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HIGH HEAD LARGE CAPACITY PUMP-TURBINE

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I. INTRODUCTION

Pumped storage power plants constructed to assure electric power at peak loads and also reserve power have gradually tended to have higher heads and larger capacities from the viewpoint of economy. Generally, high-head, large-capacity pump-turbines have not only high static stress levels because of the high water pressure and high rotating speed but also high repeated stress levels because of frequent changes in operating conditions and large pressure fluctuations.

When the blade thicknesses of such parts as the runner, wicket gate and stay vane are increased because of mechanical requirements, there is a drop in efficiency due to flow separation, pressure fluctuations, noise and cavitation.

The development of a high head pump-turbine with good performance and high reliability based on compromise between mechanical and hydraulic requirements has been almost impossible with the traditional trial and error techniques, based on mainly model tests.

Therefore, for several years, Fuji Electric has been establishing techniques for both static and dynamic stress analysis by means of the finite element method and has developed computer programs for flow analysis and hydraulic design. Investigations from various aspects have been performed on hydraulic performance by means of computer software and model tests. With improved production techniques, a high head large capacity pump-turbine with good performance and high reliability has been successfully developed.

This article describes Fuji Electric high-head pump-turbine technology using the pump-turbine manufactured for the Chongpyong Power Station, Korea Electric Co., as an example.

II. HYDRAULIC DESIGN AND PERFORMANCE OF THE PUMP-TURBINE

1. Hydraulic Design by Computer

In the case of high-head pump-turbine, the vane thickness increases for reasons of strength and therefore, the design of the vane profile of stay vane and wicket gate as well as runner becomes very important. The number of design parameters which control the pump-turbine characteristics is very large in consideration of both the generating and pumping cycles, and investigations of all combinations of design parameters in model tests are impossible. There are limitations on the development of an optimum pump-turbine using only classical design procedures.

Therefore, Fuji Electric has developed various computer programs including a runner design computer program (FURDES), a runner flow analysis computer program (RUNFLAS) and a flow analysis computer program for double circular cascades combining the stay vanes and wicket gates (STAFLAS). As can be seen in *Fig.1*, it has been possible to conduct integrated investigations by computer of a vast number of combinations of pump-turbine design parameters by means of a systematic combination of these programs.

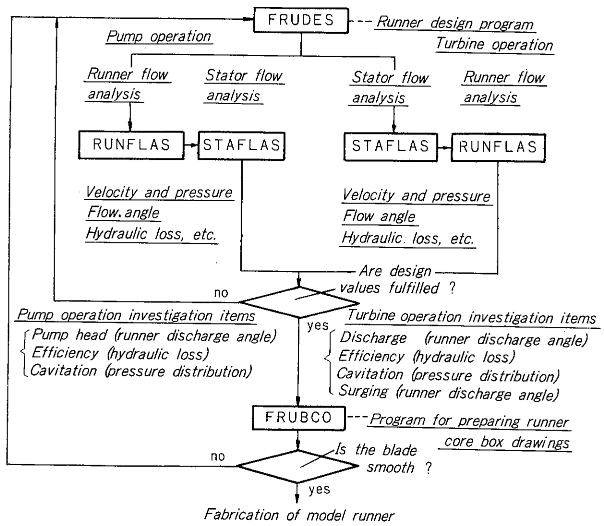


Fig. 1 Hydraulic design computer program of pump-turbine

Calculation results by means of these flow analysis computer programs show good agreement with the model tests results with respect to the following points and it has been shown that it is possible to estimate pump-turbine hydraulic characteristics by means of numerical experiments:

1) Estimation of incipient cavitation coefficient of pump operation

2) Estimation of pump input power
3) Estimation of pump head-discharge characteristics
4) Estimation of pressure fluctuation region due to flow separation at runner inlet during turbine operation
5) Estimation of pressure surge in draft tube at partial load
6) Estimation of turbine discharge and output

Figs. 2 and *3* show results of pump-turbine flow analysis.

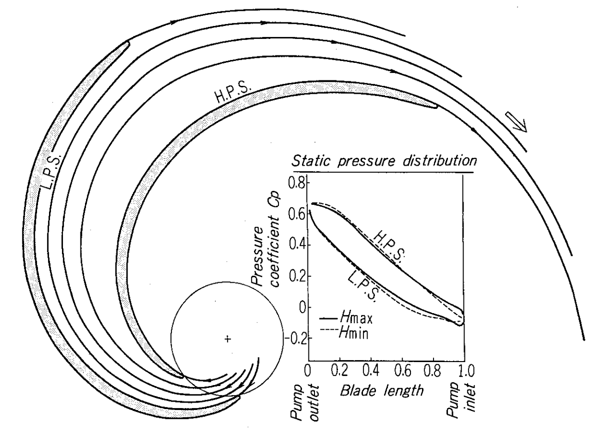


Fig. 2 **Runner flow analysis (pump flow)**

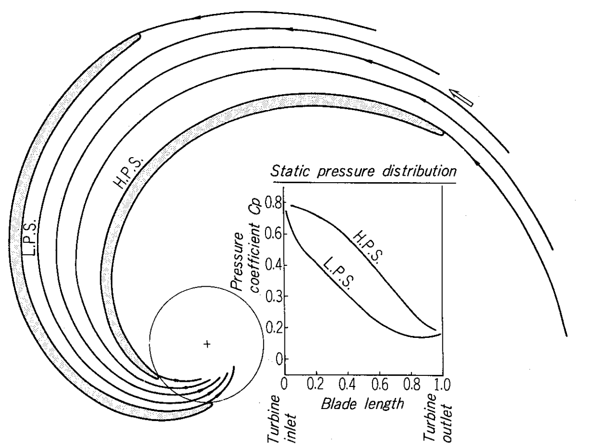


Fig. 3 **Runner flow analysis (turbine flow)**

In the development of a pump-turbine, successive investigations are performed for many combinations of pump-turbine design parameters using a computer and the optimum design profile can be selected out of these. A model pump-turbine is then manufactured on the basis of this optimum design. Model tests are carried out in our hydraulic laboratory and performance is confirmed. In this way, it becomes unnecessary to manufacture several models and perform comparative tests as in the past, and it has been possible to speed up the development.

2. Pump-turbine Performance

The model pump-turbine designed by repeated numerical experiments using a computer has been further improved by means of detailed model tests. As a result, a 500m class pump-turbine with excellent efficiency, cavitation and pressure fluctuation characteristics has been developed for the Chongpyong Power Station. Typical model test results are outlined hereinafter.

1) Specifications of pump-turbine

Turbine			
Effective head (m)	473.0	452.0	437.5
Turbine output (MW)	206.0	206.0	196.0
Rotating speed (rpm)		450	
Specific speed (m·kW)		98	
Pump			
Head (m)	498.5		474.0
Discharge (m^3/sec)	34.3		39.0
Maximum pump input (MW)		220	
Specific speed (m-m^3/sec)		27.4	
Maximum runaway speed (rpm)		660	
Maximum speed rise ratio (%)		40	
Maximum pressure rise (m)		730	
Suction head (m)		−52	

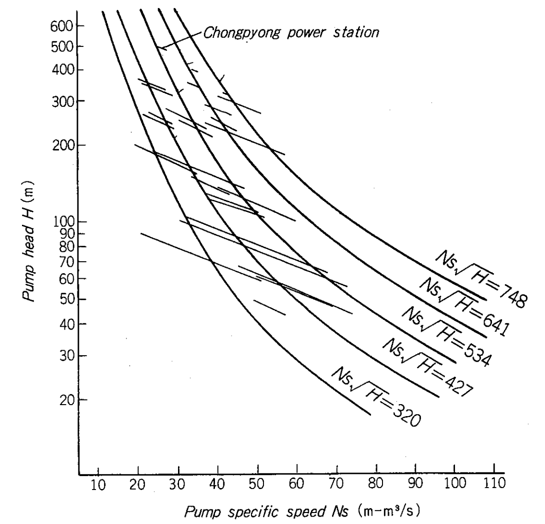


Fig. 4 **Relation between pump specific speed and pump head**

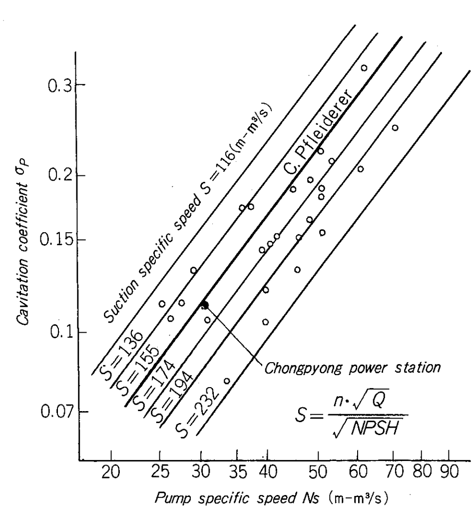


Fig. 5 **Relation between cavitation coefficient and pump specific speed**

Fig. 4 shows the relation between the pump specific speed and head of the Chongpyong Power Station. *Fig. 5* shows the relation between the cavitation coefficient at the maximum pump head and the pump specific speed.

2) Model pump-turbine and model test

With the recent excellent model test rigs, measuring techniques and analysis techniques, it has become possible to simulate almost all hydraulic phenomena of prototype pump-turbine using large scale models with water passages completely homologous with those of prototype.

The following is an outline of the model tests.

(1) By using a large scale model with a runner diameter of 620mm, it has become possible to produce models which completely homologous with the prototype with respect to not only the runner, wicket gate, stay vane, spiral case and draft tube but also with respect to the such items as the inner shapes of the head cover, discharge ring, runner band drain holes, thrust relief pipes and thrust balancing pipes which have subtle effects on the efficiency and hydraulic thrust.

Fig. 6 shows the Hydraulic Laboratory model pump-turbine.

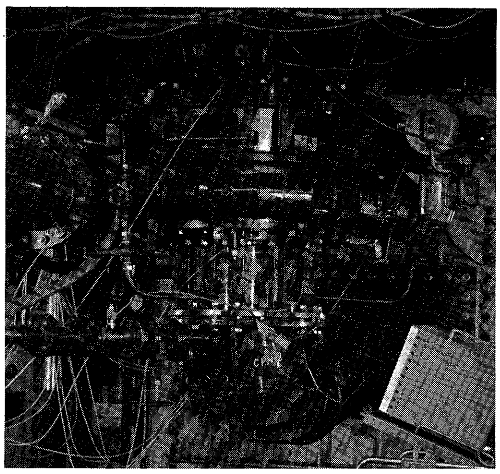


Fig. 6 Testing rig of model pump-turbine

(2) In order to know beforehand not only the cavitation phenomena but also the occurrence of flow separation hydraulic instabilities at various operating conditions far apart from the design points, and to take counter measures against them, it is essential to measure pressure fluctuations and noise and to visualize the flow pattern.

Therefore, the runner band, bottom ring and discharge ring were made of transparent plastic so that it was possible to observe the flow patterns both around the runner vane and around the wicket gate and stay vane.

Fig. 7 shows the model pump-turbine during testing.

(3) Because of the thickness of the stay vane and wicket gate in high head pump-turbines, the hydraulic losses increase, and their profiles design become important similar to the runner blade design.

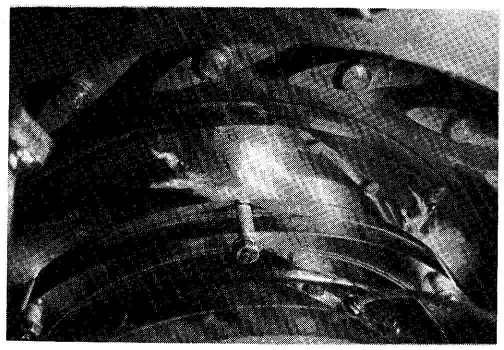


Fig. 7 Model pump-turbine

Therefore, comparative tests were performed on two types of stay vane profiles, and the profile which gave a high turbine efficiency and lower pressure fluctuations in the smaller pump discharge region was adopted.

(4) The wicket gate plays the role of providing the flow circulation required by the runner during generating operation, but the hydraulic loss becomes large because of thickness especially at the smaller opening of the wicket gate. The flow circulation then becomes insufficient, causing partial load efficiency drops, hydraulic vibrations due to flow separation from the runner vane inlet and the possible instability phenomena near no load at low heads.

Comparative tests were performed on two wicket gate profile and the one with better S-shaped characteristics in the N_{11} - Q_{11} curves was chosen. *Fig. 8* shows the relation between the wicket gate profile and S-shaped characteristics.

(5) Since the circumferencial relative positions of the stay vane and wicket gate also have a major effect on efficiency, many difference circumferencial relative positions were investigated and the positions with the best efficiency and small pressure pulsation were selected.

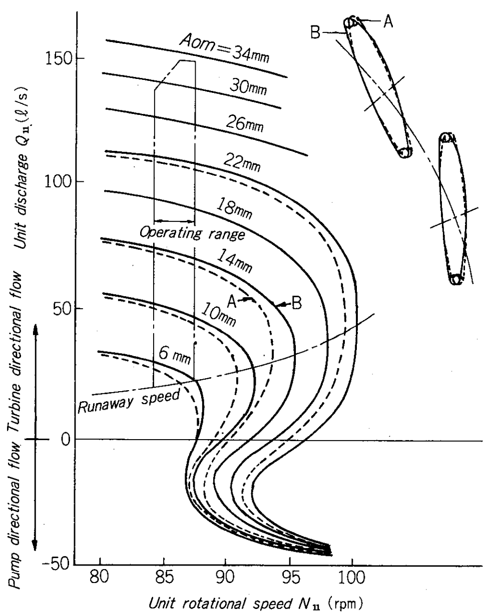


Fig. 8 S-shape performance depending on the wicket gate

(6) Model simulation concerning the blow down of the water level in the draft tube, air exhaust, flooding of the runner and the establishment of priming pressure at the time of pumping start-up was also performed. Studies were performed concerning the relation between the runner band drain holes and the starting torque in the air, and the relation between the air release holes at the runner crown and the period of establishing the priming pressure.

3) Turbine performance

The pump-turbine must operate in as wide operation zone as possible to cope with the rapid load variations in the electric power system. Therefore, the development work of a pump-turbine with low pressure pulsation and improved efficiency at partial load was performed by making improvements in the runner vane inlet profile. *Fig. 9* shows the turbine performance of the prototype pump-turbine converted from the final model.

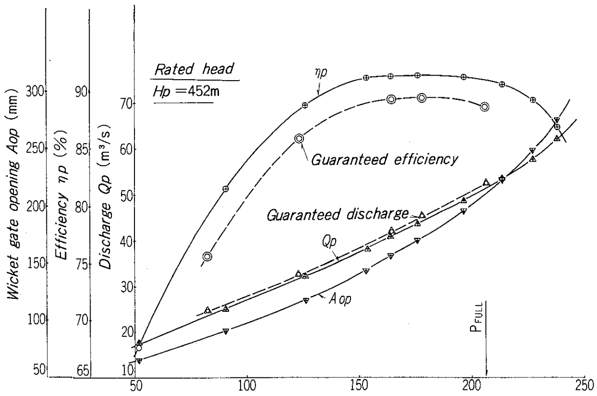


Fig. 9 **Turbine performance of the prototype pump-turbine**

4) Pump performance

The design was such that the maximum pump output was kept approximately equal to the maximum turbine output and that reverse flow region occurring in the small discharge zone was as far away as the normal operating region as possible. A pump-turbine with high efficiency and low pressure fluctuations in the normal operating region was developed. *Fig. 10* shows the prototype pump performance converted from the final model.

Fig. 11 shows the cavitation performance in the pumping cycle. It is evident that there are no cavitation bubbles on both the pressure surface and suction surface the inlet part of the runner blade within the normal operating region.

5) Hydraulic transient phenomena

To estimate the transient operation characteristics, the four-quadrant characteristics were obtained and the pressure fluctuations in each quadrant were measured. The pressure and speed variations during turbine load rejections and pump trip were calculated by means of the hydraulic transient penomena analysis computer program (HYTRAN) based on the characteristics method.

This computer program calculates not only the water pressure of any position of the penstock and tailrace but also calculates the pressure fluctuations of various parts and

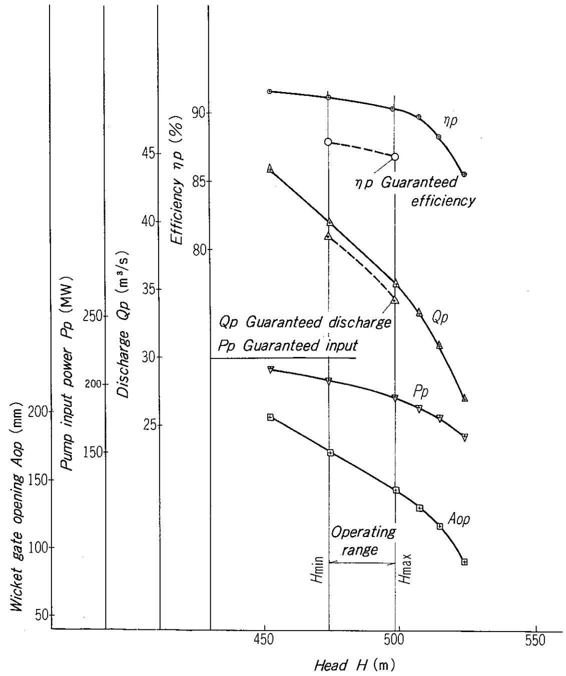


Fig. 10 **Pump performance of prototype pump-turbine**

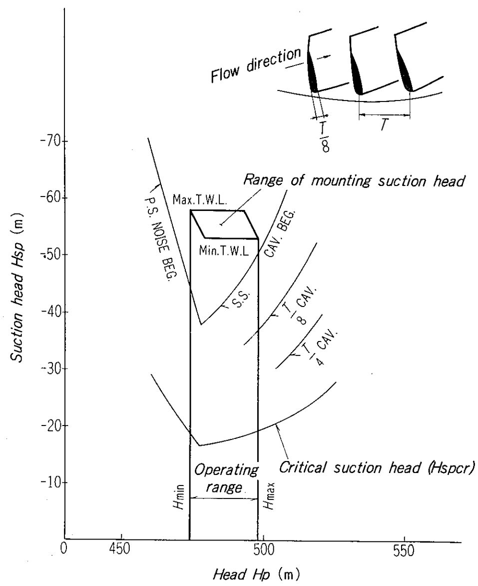
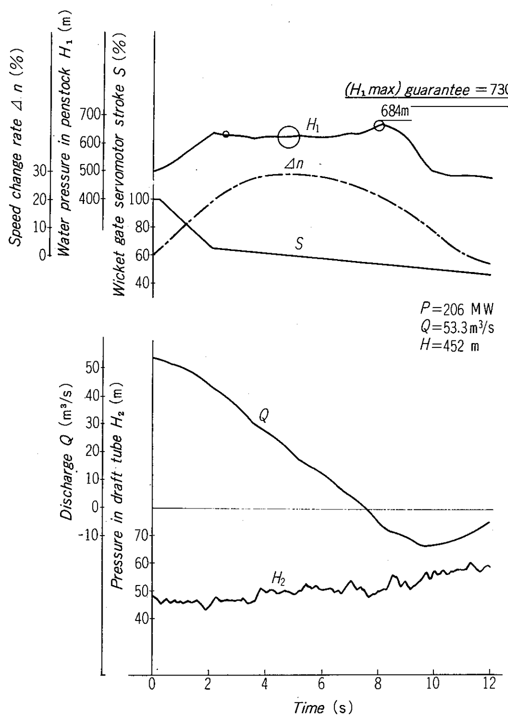


Fig. 11 **Pump cavitation performance of prototype pump-turbine**

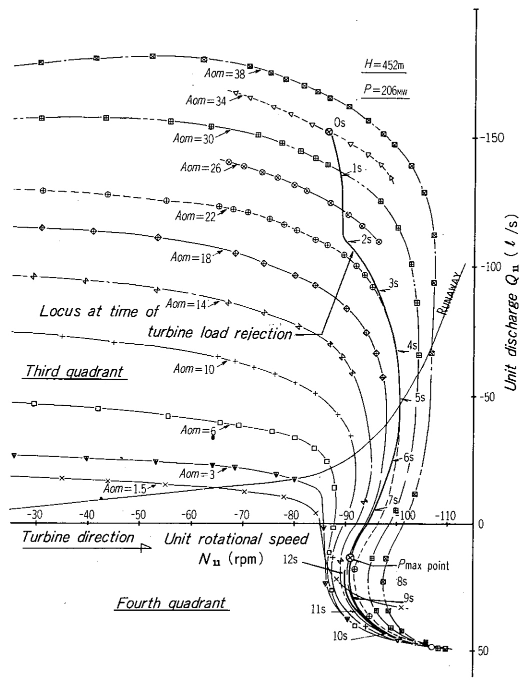
the hydraulic torque of the wicket gates utilizing stored model test data.

Fig. 12 (a) shows the calculated results of the hydraulic transients at the time of turbine load rejection and *Fig. 12* (b) shows the trace of operating point on the four-quadrant characteristics. The calculated results of hydraulic transients at the time of pump trip are shown in *Fig. 13* (a) and *Fig. 13* (b).

In addition to these hydraulic transients at the time of emergency stops, analysis were also performed on transient phenomena, including pressure surges, for turbine and pump starting. Detailed investigations were also conducted

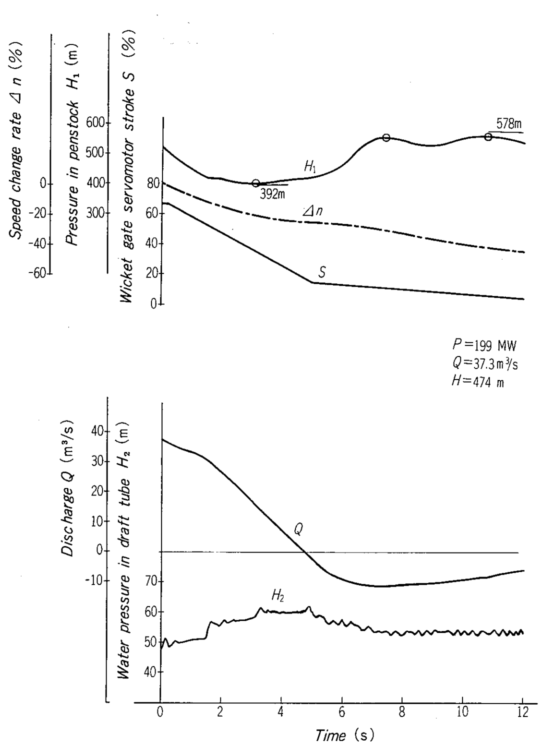


(a) Transient phenomena at time of turbine load rejection

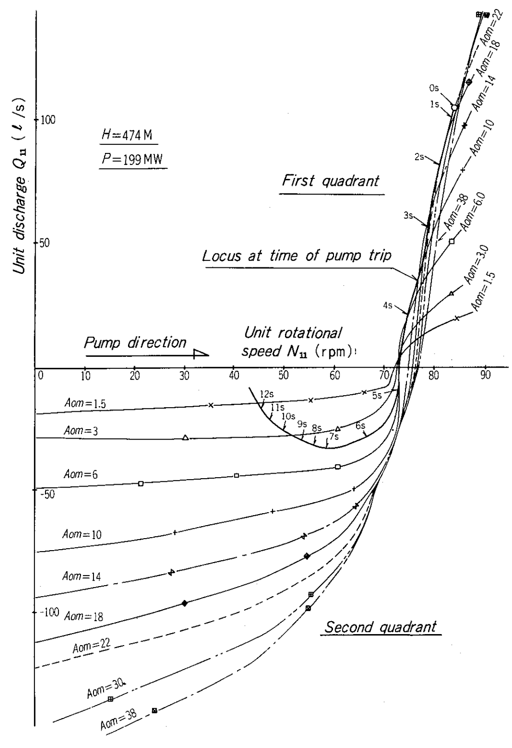


(b) Locus of complete characteristics of turbine load rejection

Fig. 12 **Hydraulic transient phenomena of turbine load rejection**



(a) Transient phenomena at time of pump trip



(b) Locus of complete characteristics of pump trip

Fig. 13 **Hydraulic transient phenomena of pump trip**

to optimize wicket gate opening and closing time.

6) Hydraulic axial thrust

The hydraulic forces acting on the top and bottom surfaces of the runner exceed 5,000 tons and the slight unbalance between these two forces causes lift up of the rotating parts or an excessive downward axial thrust.

The static pressure distribution on the top and bottom surfaces of the runner outside of the runner seal differs according to the differences in the water chamber shape above and below the runner, and the differences of the peripheral monentum of the leakage water and the runner band drain holes.

Since the drainage hole in the discharge ring in particular disturbs the water circulation in the chamber below the

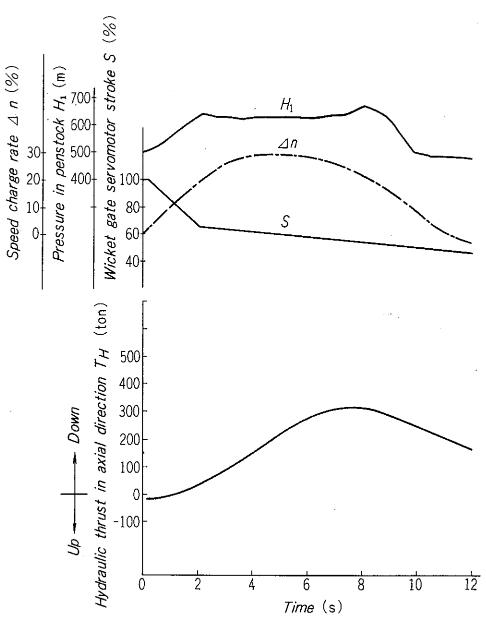


Fig. 14 Hydraulic axial thrust during load rejection

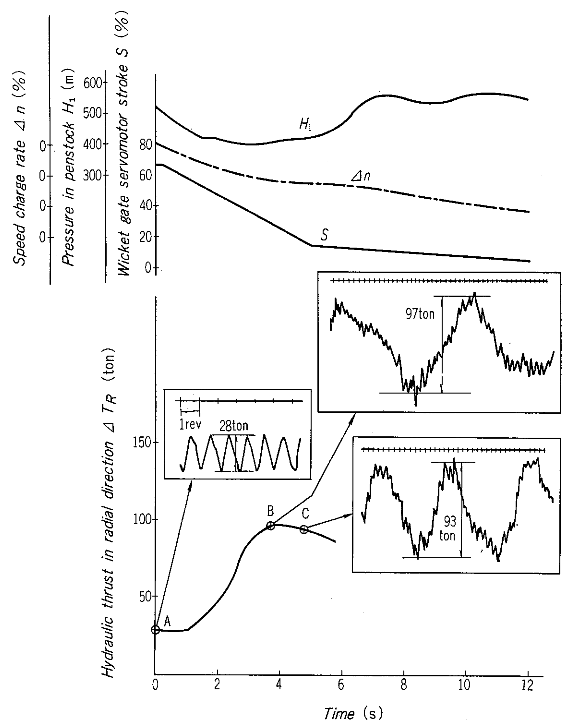


Fig. 16 Hydraulic radial thrust fluctuation during pump trip

runner, the static pressure below the runner becomes higher than that of the above the runner. This may cause lift up of the rotating parts. It was possible to greatly reduce this upward hydraulic thrust by providing a pressure balancing pipe and introducing high pressure water below the runner to the upper side of the runner. *Fig. 14* shows changes in the hydraulic thrust at the time of turbine load rejection.

7) Hydraulic radial thrust

The hydraulic radial thrust arises because of non-uniformity of the static pressure around the runner in the circumferential direction. Therefore, it increases at the time of turbine load rejection or in smaller pump discharge. With the recent progress in measuring techniques, it has become possible to measure the hydraulic radial thrust which is useful in vibration analysis and bearing design of the pump-turbine and generator-motor.

Fig. 15 shows hydraulic radial thrust pulsation during load rejection and *Fig. 16* hydraulic thrust pulsation during pump trip.

8) Hydraulic torque of the wicket gate

Hydraulic torque and torque fluctuations acting on the wicket have been measured under various operating conditions and the results have been used in deciding the correct wicket gate servomotor capacity and wicket gate fatigue calculations.

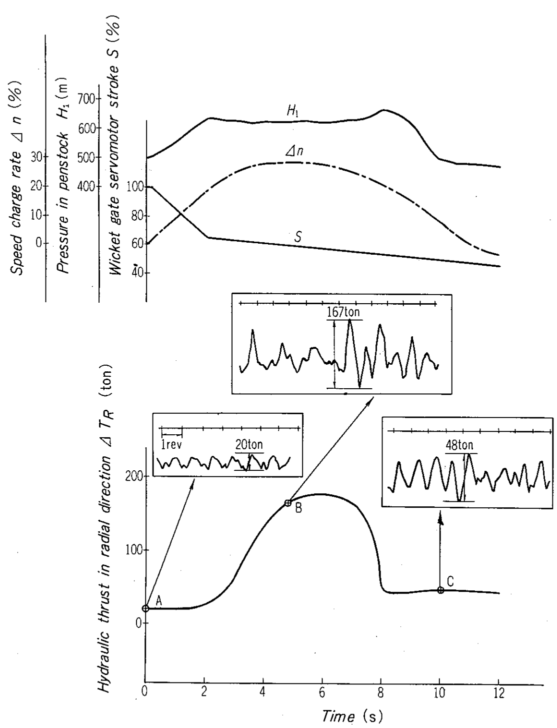


Fig. 15 Hydraulic radial thrust fluctuation during load rejection

III. PUMP TURBINE CONSTRUCTION

The results of extensive research and development concerning Fuji Electric pump-turbine construction will be explained using the Chongpyong Power Station pump turbine as an example. *Fig. 17* shows a section of the pump-turbine.

1. General

The main parts of the pump-turbine are subjected to severe stress because of the high head and high speed, and the following measures were taken at each stage of design and manufacture to improve the reliability.

(1) Stress and deflection of the main parts of the pump-turbine under all operating conditions (generating, pumping, load rejection, runaway speed, pump shut-off and pump

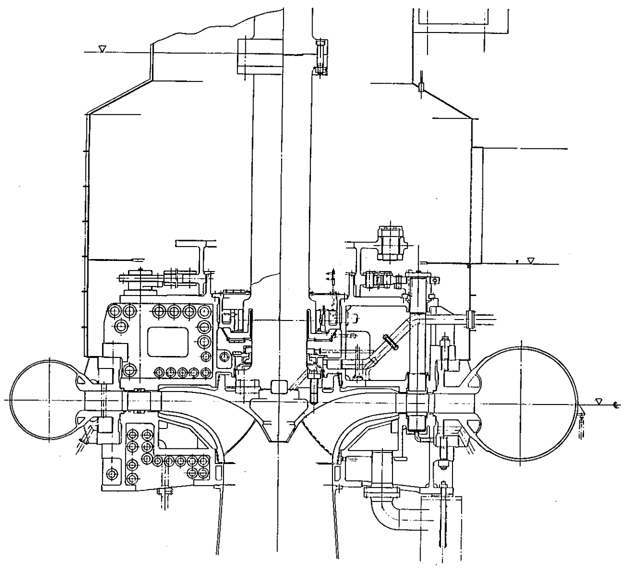


Fig. 17 Distributor section of pump turbine

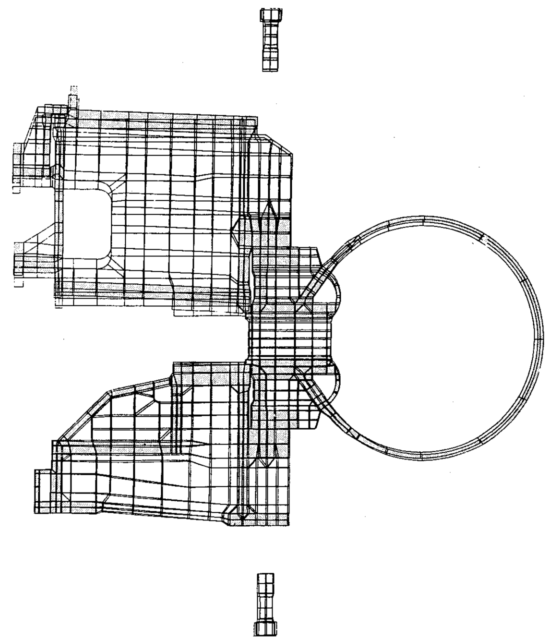


Fig. 18 Deflection of pump turbine assembly

trip) were analyzed by the finite element method. These analysis included not only static analysis by axisymmetric, two and three dimensional computer programs, but also dynamic analysis including wicket gate dynamic analysis and main shaft vibration response analysis.

(2) Since various parts of the pump-turbine are subjected to severe alternating stress, the life expectancy was estimated for each part by the linear cumulative damage rule on the basis of the following operating patterns and it was found that there is sufficient margin for 50 years of operation.

(a) Turbine operation: 7 hours/day
(b) Pump operation: 10 hours/day
(c) Pump starting: 4 times/day, 90 sec continuous
(d) Load rejection: 1 time/week, 30 sec continuous
(e) Pump trip: 1 time/week, 30 sec continuous

(3) Research was performed on fatigue strength, fracture toughness and other properties of the main materials such as high nickel 13% chromium steel, and steel plate with excellent mechanical properties in the thickness direction has been developed.

(4) Fracture strength evaluation methods were employed to rationally evaluate material and welding defects.

(5) Electroslug welding was applied mainly for thick plate welding in the stay ring, head cover and other parts but heat treatment was performed to improve fracture toughness after welding.

(6) According to the stress levels obtained by the finite element method and the operating conditions, welding procedure and non-destructive test levels for each welding seam were specified and controlled.

2. Pump-turbine Structure Analysis by the Finite Element Method

Analysis of the pump-turbine assembly was performed for each operating condition by means of an axisymmetric computer program. *Fig. 18* shows a typical example of the analysis results (deflection diagram). From these analysis results, the following important data for design work can be obtained:

(1) Stress in each part
(2) Deformation in each part, especially:
  (a) Radial deformation of the main bearing support joint of the top cover
  (b) Changes in the gap in the runner seal
  (c) Angular rotation of the center line of wicket gate bearings
  (d) Changes in the wicket gate side clearance
  (e) Gap changes in the gate operating ring bearing
(3) Tensile and shear force applied to the connecting bolts between each part, particularly:
  (a) Connecting bolts between stay ring and head cover
  (b) Connecting bolts between head cover and bearing support
  (c) Connecting bolts between stay ring and discharge ring

  To prevent fatigue failure of such bolts, the necessary bolt dimensions and initial stress were obtained.
(4) The foundation load and necessary initial stress of the anchor bolts

3. Stay Ring

The forces applied to the stay ring are the spiral casing tensile force, the head cover tensile and radial force and the direct hydraulic force acting on its wetted surfaces. The stay ring is subjected to the most severe strength conditions among the pump-turbine stationary parts. The parallel type stay ring which has been adopted in many Fuji turbines including the 306MW turbine in the Portage Mountain Power Station in Canada was used.

Fig. 19 shows the stress distribution in the classical

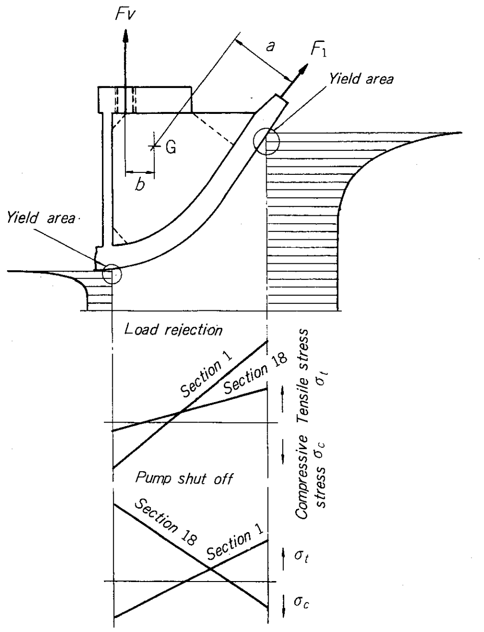


Fig. 19 **Stress distribution of classical design stay ring**

type stay ring but it has the following defects:
(1) Because the moment acting in the stay ring changes considerably in accordance with the casing cross section and operating conditions, a high level of stress occurs in the outer and inner edge of the stay vane.
(2) There is a large stress concentration in the intersection of the stay vane and crown plate.

In the case of parallel type stay ring acting points of the casing tensile force were changed in each section so that the moment due to the casing tensile force and that due to the head cover tensile force were balanced in each casing section. Therefore, changes in the stay vane stress due to the casing sections were minimized and changes in stay vane stress in accordance with operating conditions were also decreased.

Fig. 20 shows an example of stress analysis results by means of the three dimensional FEM analysis. According to these analysis results, the crown plate is subjected to a high tensile force in the thickness direction. Therefore, steel plate with excellent strength in the thickness direction was used for the crown plate.

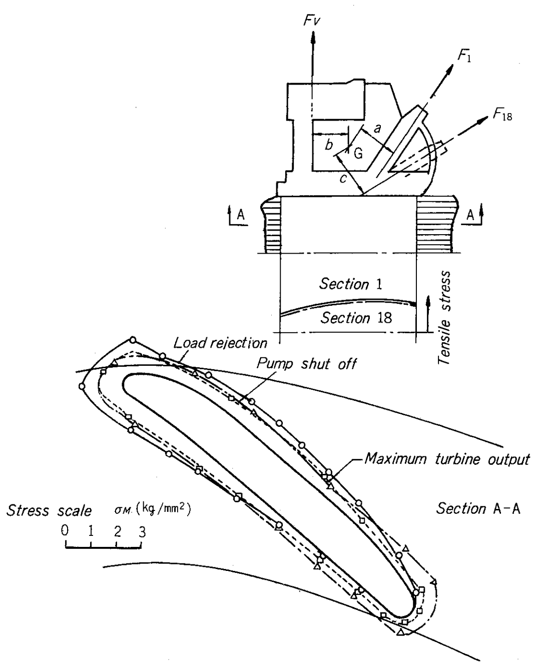


Fig. 20 **Stress distribution of new design of stay ring**

4. Head Cover

Various types of comparative investigations were performed concerning the position and shape of the connection flange with the stay ring which is particularly important for increasing head cover rigidity, and finally, a solid flange was adopted. In this way, it was possible to keep the radial deformations of the bearing support and main shaft seal attachment parts and the angular rotation of the wicket gate bearing center line within the permissible values.

The complete surface facing the wicket gate was lined with 18-8 stainless steel against cavitation damage and rubber backed bronze wicket gate seal was provided to prevent water leaks when the wicket gate is completely closed.

5. Discharge Ring

The discharge ring in classical medium and high head pump-turbines had the following problem points because of its lower ridigity.

Normally, a very high hydraulic load of the same level as the head cover load F_V (about 11,500 tons in case of Chongpyong) acts on the discharge ring in the downward direction. However, when the rigidity of the flange connecting the discharge ring and the stay ring is low, the major portion of this downward force is transmitted to the foundation concrete and only small portion reaches the stay ring.

Therefore, the hydraulic force F_V applied to the head cover is transmitted directly to the stay ring anchor bolts via the stay ring. Because this force acts as an alternating load in accordance with the frequency of pump-turbine starting, lifting of the stay ring may occur and the wicket gate side clearance increase after long-term operation because of the fatigue of concrete around the stay ring.

The new discharge ring developed to solve these problems limits the supprot from the foundation concrete to inner and outer support rings. According to the calculations, 40% of the hydraulic force applied to the discharge ring was applied to the inside support ring and 60% to the stay ring. Therefore, the tensile force acting on the anchor bolts is $0.4F_V$, i.e. 40% of the conventional value.

6. Wicket Gate and Gate Operating Mechanism

The wicket gates of high head pump-turbines is subjected to the highest stress and their fatigue strength was therefore investigated together with a detailed stress analysis in

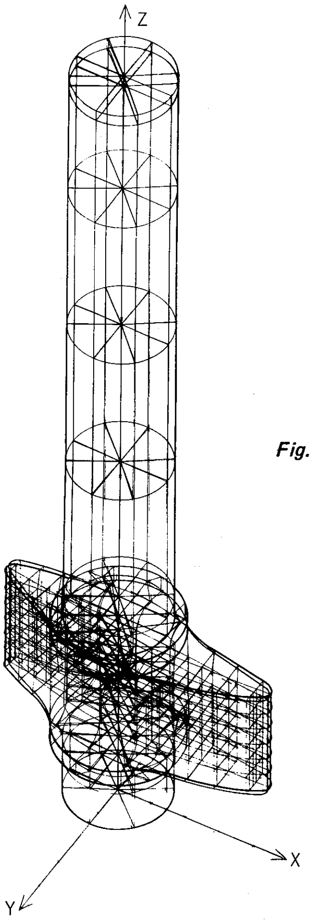


Fig. 21 Stress analysis of wicket gate by FEM

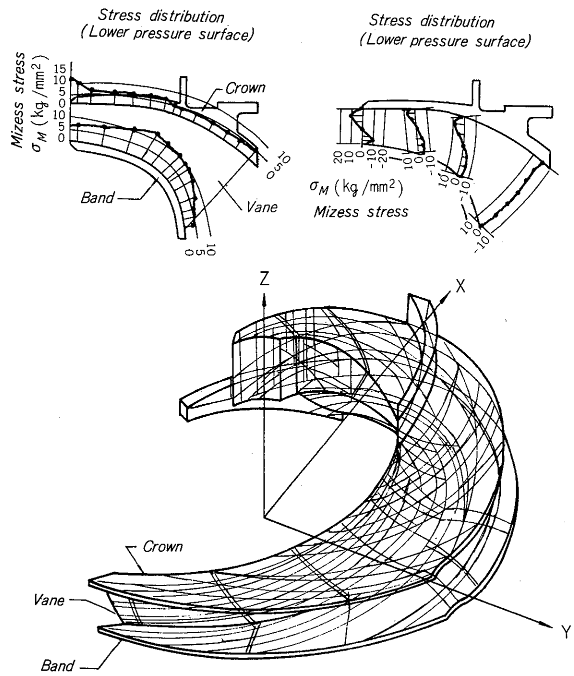


Fig. 22 Stress analysis of runner by FEM

consideration of the following conditions in addition to those given in **III**, **I**, (2) by means of a three dimensional FEM computer program:

(1) Shear pin break: 1 time/year (each wicket gate)

(2) Gate squeeze conditions: 8 times/day

The fillet shape of the vane shaft interrections was decided in order to alleviate the stress concentration and prevent the occurrence of cavitation. The wicket gate material used was high-nickel 13% chromium steel for reasons of strength and cavitation.

A wicket gate restraining mechanism was provided to prevent successive shear pin break and wicket gate damage due to striking the adjacent wicket gate when a shear pin was broken.

7. Runner

Detailed stress and deformation analysis were performed for various operating conditions by means of the three dimensional finite elements method, and a rational vane thickness distribution was employed. In the case of the outside periphery of the runner which has especially severe strength conditions, a fillet profile with small stress concentrations was used. *Fig. 22* shows typical stress analysis results.

Because of the strength and cavitation requirements, high-nickel 13% chromium steel was used as the runner material.

8. Guide Bearing

Because of the large radial thrust fluctuations of the high-head pump-turbine, a segmental bearing with large load capacities was used and to increase the bearing rigidity, a taper key was employed instead of the conventional screw for bearing clearance adjustment.

9. Main Shaft Seal

The main shaft seal of the pump-turbine must have excellent sealing characteristics against water and air leaks under varying high sealing hydraulic pressure. To meet these requirements, Fuji Electric's axial balanced type seal was employed and the optimum sealing materials were selected by large-scale model tests.

IV. CONCLUSION

There is a gradual increase tendency in head and capacities in pump-turbine power plants which has resulted in stress on economy and reliability and the development of appropriate design and manufacturing technology. Basic design investigations by means of flow and stress analysis computer programs have become closely linked with confirmation by model and field tests and these have made possible the manufacture of high reliability products by the establishment of a quality control system at each stage of manufacture using non-destructive test, fracture mechanics applications, etc.