UNDERGROUND HYDROPOWER PLANTS

Proceedings of the International Conference (Hydropower '87), Oslo, Norway June 22 - 25 1987

Edited by

Einar Broch and Dagfinn K. Lysne

The Norwegian Institute of Technology, Trondheim

STRUCTURAL DESIGN OF "HIGH HEAD FRANCIS TURBINES" (H.F.T.)

A. Bjørnsgaard
Master of Science
Head of research Section

Kvaerner Brug

Norway

Abstract

There are described some features of the development of the main components of H.F.T. at Kvaerner Brug (KB) during the 5 latest decades. The development is marked by the break-through of welding of heavy steel components such as stay rings, spiral casings and covers. This development has requiered more knowledge about material technology.
The fatique problem for these components does not seem to have been more pronounced during the change from cast steel to welded structures. For the runners and pump turbines the precaution due to fatigue have to be taken more into account.
Further the construction of the main components is characterized by the relationship between workers level of wages to material cost. Upper and lower cover are therefore for output up to about 70 MW, made massive instead of welded.

Introduction

If we look at the development of Kvaerner H.F.T. during the latest 6 decades, see fig. 1, we will see that there have been a continuous increase of the head. Depending on the decade, in which the turbine is built, we can tell if it is a H.F.T. or not.

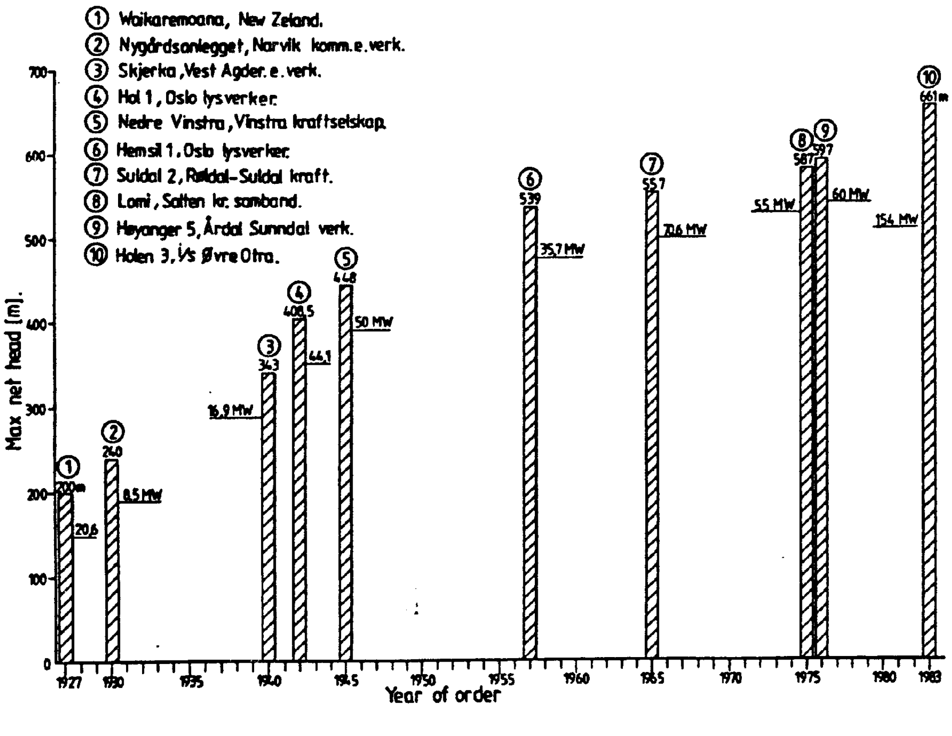


Fig. 1 Development of H.F.T. at KB in the latest 6 decades.

Today a new built H.F.T. has a minimum head of approx. 250 m to be mentioned as H.F.T.

Which structural design criteria are most important in order to obtain high efficiency and high reliability over years?

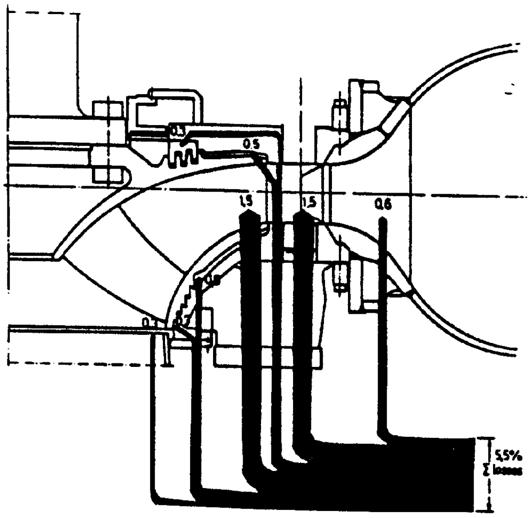


Fig. 2 Distribution of hydraulic losses in a 220 MW H.F.T. at 450 m head, n_q=30.

Two of these criteria are (1) the axial clearance between guide vanes and the guiding faces of the covers (the cheek plates) and 2) the resistance to erosion in the high water velocity region of the turbine. In order to obtain low axial clearance it is necessary with high stiffness of the covers and the stay ring. The intention of small clearance is to guide and accelerate the water through the guide vanes with a minimum of leakage between the vanes and the cheek plates on the covers. For a new turbine with 450 m head we have the approximate distribution of losses as shown in fig. 2.

To get an idea about the dependancy of this clearance we have done some large scale tests on a 50 MW turbine with 340 m head /1/. The result is given in fig. 3. It shows that a clearance of 0,5 mm on a guide vane height of B=500 mm yield an efficiency loss of about 0,4 %.

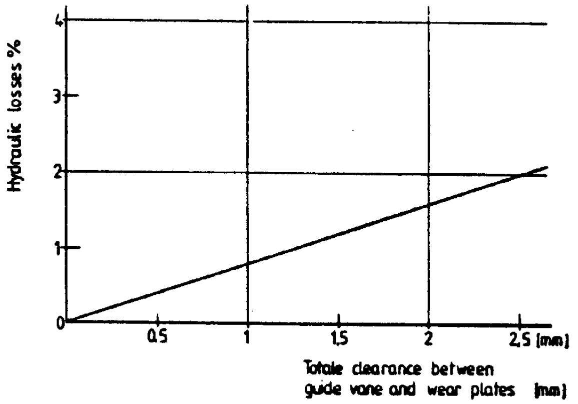
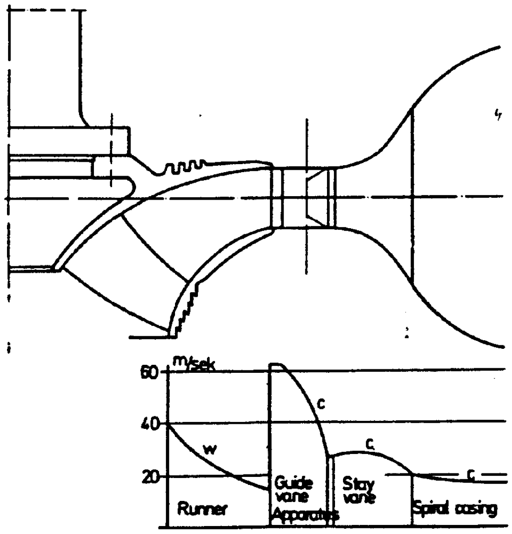


Fig. 3 Approx. hydraulic losses due to the axial clearance in guide vanes n_q=30.



To get an idea of where it is important to use high resistant material, we can from fig. 4 see the velocity distribution through a turbine with 450 m head. The highest velocity (about 62 m/sek) occures at the outlet region of the guide vanes.

Fig. 4 Approx. distribution of the velocity through the turbine H_n=450 m n_q=30.

Since it is the kinetic energy of the water that express the erosion of the surrounding material, these parts of the turbine, especially the cheek plates, will be most exposed to wear.
The wear, however, is low also for 6-700 m head if the water is clean, that means if there is no sand. Especially quartz sand with grain size larger than 0,1 mm diameter is detrimental.

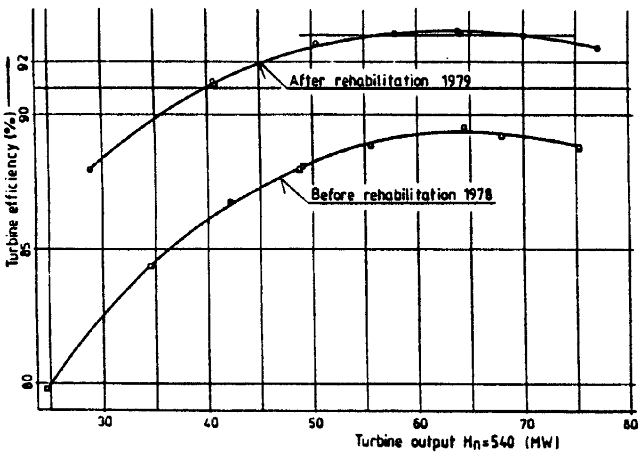


Fig. 5 Efficiency measurement of H.F.T. before and after rehabilitation $P=70$ MW $H_n=545$ m.

To underline this, fig. 5 shows the efficiency of a 70 MW turbine of 545 meter head before and after rehabilitation.
We can see that the difference of is more than 4 % over the whole load range. It is known, however, that there are in operation turbines with spring loaded end seals on the guide vanes which avoid this detrimental leakage /2/. I am not, however, convinced about their resistance to wear from coarse grained quartz sand.
Other components such as the labyrinth seal are also exposed to erosion, but the losses are not so large, see fig. 2.

Development of the main structure during the latest 50 years

There will be of some interest to show the construction of H.F.T. during the latest 50 years. Fig. 6 shows a cross section through a 40 years old H.F.T. of 50 MW output and 420 m head.
We will see that all main components are made of cast steel or cast iron. The complicated cover design serves different purposes. The cover is split in order to obtain balance on the covers and therewith obtain a reduction of the axial clearance against the guide vanes.

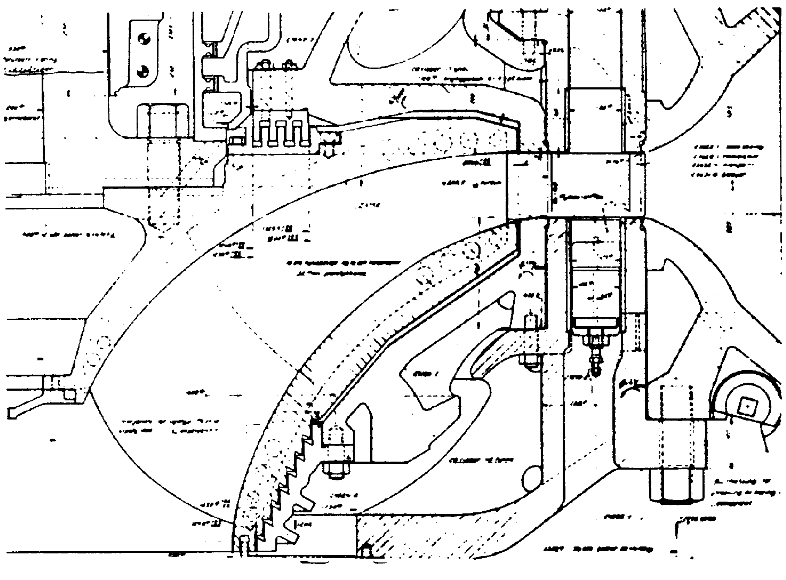


Fig. 6 Cross section through a 40 years old H.F.T. $P=50$ MW, $H_n=420$ m.

In earlier decades it was usual to line the guiding faces on the covers with 18-8 CrNi stainless steel.
This deposit has a hardness of 160-230 HV. In the latest decade it is developed weldable stainless steel with hardness about 300-400 HV. This lining is much more resistant to sand erosion.
It was also important to reduce the weight and thickness of the covers. Forty years ago the ratio workers wages/steel prices was less than 20 % of the corresponding 1987-ratio (refered to western countries.)
The German cast steel standard DIN 1681 from 1929 was the basis for the corresponding Norwegian Standard Sst 45.81 from 1937, with requirements to toughness, which was not common for steel plates at that time.

Since the use of Si killed steel plates first started about 1950 and the use of aluminium as fine grain element in the latest of the 50th at Kvaerner Brug, structures of cast Al-fine grained threated steel were superior to welded structures.*

Up to 1960 the repair welding of cast steel components was an important part of the heavy welding.

With the delivery of the 4 units of 110 MW at 377 m net head for Tokke I turbines in 1960, the use of cast steel culiminated in water turbine construction from Kvaerner Brug, see fig. 7.

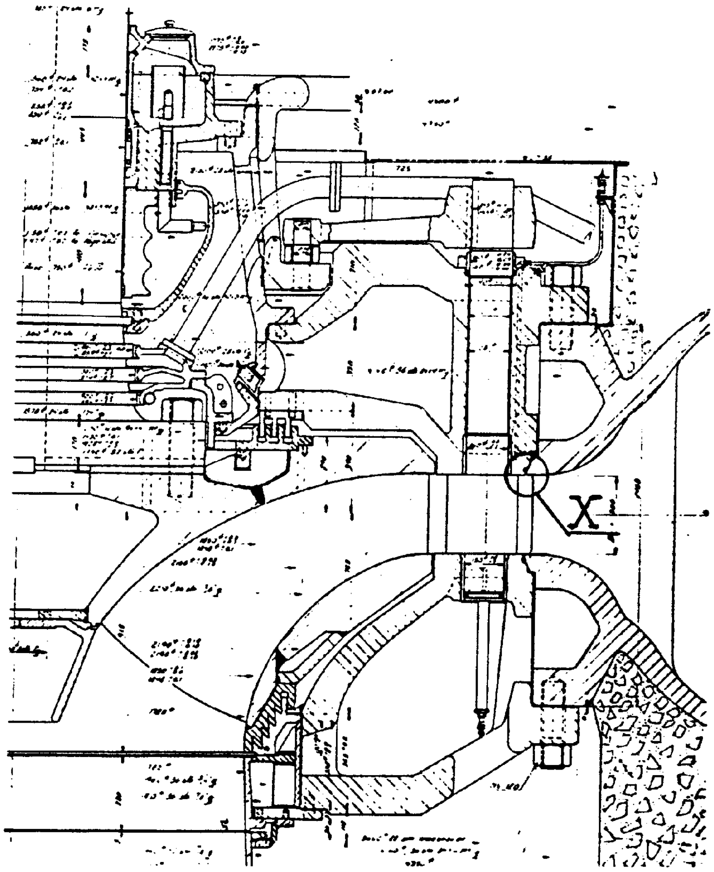


Fig. 7 Cross section through a 30 years old H.F.T.
$P=110$ MW, $H_n=377$ m.

---

* Our earlier General Manager at the Water Power Division Henrik Christie said: "One of the most important developments in the field of material technology for water turbines between 1930 and 1960 was the use of excellent weldable cast steel."

The main components of these turbines were cast except the runner which was made of vanes welded to the cast crown and band.
These runners and the runners for the Hemsil 1 power plant with max head of 540 m were the first H.F.T. runners which were welded at Kvaerner Brug. The region which is exposed to erosion, is lined with stainless 18-8 CrNi steel. The outlet region of the runner vanes is, however, carried out in 13-1 CrNi stainless steel. Our first runners in 100 % stainless steel (16-5 CrNi steel) were to the 4 units of 165 MW turbines for Tonstad mentioned below. These runners were welded in the same way as the above mentioned for Tokke I.

The development of welded spiral casings and stay rings at KB

The first spiral casings with more than 400 m head welded at Kvaerner Brug, were the 4 spiral casings for the Tonstad Power Station. These turbines were delivered during the years 1967-71. *

The stay ring was split into 4 parts and each part was monolitic cast with integrated flanges in quality Sst 45.3. The shell of the spiral casing was welded in steel plate quality TTStE29 (fine grained steel). The max. shell thickness was 50 mm.

15-20 years ago I was asked from a welding engineer why we were using fine grain steel plates with toughness requierment at -20°C or lower.
The following tells us why fine grain steel with such requirements was entitled. In /3/ it is mentioned that in the middle of the 50th there was on the Continent welded a bifurcation for a penstock in the quality DIN MII (C-Mn steel without fine grain elements). Normally this steel has no toughness requirement at lower temperatur than ambient (20°C).
During the load rejection at the commission test the bifurcation cracked and disabled. The bifurcation was rebuilt, but now with fine grain steel.

The stay ring is beside the runner the most critical component of the turbine. It must widthstand all the axial load from the upper and lower cover plus the load on the stay ring and spiral casing limited to the centerline of the torus. On our largest H.F.T. this axial load can be 250-300 MN.

---

* As far as we know the output of 165 000 kW made these turbines the largest in the world with head more than 400 m.

How to balance as much as possible of the vertical forces on the stay ring, is shown in fig. 7. It is our practice to transfere all the vertical load from the lower cover to the stay ring. That means that no axial load is taken up on the lower cover outside the bolt connection to the stay ring. Therefore it is possible to calculate the strength of the stay ring with the guide vane apparatus centerline as symmetry axis.

The monocast stay rings functioned well, but they had several drawbacks because of large regions with low stress level ("proud flesh"). With low serial numbers the model cost is also a considerable part of the total cost.
The advantage of a monocast stay ring is that it is easy to obtain a more successive transition of the cross sections, analogeous to the root shape of a tree. This is beneficial from the fatigue point of view. For pump turbines with extremely high number of start and stop cycles, this geometri is entitled, but not for an ordinary Francis turbine with start/stop cycles about 10^4.

Due to the mentioned drawbacks of cast stay rings we started to weld stay rings in 1961. In 1963-64 we were able to weld stay vanes with thickness up to 85 mm. All that time there were large problems with lamellar tearing, and the stay ring itself was cast. Nor did we have sufficient knowledge about hydrogen induced cracing.*

The main welding connection was the T-joint between the stay vanes and the stay rings, see fig. 8. We started with a K-joint which caused some problem with the root layer. After having gained practical experience we found that using a 90 % V-joint made it easier to avoid cracks. Some years later english and japaneese investigations /4/ stated empirically and theoretically why this V-joint was easier to weld without cracks. At that time, however, problems with hydrogen induced cracking at Kvaerner Brug were almost ended.

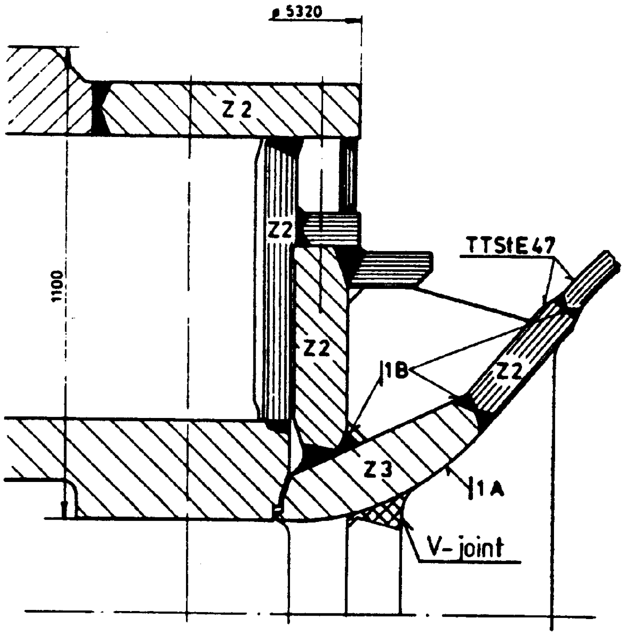


Fig. 8 Detail of stay ring and head cover for the Kvilldal H.F.T. P=315 MW, H_n=520 m.

---

* It is known that overseas a stay ring for a pump turbine in high strength T1 steel in the beginning of the 70th cracked and disabled during proof testing before the normal pressure was reached. The stay vane thickness was 150 mm.

Beside the hydrogen induced cracking, lamellar tearing in plates was the other main problem in welded structures until the middle of the 70th. This problem was in my opinion the main reason why fine grain cast steel with more isotropic material properties could compete with welded structures in water turbines.

With the breakthrough of Z-steels from the middle of the 70th, the welding of thick rolled plates was, in my opinion, the largest technological development in this field since the breakthrough of fine grain steels 20 years earlier.

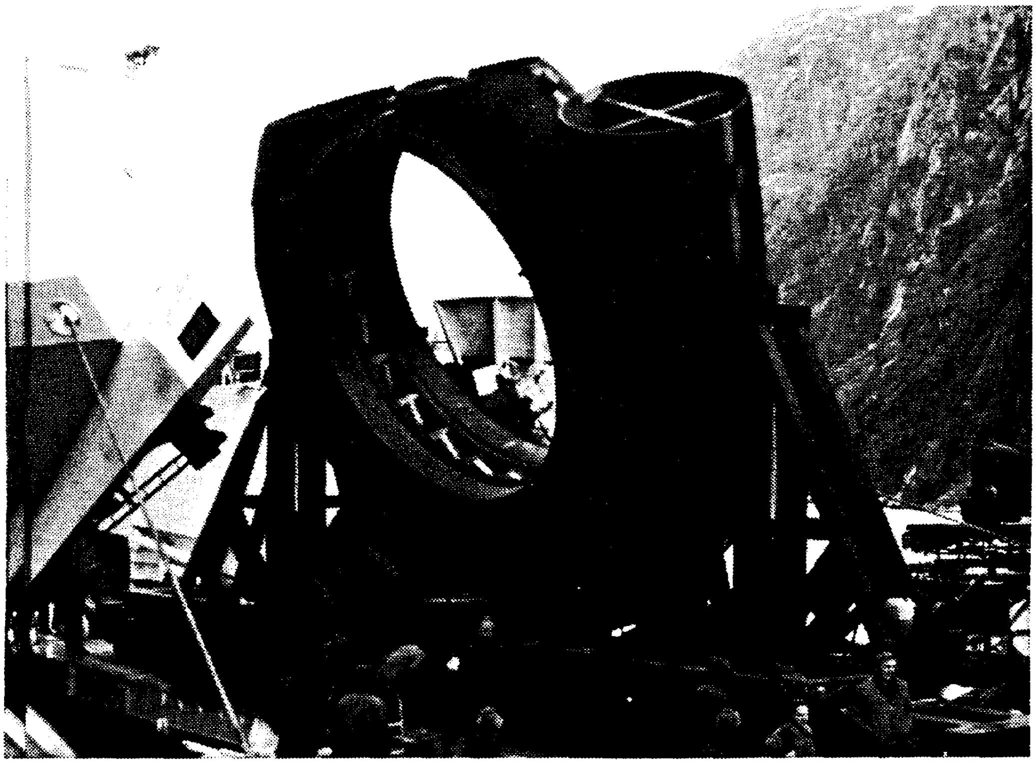
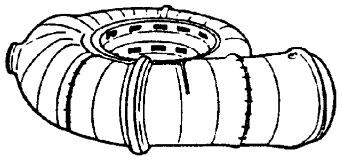


Fig. 9 The 100 tons Spiral casing for Kvilldal with min. diameter 8,2 m during transport through the 4,8 m wide front gate of the vessel "Elektron".

A typical application of Z-steels is shown in the welded stay ring and head cover of Kvilldal, see fig. 8. Most of this fine grain steels (e.g. TStE355 DIN 17102) have requirements concerning through thickness ductility, see details Z2 and Z3 (Stahl-Eisen Lieferbedingungen 96). That means the material has a minimum average value of contraction in the thickness direction of respectively 25 and 35 %. The quality class of the weld shown in fig. 8, is KB1. This corresponds nearly to the ASME SECTION VIII DIV.2. The quality of the material in the spiral casing is DIN 17102 TStE460. After welding in the workshop the spiral casing is stress relieved at 550°C and pressure tested with 9,1 MPa at site.

Fig. 9 shows a spiral casing with min. 8,2 m diameter transported through the 4,8 m wide port of the RoRo vessel "Elektron". The vehicle is equipped with a system of hydraulic cylinders in order to change the position of the spiral casing from vertical to horizontal, and vice versa. Therewith it is also possible to pass through any "needle eye" with low height, e.g. tunnels.



	Spiral 1	Spiral 2	Spiral 3
Design year	1957	1965	1976
Turbineoutput MW	110	165	315
Net weight Tons	100	100	100
Weightfactor			
$\left[\frac{D_{sn}}{D_{s1}}\right]^2 \frac{D_{in}}{D_{i1}}$	1.0	1.46	2.58
Weight refered to 1957-design Tons	100	146	258

Fig. 10 Development of Spiral Casings between 1957-76.

To explain the development of H.F.T. structures at KB during the 20 year period 1957-1976, we should look at 3 (I, II and III) 100 tons spiral casings with max. head 400-550 m. The design year was 1957, 1965 and 1976 respectively. Since the turbines were approximately homologous (geometric similar) the spiral casing of turbine III of 1957 technology would have weighed 250 tons compared with one manufacted in 1977. See fig. 10.

Fabricating the stay ring in one piece is advantageous since we avoid the complicate T-joint between the circumferencial O-ring seal and the seal on the stay ring flanges, see detail X fig. 7.

The size limitation of the spiral casing is decided by "needle eyes" on the transport route to the plant. Both the transport equipment (trucks, vessels, train) as well as tunnel openings have to be considered.

The splitting of the spiral casing and the size of the components depend of course on the location of the power plant. If the plant is near to the sea, for instance at the bottom of a fjord, it is obvious that the spiral casing can be made in one piece. It is of course necessary to plan the whole transport, including the access tunnel in detail, and well in advance.

Intricate transport routes might require a "test transport" with a full scale template of the transport profile.

What is the upper limit of the strength of the spiral casing?

There are of little intention using yield strength higher than 355 MPa for the upper and lower cover as well for the stay ring. This is due to the stiffness requirements of these components.

For the spiral casing, however, there are other criterias limiting the upper level of the working stress.

In /5/ it is shown by means of fracture mechanics and fatigue assessments that the upper limit for the yield strength in order to obtain: "Leakage before fracture" is about 450 MPa. This is further based on the assumption that the number of starts and stops is max. $5x10^4$ (3 times every day in 333 days every year in 50 years). A further requirement is that an undetected crack caused by fabrication should have penetrated through the wall thickness and caused water leakage before an unstable fracture occure.

The risk of an unstable fracture due to flaws not detected during the investigation of the welded structure by means of NDT-methodes is in our opinion, minimal. This is due to the fact that the spiral casing and the covers are proof tested at a pressure of 1.5 times the max. normal working pressure. To quantify the benefical effect of proof testing the Welding Institute is for the time being working at a research project concerning this question (Influence of proof testing component reliability) /6/.

Fabrication of upper and lower cover for H.F.T.

In welded covers (see fig. 8) the shear and bending deformation of the ribs are an essential part of the cover angular and hence the axial deformation of the covers. The reason for using covers with high stiffness is as mentioned earlier to obtain small leakage between covers and guide vanes. The deformation (the black area) of the head cover (fig. 11) is calculated by means of F.E. analysis which gives an axial deformation at the runner inlet of 0,57 mm and a radial deformation of 0,28 mm. This indicate that even the 75 mm thickness of each 24 ribs could have been larger.
Especially for smaller H.F.T. we have in many years found it advantageous to use a massive cover construction. Deformation measurments of these covers have shown a good correspondence of the "by hand calculated" axial and radial deformation.

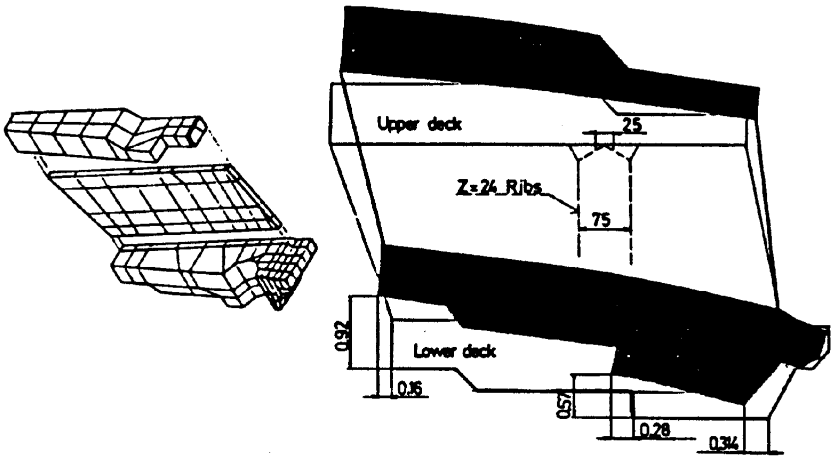


Fig. 11 Head cover for a 220 MW H.F.T. H_n=450 m, Deformation calculated by FE analysis.

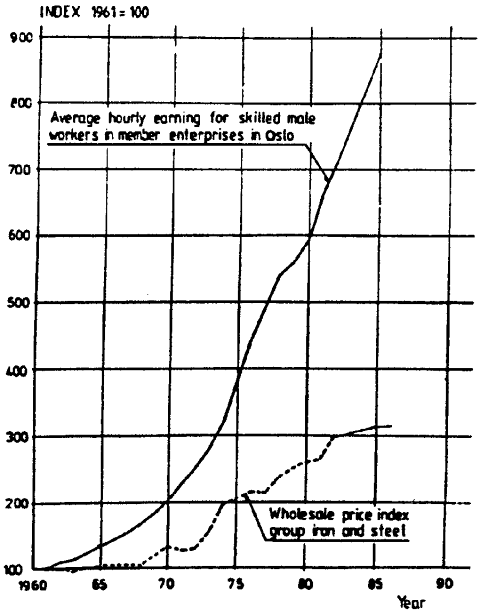


Fig. 12 Development of material cost and workers earning in Norway during the latest 25 years.

The development of materiel cost and workers earning in Norway during the latest 25 years (fig. 12) proves that the materials have been contineous cheaper compared to the wages. For turbine components such as covers where it is important to obtain high stiffness, we have therefore during the latest decade changed to massive construction of the covers for H.F.T. up to about 70 MW output. These covers are heavier than the "double decker" covers if they both have the same axial deformation in spite of the reduced height.
The stress level is, however, higher at the same level of axial deformation as for the welded "double decker" covers.

Due to this large thickness, up to 400 mm, there are mechanical properties such as ductility and toughness that have to be taken into account. Therefore we have to use a fracture mechanic assessment. As the thickness and the stress level increase, the requirement of fracture toughness has to be increased.

Simplified we may say that the material seems to be more brittle when it is thicker (the 3 dimensional stress condition is blocking the deformation). To state the value of the fracture toughness K_{1C} at such large thickness is however very expensive.

In /7/ it is referred to a french research work that yields the correlation between the temperature TK27 (the temperature at which the impact value ISO V is 27 Joule and the temperature $TK_{IC}100$ where $K_{IC}=100$ MPa $(m)^{\frac{1}{2}}$.
This brittle fracture criterion means that the cheaper Charpy test have to be done in dependancy of the material thickness at severeal decades of centigrades lower than the working temperature of the turbine.
All main components (spiral casing and covers) of Francis turbines from Kvaerner Brug with normal working pressure above about 1,5 MPa (about 150 m) have to be pressure tested.

When using massive covers the proof testing is more critical concerning brittle or instable fracture compared with a correponding welded "double decker" cover with plate thickness about 100 mm.
Figure 13 shows a massive cover indicated with solid lines and a corresponding welded "double decker" indicated with drawned lines.

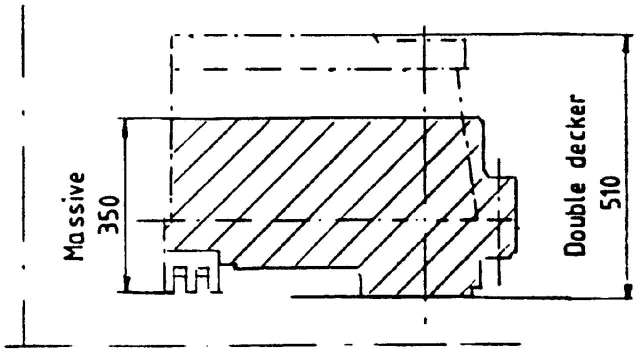


Fig. 13 Diagram showing the geometry of a massive head covers for P=60 MW H_n=485 m.

Short description of the main components of the H.F.T. from KB

Fig. 14 shows a cross section of H.F.T. with max output of 370 MW. *

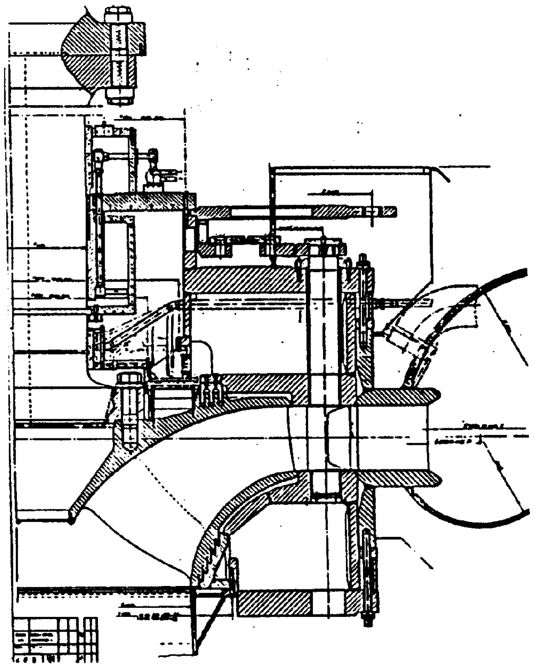


Fig. 14 Sectional drawing of H.F.T. Tonstad 5, Norway. P_{max}=370 MW, H_n=430 m, n=300 (year of order 1985).

The stay ring has a rather simple geometry as it has been carried out in the latest 8 years by KB. The Z-steel with through thichness toughness and ductility requierments is used where it is entitled, cfr. fig. 8. All component are welded except the runner crown and band plus the main shaft and guide vanes which are forged.
To optimize the shape and thickness transition of the different critical parts of the stay ring, it is absolute necessary to use comphrehensive F.E.-analysis.

---

* As far as we know this is the most powerful water turbine with head more than 400 m.

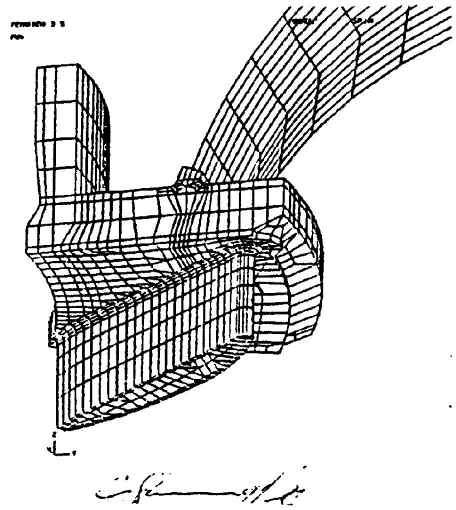


Fig. 15 Structural model of a stay ring for F.E. analysis.

Fig. 15 shows a structural model of a stay ring for F.E.-analysis. An analysis like this is so large that it is only possible to carry out for a few main types.
The upper and lower covers are welded of fine grain steel of type TStE 355 with thickness up to 250 mm. All the faces between the stay ring and the labyrinth rings are lined with stainless steel. The guide vanes are made of forged 13-4 CrNi steel. All the bearings of the guide apparatus and regulating ring are lubricate free and made of PFTE-bronze-lead with steel backing. The runners are welded in 16-5 CrNi stainless or monolitic cast 13-4 CrNi stainless steel. According to strength and ductility these two material qualities are almost equal. Against cavitation erosion our latest experiences shows that the 16-5 CrNi has the highest resistance.

The runner has on its upper side a pump arrangement at the inner side of the runner seal which give enough pressure to the labyrinth leakage flow for the cooling water system for both generator and transformater /8/. This pump arrangement also keeps the shaft seal dry which again yields free admittance for air to the runner centre. The torque between runner and shaft is transfered by a pure friction connection by means of high preloaded high strength bolts. During the mounting and the demounting procedure it is therefore no problem with seizure between the bolts and surrounding stainless steel.

Concluding remarks

Runner erosion due to cavition is not mentioned since it has not been a problem at KB.
To be seen from fig. 1 the H.F.T. has during the latest 60 years taken much of the head region that earlier belonged to the Pelton turbine. It seems to me that this trend also will continue into the next century, though not so fast, as the advantage in higher efficiency is smaller compared with that for lower heads. This is due to thicker vanes in the stay ring, guide apparatus and runner.
On the other hand it is possible to use multistage Francis turbines with more optimal specific speeds. If though the water over years contains large quantities of coarse grain quartz sand, the Pelton turbine will show its legitimate due to easier maintenance work.

/1/ Arthur Teigland
Kvaerner Brug A/S
CLEARANCE LOSSES IN THE GUIDE VANE SYSTFM OF
LOW SPECIFIC
SPEED FRANCIS TURBINES. (High pressure turbines.)
IAHR - Symposium Colorado June 1978.

/2/ Mechanical Engineering Publication (MEP)
London 1985
"The Dinorwing Power Station" p. 51.

/3/ K. Wellinger und P. Gimmel, Stutgart
ROHRSCHADEN IN EINEM WASSERKRAFTWERK
Brennst.-Wärme-kraft 17 (1965) nr. 9, September.

/4/ T.G. Davey, BSc (Eng.) abd P.H.M. Hart, BSc (Eng.), ARSM, MIM, MWeldl: "The influence of weld preparation and restraint on the risk of root pass HAZ cracking in butt welds in a C: Mn:Si:Nb Steel".
The Welding Institute
Research Report 135/1981.

/5/ Stud.techn. Jens R. Davidsen, avd. VI
Institutt for mekanisk teknologi
Norges Tekniske Høgskole
Hovedoppgave, høsten 1975:
Valg av material til Ringledning for Peltonturbin.

/6/ The Welding Institute:
CP/FRA/2021-2
March 1986
DJS/SJG
INFLUENCE OF PROOF TESTING ON COMPONENT RELIABILITY
For: A Group of Sponsors.

/7/ G. Sanz: "Essai de mise au point d'une methode quantitative de choix des qualites d'aciers vis a vis du risque de rupture fragile",
IRSID - Inst. De Recherches de la Siderurgie Francaise, Feb. 1980.

/8/ H. Brekke
Kvaerner Brug A/S, Oslo
A discussion of Pelton Turbines versurs Francis Turbines for high head plants.
IAHR Symposium Colorado June 1978.