turbine design on the map of Europe. The first industrial project was the competitive model tests for Solbergfoss Power Plant between the Norwegian manufacturer kvaerner brug and another Norwegian manufacturer, myrens verksted. (later myrens verksted closed down its turbine production.) The model test for the turbines for Solbergfoss was won by KVAERNER BRUG with peak efficiency on the model of 94.4%. The technical data for the prototype was P=8 500 kW, with a net head of Hn=21 m and a speed of n=150 rpm. / Ref. 1/. (The name KVAERNER BRUG will be denoted as KVAERNER in the following.)

ii. The hydro power plant development in Norway

The mountains in Norway consist of good quality rock so the conduit tunnel systems in all high head projects built after 1950 normally consist of a long, often complicated main tunnel system with a surge shaft, leading the water from the main reservoir and small rivers in the catchments to a lined or unlined pressure shaft down to the a cavern power house.

In some cases the tunnel system has been connected to more than one reservoir, furnished with water level controlled gates.

b) Large scale turbine production in Norway up to present time

Model test facilities for Pelton Turbines was built at the workshops at Kvaerner Brug while model tests for Francis turbines was run at the laboratory at The Norwegian University of Science and Technology (NTNU) up to 1984, when a new laboratory was built and operated by KVAERNER near the NTNU.
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i. Francis turbine development

It is of interest to study the list of the most important high head Francis turbines produced by Kvaerner, and put in operation in Norway after World War II.

These turbines were made for Hol: \(H_n=395\text{m},(2\times30 \text{ MW})\) 1946, Vinstra: \(H_n=420\text{ m},(2\times50 \text{ MW})\) 1948, Hemsil I: \(H=510\text{m},(1\times35.7 \text{ MW})\) 1959, Kvisdal: \(H=520\text{m} (4 \times315 \text{ MW})\) 1983, Kobbelv, \(H_n=590\text{ m} (2\times150 \text{ MW})\) 1985, Svatrisen: \(H_n=543\text{m} ((1+1)\times350 \text{ MW})\) (1990 and 2008).

In the 60th the turbine design changed from a fully cast steel design to a welded design based on rolled plates with a higher strength (Fig 1.). During the period from 1958 to 1976 the weight of spiral casings per kW turbine power was reduced to 1/3 when changing from cast steel design to welded steel plate structure.

The ultimate strength of the plates used also increased from the TSt E355 to TSt E460 during this period, in order to meet the demand of increasing size of the turbines.

However, the control and repair of possible weld defects was very important in order to avoid problems with fatigue ruptures caused by low frequency cyclic load of start stop and water hammer oscillations. This is because the crack propagation speed of a given welding defect is increasing with increasing stress. And the life time should be based on at least 50 000 load cycles or start- stop cycles, for the pressurized parts.

The material used allowed for relatively large final size of growing weld defects or material defects before ruptures occurred.

For safety the final crack size that is leading to a rupture should be allowed to penetrate the plate thickness leading to a leakage that will be detected before an explosive catastrophic rupture.

Even if this criterion LEAKAGE BEFORE RUPTURE is fulfilled a crack in the stay vanes does not give a warning by a leakage before unstable rupture occurs. Then periodic control has been required of the stay vanes where no leakage occurs even for large cracks. However, for very large turbines and turbines operating at extreme high heads, the plate thickness will not allow for a crack size that will penetrates the plates before rupture.

Then periodic inspections for growing defects are very important also for the shell in spiral casings of francis turbines besides the stay vane control for large turbines.

The runners were up to the late 50th a designed with pressed steel plate blades melted into cast steel crown and band as shown in fig.1. Further all high head Francis runners made in Norway had splitter blades. of interest is also that the high head turbines at both Hol and Vinstra operating at 395 and 420 m net head respectively, were both designed with the type of runners as shown in fig.1 (top). It should be noted that these turbines were in operation without any problems until 2008 and 2005 respectively when new runners with increased power were installed.

It should also be noted that the splitter blade runners are smooth running over the whole range of operation from no load to full load, which has traditionally been required for the peak load operation in the Norwegian system with the major electric load coming from electric furnace industry on the west coast. Even at present time isolated load with unlimited variation from electric furnace industry may occur if the high voltage lines from the west coast to the east of the country are broken during winter storms.

When the turbine for Hemsil I was designed for 520 m net head, the design with presssed steel plate blades welded to band and crown was introduced for the first time for high head Francis runners.

The welded design was in the beginning around 1958 made with a mixed steel quality of stainless steel and carbon steel partly with stainless overlay welding. Later the Swedish quality now standardised as EN10283, denoted as 16/5 Cr Ni Mo steel, was used in the runners made by Kvaerner. Some welding problems occurred in the beginning when using this material, because unstable Austenite was formed in the weld composite caused by a too high content of N\(_2\) in the mantels of the welding electrodes.

After changing to another type of electrodes, this problem was solved /Ref. 2/, /Ref.3/.

By changing electrodes and steel quality to 13/4 Cr Ni Mo (EN 100088-2, X3CrNiMo13-4), the austenite problem was solved, but a higher pre-weld temperature during welding was necessary.
ii. The performance of high head Francis turbines

In fig. 2 the efficiency measuring result of 27 turbines with the same specific speed compared to the model turbine produced with a similar roughness as the prototypes, is plotted as function of Reynold’s number. The Reynold’s number Re is defined as the product of the rim speed and diameter of the runner outlet divided by the viscosity i.e. $Re = \frac{U_2 \cdot D_2}{\nu}$.

In the same diagram the efficiency of the model turbine is included for comparison.

The roughness of the polished parts of the prototype runner outlet and the machined guide vanes and the runner outside are between Ra 1.6 to Ra 3.2 while the rest of the internal parts are sand blasted and painted.

The spiral casings and stay vanes had a sand blasted painted surface.

In fig. 2 is also shown the influence from variation of the end clearance of the guide vanes by reducing the gap on the model and one of the prototypes as indicated in the figure. By sealing the gap of the end clearance on the model the highest efficiency was obtained. See fig. 2 left side.

The reason for the existing end clearance on the prototypes is the deflection of the head and bottom covers caused by the water pressure. In a field test a reduction of 0.35 mm was obtained in one prototype by changing the pre stressing of the bolts on the bottom cover as shown in fig 2 top right.

Today normally end seals in the guide vane facings are used to reduce the end clearance for large Francis turbines.
Figure 2: Efficiency versus Reynold’s number of low specific speed turbines with speed number $\Omega = 0.28$. The speed number is defined as $\Omega = \omega Q^{0.5}/(2g^*H)^{0.75}$ referring to best efficiency i.e. $N_{OE} = \Omega(Q_{ED}/Q)^{0.5}/(2^{0.75} \pi) = 0.297 \Omega$. $U_2$ = circumferential speed, and $D_2$ = outlet diameter of runner.

At KVAERNER little effort of making low head Francis turbines was made in Norway, because this expertise was in Sweden at the companies NOHAB and KMW which were bought by KVAERNER in the 80s. However, in order to compete in the bid for the Three Gorges Project in China a new design of Francis runners, the so-called X-BLADE runner was initially designed by the author of this paper and patented in collaboration with two engineers at KVAERNER, Olav Rommetveit and Jan Tore Billdal who ran the CFD analyses.

The advantage of this runner design was that the blades had a strong negative blade lean at the inlet by letting the joint between the band and blade run in front at the joint between blade and crown. In addition a balancing of the blade lean towards the outlet was necessary in order to increase the pressure at the band all the way from inlet to outlet.

In fig. 3 the pressure distribution on the suction side of the blades is compared for a traditional runner to the left, a runner with increased negative blade lean and a fully pressure balanced X-blade runner to the right.

Figure 3: CFD analyses illustrating the pressure distribution on suction side of a traditional runner (a), on a runner with increased blade lean angles at the inlet (b) and a runner of new design with balanced blade lean angle i.e. similar to the developed X-blade Runner for Three Gorges in China (c).

II. High Cycle Fatigue

a) High head Francis turbine runners

High cycle fatigue problems in hydraulic machinery will always be related to the rotating parts, normally driven by the blade passing frequency and for high head turbines normally without any resonance with the runner structure.

From time to time, blade cracking problems of high head runners of Francis turbines has also been reported.
An example of such problem was presented in /Ref. 4/. From experiences in Norway where the installed hydropower capacity is mainly high head turbines, a brief discussion of some blade cracking problems will be given. The reason for the blade cracking and the solution of the problem will also be given.

Further it should be noticed that the turbines in Norway normally have been operated from no load to full load without restrictions. The reason for this is our high electric furnace production and the requirement that all major power plants should be able to be operated on isolated load in case the long transmission lines across the mountains are broken during the winter by snow, ice or storms.

b) Design and hydraulic dynamic load on Francis runners

The first fully welded high head runners in operation in Norway was the runners for the two 35.7 MW turbines operating at 510 m net head Hemsil I Power plant.

These turbines had 28 guide vanes and 30 runner blades (15 full length blades and 15 splitter blades. The technical data for the turbines are: P=35.7 MW, Hn= 510 m and n=750 RPM .

At the inlet of a Francis runner a pressure shock in the water occurs each time a runner blade is passing through the wakes from a guide vane giving a pressure shock in the flow. The reason for this is the lower velocity in the guide vane wake compared to the average velocity from the guide vane system. Interference may also occur if the shock wave in the water from the blade passing reaches the runner blade in front of the regarded blade when it is passing through the next guide vane wake. This pressure shocks are travelling down the runner blade channels with the frequency of the blade passing of the guide vanes. In the upstream side of the runner inlet the frequency of the runner blade passing through the guide vane wakes in the opposite direction through the guide vanes and stay vanes channels into the spiral casing and on the outside of the runner creating the sound of the turbine.

The high speed combined with the chosen number of runner blades and guide vanes of the high head turbines at Hemsil power plant caused an amplification of pressure pulsations resulting with a sound level of 120dB in the power house as described in fig. 4.

c) Discussion of blade fractures in high head Francis runners

From time to time blade cracking at the outlet of high head Francis runners have been reported. A short description of observed blade cracking and the measured stresses and the reason for the problem which was mainly residual stresses caused by differences in the Martensite/Austenite balance in the welding composite of 16/5 CrNi steel around 1970 in Norway.

Blade cracking in high head Francis runners have also been reported after 2004. In those cases the reasons have been the geometric shape which was introduced in order to improve the efficiency. The measured stress amplitudes on a runner blade outlet caused by blade passing of the guide vane wakes pressure are illustrated in fig.3 for a typical high head Francis runner At Tonstad Power Plant. Turbine data: P=165 MW, Hn=430 m, n=375 rpm. (Measurement made by J.E Syljuset 1969).
In addition to the hydraulic blade loading, residual stresses from welding has a negative effect on the resistance against blade cracking but this problem will be discussed in the next chapter.

![Figure 5](image1)

**Figure 5**: Location of strain gauges on blade outlet of the blade (Left) and the measured stresses (Right) of measuring point 1 = pressure side band top and measuring point 2 = suction side band bottom. (kP/cm² = 10^6 MPa) of one of the Francis turbines for Tonstad at full load. (Turbine data: P=165.4 MW, H=430 m, n=375 rpm.) (Ref. Report J.E. SYLJESET 1969)

![Figure 6](image2)

**Figure 6**: Example from registration of strain amplitudes at blade outlet of the turbine at Tonstad. Note the 5 small impulses superimposed on the main impulse. The reason is the small pressure shocks in the water from five other blades passing their wakes inbetween each passing of the regarded blade.

From the measurement at Tonstad we find the maximum static stress in measuring point 1 on pressure side at the band as the critical point with an amplitude of 690 N/mm² and a static stress of 100 N/mm². Because of the skewed outlet edges of the blades the stresses at the hub was low in this case. This will be different for other geometries.

d) **Welding and Heat Treatment of 13/4% Cr/Ni and 16/5% Cr/Ni Steel**

In this chapter a brief description is given on the balance of creation of Ausenite during welding and heat treatment of 16/5% Cr/Ni steel and the reason for cracking of Francis runners in 1969 in Norway.
A similar problem also occurs during welding of 13/4% Cr/Ni steel but the only difference is that creation of Martensite during cooling of from melted condition to solid condition happen at 100 degree higher temperature.

A short summary of welding problems with influence from the chemical components will be given.

In fig. 9 the influence from both Ni and N on the creation of the temperature where Austenite is transformed to Martensite denoted as Ms in the diagramme.

![Figure 7](image)

**Figure 7**: The influence on the temperature where Martensite starts to form during cooling (Ms) as a function of the content of (Ni + N). In addition the content of (δ ferrite) as a function of (Ni + N) temperature is shown. [Ref. 3]

The high content of Austenite in the welds was proven to be caused by a high Ni content in the weld deposit as described in [Ref. 3]. Further investigations in 1969 at Kvaerner Hydro showed that in this case the N2 creation in the weld composite was caused by N2 added in the mantels on the covered electrodes.

The reason for adding N2 in the mantels on the electrodes was to improve the welding process by pushing the Ms point to a temperature of approximately -90 °C and thus avoiding Martensite creation and brittleness in the weld deposit that might cause cracking in the welds during welding.

However, the Austenite caused residual stresses close to yield point in the welds causing blade cracking starting from very small weld defects caused by the high frequency dynamic hydraulic load. The electrodes were changed to normal electrodes without added N2 in the mantels for 16/5 CrNi steel and the problem was solved. The welding procedure was more difficult, but it was necessary to obtain homogenous composition of the welds against the base material.

In fig. 10 is illustrated the dilation (relative elongation) during cooling and solidification of the weld deposit during welding (1) followed by the stress relieving heating 2 and final cooling 3 with dotted line. The stress relieving heating temperature will be around 580 °C and the Ms POINT WILL BE AROUND 100 °C and the finished quality Mf is obtained at room temperature.

It should also be noted that the fillet radii were too small, creating high stresses from the static and dynamic hydraulic load, in the case shown in fig.7. Similar measuring results with similar results were made in 3 other power plant in Norway at that time around 1969 for the same reason. After repair and change of electrodes this problem was solved.

However, in 2004 a fatigue problem on a high head turbine were reported in Canada (Ref. 1).
The reason for the blade cracking on this case was not residual stresses from welding, but probably the shape of the runner blade outlets. In addition the periods between inspections for a possible crack growth were too long, resulting in large blade ruptures at the first inspection. The runner outlet is shown in fig.11 to the left published in /Ref.4/.

Similar problems occurred also in two power plants in Norway in 2008 at Driva (Hn=540 m, P=71.3 MW, n=600 RPM) and Sunnå Høy (Hn=540 m, P=106.2 MW, n=600 o/min).

No report on the blade cracking at Driva has been published, but for Sunnå Høy the cracking stopped by cutting the blades towards a straight line. However the output of the runner increased so the new runners had to be made for this plant.

There are similarities of the blade cracking at San Marguerite and Sunnå Høy. In spite of the more skewed blade outlets at Sunnå Høy the similarity is the strongly curved outlet edge of the blades reducing the angles towards the rim and hub.

The difference between a straight outlet edge and a curved outlet edge is illustrated in fig. 11 to the right.

For a comparison the outlet of a traditional high head runner with high efficiency made by Kvaerner, is illustrated by the a small model in fig. 12 to the left, and compared with the runner for Sunnå Høy before modification to the right.
Figure 10: Traditional Kvaerner runner (Lu Buge Power plant in CHINA) with skewed relative straight blade outlets (left) compared with the blade outlets of Sunnå Høy, where the cracked blades are shown. After repair and cutting the blade outlet edges straight no further cracking was observed.

e) High Cycle Material Testing of Crack Propagation

Referring to the material testing based on high cycle fatigue, tests of test pieces with a defined start defect and given material quality have been used. Defects have been made by SULZHER and presented by H. Grain et al. in /Ref. 5/. The inspection of new low specific speed runners should be made after full load operation approximately each week during the first month of operation.

The reason for this is illustrated in the Paris (Improved Forman equation) diagram illustrated in fig 13. /Ref 5/

In fig 13, the threshold value $\Delta K_0 = \Delta \sigma \sqrt{a} \varphi$, where $\Delta \sigma =$ peak to peak stress amplitudes and $a =$ depth of a surface crack in mm and $\varphi = 1.26$ for a semi elliptic crack with length $c = 2a$ where $a$ is the depth of a surface crack. For a buried crack below surface the depth is $2a$.

As an example we can use the values presented in fig.13, where $\Delta \sigma = 70$ MPa and the static pressure is 100 MPa. Then the value of $R = \sigma_{\text{min}}/\sigma_{\text{max}} = 70/140 = 0.5$. From fig.13 we find $\Delta K = 72$ and then the size of a crack or defect with sharp edges that will not grow even after infinite number of stress cycles as follows based on the test results shown in fig. 13.: 

Depth: $a = \Delta K_0/(\Delta \sigma \varphi) = (72/(70*1.26))^{1/2} = 0.67$ mm,

Length: $c = 2a = 2*0.67 = 1.34$ mm.

It should be noted that according to /ref. 5/ the geometry $\varphi = 1.26$ was found by a thoroughly study during the material test.

However, if a crack is larger than calculated above, the crack will grow and the crack growth should be inspected within $10^8$ number of cycles and for a turbine with 24 guide vanes and speed 375 RPM, the frequency of the pressure pulsations at the blade outlet will be: $375*24/60 = 150$ Hz and $10^8$ stress cycles will be reached in 7.7 days in continuous operation. However if the speed had been $n = 600$ rpm with the same number of guide vanes $10^8$ cycles would be reached in 4.8 days.

It should be noted that if the periodic inspection is neglected a blade rupture may occur within the next $10^8$ cycles.

It should also be noted that the natural frequency of the high head runners has not been proven to have any influence on the stress amplitudes. However, on high specific speed Francis runners operating at low head the resonant frequency of the runner structure may be triggered by draft tube surges causing problems during operation at low load.
Normally blade cracking occurs in location of highest stress concentration i.e. in the fillets of the welds.

However, in the case illustrated in fig 14, the crack did not start in the fillet of weld material, but where the increasing thickness of the blade towards the weld started.

At the high head turbine at Svartisen in Norway a crack in a runner blade was detected. The problem has been solved but it is interesting to see that the crack started on the blade just at the point where the filler started and the blade had its original thickness. See fig.14 left. Svatisen is operating at \( H_n = 540 \) m net head and is a typical high head Francis turbine.

A Finite Element Analysis to find the blade stresses has been made of the stresses from the static hydraulic load on the other turbine in operation at Svartisen. See fig.14. right.

![Figure 12: Left, cracked blade in the second turbine put in operating Svartisen. The runner is repaired and modifications have been made and no more blade cracking has occurred. Right, FEM analyses of the blade stresses loaded by the static pressure on the blades on the blades. Note the stress concentration on the same location where the fatigue crack started on the other turbine.](image)

It is interesting to know that stress concentrations may occur on the blades at uniform thickness at the outlet edge if the thickness is not gradually increasing towards the band and hub. There will always be a balance between hydraulic performance and safety based on safe stress limits.

It should be noted that modifications have been made at Svartisen and no further cracking have been observed. /Ref.8/

f) **Low head Francis runners**

The pressure pulsations inside the low head Francis runners are different from the low specific runners for high heads. The reason is that the distance between guide vanes and runner inlet is larger and with different distance on the band than on the crown.

Further, the hydraulic forces and the impact from the swirl flow in the draft tube is much stronger specially at part load and the relative variation in operational head is much larger. Because of this, the aim for the design of the so called X-BLADE runners for Three Gorges was aimed for a restricted variation of pressure surges.

Because of the much lower frequency of the pressure surges in low head turbines and the fact that the dynamic load is larger at part load than at full load the inspection and risk for fatal blade cracking is different from high head runners. It should also be emphasized that the influence of the design is of importance and the difference between a pressure balanced runner and a bad runner is significant. In fig.14 two different runners for low head turbines is shown. (Note that the specific speed is different for the two turbines, but dynamic problems driven by pressure oscillations in the draft tube and between the runner blades in resonance with the natural frequencies, have been observed in other turbines with specific speed equal and lower than for the turbine to the left in fig. 14).
The two turbine shown in fig. 15 are of different design and the turbine to the right cannot be operated continuously at part load because of heavy vibrations that will damage the runner.

Strain gauge measurements on the runner blade outlet have been measured by the Norwegian consultant company NORCONSULT and the results are presented in fig 16.

III. **Low Cycle Fatigue of Pressure Carrying Parts**

In Francis turbines the pressurized parts are exposed to fatigue growth caused by weld defects and material defects in casted parts caused by pressurizing – depressurizing during start –stop sequences and water hammer pressure surges and surges from mass oscillations in tunnel systems.

Then the crack arresting ability of materials is very important as expressed by the CRACK TIP OPENING DISPLACEMENT (CTOD). CTOD tests of plates and welds including the HEAT AFFECTED ZONE (HAZ), is described in [ref. 6].

The acceptance criterion for weld defects in pressurized parts should be aimed at a 50 000 start stops sequences and in addition the size of the defect should be big enough to penetrate the plate wall before an unstable rupture occurs, i.e. 3 start stop sequences each day in 50 years.

It is then important to make a thoroughly stress analysis of spiral cases in Francis turbines and manifolds for Pelton turbines to detect the stress concentrations and make inspection each year and carry out repair of growing cracks of sizes approaching critical sizes.

In fig.16 a typical stress concentration in a spiral case of a Francis turbine is shown.
It should also be noted that weld defects in the stay vanes does not give any leakage before rupture and it will always be weld defects in the very thick plates of a stay ring. Then inspection by ultrasonic examination should be made for possible growing crack in the stay ring.

Low cycle fatigue is based on material tests where the Crack Tip Opening Displacement is measured in large test pieces.

In such tests a given crack size will grow for any increase in load normal to the crack and then a new equilibrium is established with a larger crack size with a larger zone of yielded material in front of the crack.

By a further increase the crack will grow slowly until it reaches a size where the test piece collapses and an unstable rupture occurs with infinite speed of the crack growth.

This crack size is defined as the critical size or the CTOD (Crack Tip Opening Displacement Critical) which is defined for the material type and thickness in question.

It should be noted that the CTOD value will be smaller for a high tensile strength material compared with a more ductile material with a lower yield point. Then care should be take when using materials with yield points exceeding 450 MPa even if the ductility for high strength steel has been increased during the last years./ Ref 7/.

**Concluding Remarks**

This report is a summary of the authors experience during his work on design, performance and safety on turbines designed in Norway from 1970 to 1987 working at the turbine manufacturer KVAERNER and during his work as Professor at the Norwegian University of Science and Technology until 2003 and up to present time as consultant.

It is highlighted on the most important factor which is safety in power production in Norway where 100% of the electricity supply comes from hydropower.

In addition Norway is exporting peak power to the continent of Europe for a backing of the growing production from wind mills and in order to reduce the variation in load which will increase the pollution from the dominating thermal production on the continent.

**References Références Referencias**