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# APPLICATION LIMITS OF FRANCIS REVERSIBLE PUMP-TURBINE TO HYDRO-PLANT WITH LARGE HEAD VARIATION

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## I. INTRODUCTION

In case of applying Francis type reversible pump-turbine to pumped-storage hydro-plant with large head variation, a special consideration shall be taken for operational stability both in high head pumping mode and in low head generating mode.

In high head pumping mode, there is possibility that the flow separates from the low pressure surface of impeller vanes at pump entrance and/or the back-flow appears from impeller inlet into suction tube. As a result of it, flow through impeller is disturbed and hydraulic pressure fluctuation increases and so it is feared that stable operation cannot be carried out. On the other hand, in low head generating mode, smooth operation is limited due to occurrence of the flow separation from the high pressure surface of runner vane leading-edge. These are caused by increase of flow attack angle against hydrofoils of impeller-runner, and these phenomena are inherent to Francis type pump-turbine with fixed vanes.

In order to confirm operational limits due to such hydraulic unstable phenomena, it is the most effective to investigate them through field test of prototype machine together with careful model testing at hydraulic laboratory. Hatanagi pumped-storage power plant in Japan is one of the plants with the largest head variation in which net head changes 50% from 100m to 50m. The orginal Francis type pump-turbine for Hatanagi plant (unit 1) started the commercial operation in September 1962. In high head pumping mode with more than 94m gross head, operation had been limited at that time due to heavy vibration, and in low head generating operation with less than 70m gross head, roughness was heavy, too.

In order to expand stable operating range of Hatanagi pump-turbine, the following countermeasures were taken. The back-flow phenomena in high head pumping mode were judged to be very harmful and so such new impeller-runner was newly developed and replaced as make back-flow starting point at pump-turbine shift to higher head (lower discharge) as much as possible. In order to reduce mechanical vibration of pump-turbine, new head cover with more stiffness was adopted.

The new impeller-runner was developed under the severe restriction that flow passage of fixed portions including such as spiral casing and stay vanes could not be changed at all. At first, a model pump-turbine completely homologous to the original prototype machine was developed and tested carefully at Fuji Hydraulic Laboratory. Hydraulic phenomena simulated by the model were thoroughly compared with those in prototype operating experience and hydraulic relation between model and prototype was made clear. Then, as to four new model impeller-runners developed thereafter, comparative model tests were carried out and the best one which showed overall excellent test reults of performance, cavitation and vibration characteristics, etc. was adopted as the prototype impeller-runner. Details of the development on improved impeller-runner are given in the reference (1).

The adoption of new impeller-runner and head cover, and following operational tests were successfully completed in May 1974. Field tests were repeated three times in accordance with water level change of upper reservoir over one year. The test results, thus collected, covered over a wide head range from 98m (max.) to 65m (min.). As a result of total analysis of whole field test data as well as model test results, some important criteria were established as to the application limits to the large head variation of Francis type reversible pump-turbine. The following is the field test results of the application limits, mainly of comparisons between operating characteristics of old and new impeller-runners.

## II. REPLACEMENT OF PUMP-TURBINE

*Fig. 1* shows a comparison of cross sections of the pump-turbine before and after improvement and *Fig. 2* is a photo of the new impeller-runner.

### 1. Impeller-runner

A comparison of hydraulic design parameters for old and new impeller-runner is given in detail in reference (1) and further explanations are omitted here.

The maximum outer diameter of vanes is 4.08m, the same as crown outer diameter and it is 0.65% larger than that of old impeller-runner. The impeller-runner is made of 13Cr cast steel which has good cavitation resistance characteristics. The impeller-runner was splitted in two parts because of transportation limits and the parts

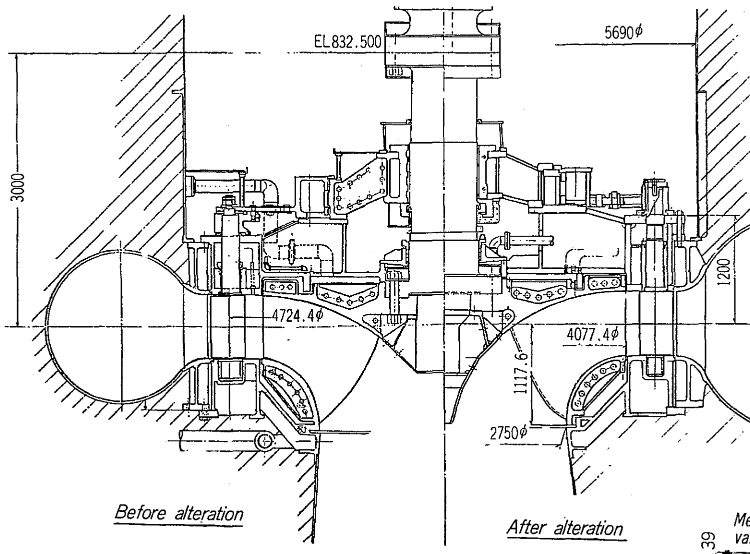



*Fig. 1* Comparative cross-section of pump-turbine

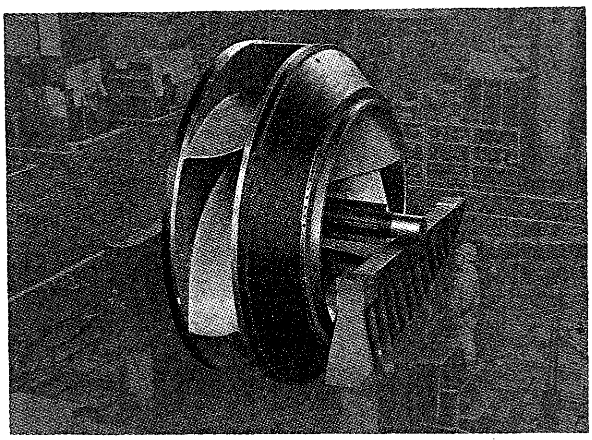


*Fig. 2.* Photograph of new impeller-runner

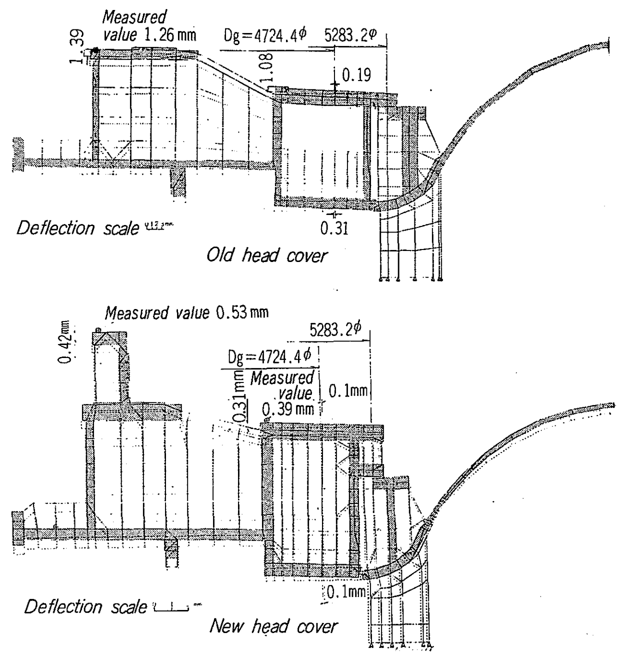



*Fig. 3.* Comparative deflection of head cover computed by finite element method

are connected by bolts. To minimize hydraulic loss due to connecting flanges, flange cover is of one-piece ring-structure by means of bolt tightening to the impeller-runner. The holes on the cover through which water enters are provided to balance pressure inside and outside the cover.

### 2. Head Cover

For the head cover, a thorough stress analysis was performed by the finite element method computer program. as a result, the height of head cover was increased to reduce deflection. The stiffness of flange for attachment to stay ring was especially increased by using a double plate structure with ribs. As a result, the deflections of bearing support and wicket gate bearings were reduced to less than half of that of old head cover. *Fig. 3* shows the comparative deflection of old and new head covers computed by the finite element method.

## III. FIELD TEST RESULTS

### 1. Pumping Mode

1) Vibration characteristics during high head pumping

The vibration during high head pumping are closely related to back-flow phenomenon at impeller inlet. The new impeller was developed so as to shift the start of back-flow to the lower-flow and higher-head region. The success of the shifting was confirmed from the field test results as follows;

The old impeller entered into back-flow region from a gross head of near 94m at which vibration started to become large. With the new impeller, there are few pressure fluctuation and mechanical vibration even at a gross head of 98m and there exists the wicket gate opening range of more than 5% which shows smooth operating conditions. The total dynamic head at which back-flow starts was improved by more than 4m when compared with the old impeller.

The hydraulic pressure fluctuation in the position between impeller outlet and wicket gates during pump operation (hereafter referred to as priming pressure fluctuation) can be considered to indicate the intensity of disturbance of main flow discharged from impeller. The priming pressure fluctuation can therefore be treated as a typical value showing the degree of stability during pumping operation.

*Fig. 4* shows the results of measurements of priming pressure flucuations at a gross head of 94m where vibration started to become large in the old impeller. This is a typical comparison of flow stability between new and old impellers. In case of old impeller, there were high vibrations outside the best gate opening. With the new impeller, it was found that the pressure fluctuations remained small overa a wide opening range and main flow was stable. The values predicted from model test also showed the same tendencies as measured values.

The vertical vibration of bearing support and shaft run-out which are typical mechanical vibrations in pump-turbines have been reduced drastically as shown in *Fig. 5*

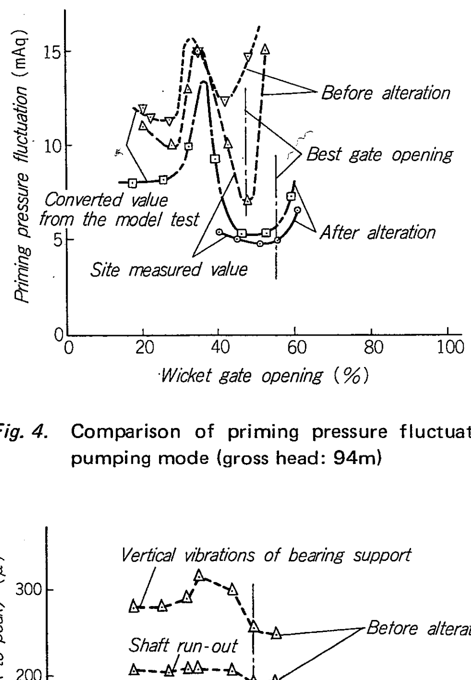



*Fig. 4.* Comparison of priming pressure fluctuation at pumping mode (gross head: 94m)

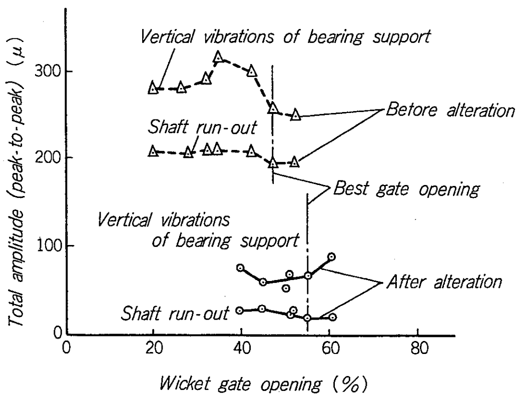



*Fig. 5.* Comparison of vibration and shaft run-out at pumping mode (gross head: 94m)

when compared with before the improvement. These were accomplished with increased stiffness due to replacement to new head cover as well as reduced pressure fluctuation due to new impeller.

2) Vibration characteristics during low head pumping

Since there are large head variations in this storage plant, it is necessary to limit the amount of discharge during pumping operation at a lower gross head of less than 70m in order to suppress excessive cavitation on the high pressure surface of impeller inlet. The careful tests were, therefore, performed concerning wicket gates throttling at low head pumping operation. The results showed that even when there was a small gate opening throttled sharply from the best gate opening, there was no increase in vibration of wicket gate link mechanism, etc. The wicket gates throttling operation at low head pumping presented no problem at all.

3) Best gate opening during pumping mode

*Fig. 6* shows the overall pumping characteristics measured with the new impeller. The best gate opening tested cincides well with the opening predicted from model test results. The limit of gate opening for favorable operation, back-flow starting opening, etc. were judged from the pressure fluctuations, vibrations, noise, etc. in the pump-turbine. These oepnings versus gross head showed almost the same tendencies as the characteristics predicted from model test. It was thus clear that it is possible to make sufficient prediction of not only performance but also vibration characteristics from carefully performed model test.

*Fig. 7* shows priming pressure fluctuation and mechanical vibration at best gate opening versus gross head. Because gate opening is throttled during low head pumping, the priming pressure fluctuation shows a slight tendency to increase. The vertical vibration at bearing support shows almost no change over a wide range of gross head.

*Fig. 8* shows a comparison between the performance with old and new impellers at best gate opening. With old impeller-runner, pump input power was too small compared with rated turbine output. The pump input power with new impeller-runner increased by about 10% on higher head operation. This is because new impeller was designed to increase discharge on higher head as described in reference (1)

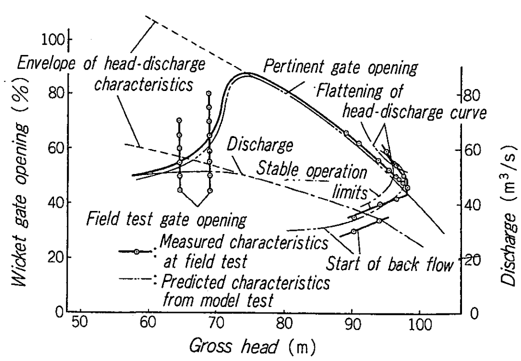



*Fig. 6.* Stable operating range at pumping mode

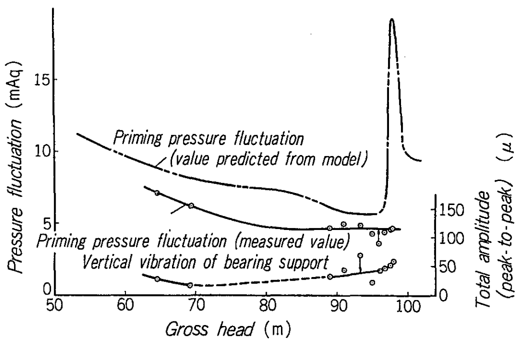



*Fig. 7.* Pressure fluctuation and vibration at pertinent pump operation versus gross head

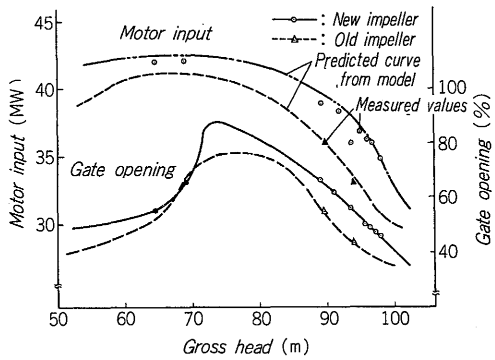



*Fig. 8.* Comparative pump input power

The measured values of pump input power with respect to gross head coincide with the values predicted from model test.

## 2. Generating Mode

1) Vibration characteristics during high head generating

*Fig. 9* shows a comparison of vibration characteritics at a gross head of 90m between old and new runners. There were almost no differences between before and after improvement in the pressure fluctuation values in spiral casing inlet and upper draft tube. The gate opening range in which draft surging noise occurred at partial load operation was almost the same.

On the other hand, there were wide ranging improvements in bearing support vertical vibration and shaft run-out. In case of old runner, the flow separated from low pressure surface of runner vane inlet at gate opening of more than 60%. Vibration and noise were high as can be seen typically in vertical vibration of bearing support shown in *Fig. 9.* However, with new runner, this flow separation was suppressed so that pressure fluctuations at runner inlet became smaller. The increase of head cover stiffness, of course, contributes to the reduction of mechanical vibration.

*Fig. 10* shows the relation between head-discharge characteristics and the range of occurrance of flow separation at runner vane inlet based on model tests result of new

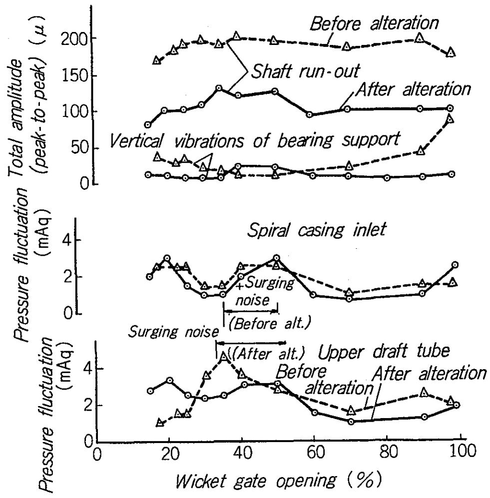



*Fig. 9.* Comparison of vibration characteristics at generating mode (gross head: 90m)

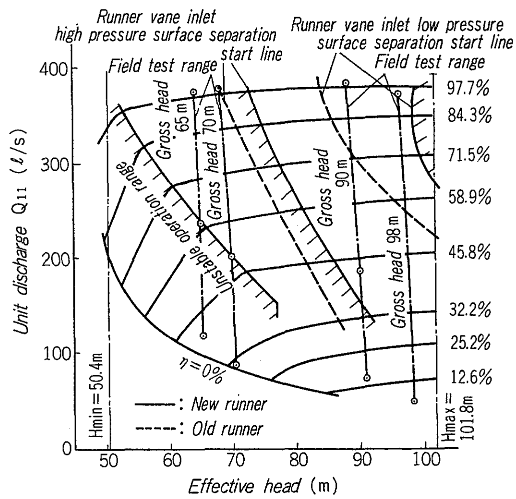



*Fig. 10.* Stable operating range at generating mode

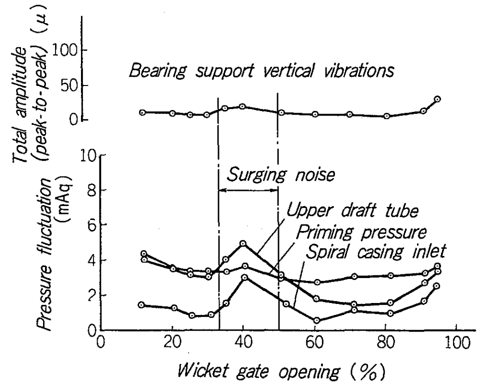



*Fig. 11.* Pressure fluctuation and vibration at high head generating mode (gross head: 98m)

runner. The flow separation range of old runner is also shown in this figure. With the new runner, there was a wide range of heads at which there was no separation at inlet when compared with old runner. The stable operation range at higher heads was remarkably improved.

*Fig. 11* shows the measured results of pressure fluctuations etc. of new runner, at a gross head of 98m which is near the maximum head. Even at this gross head of 98m, the results being the same as those of model tests, no separation occurred in the flow from the runner vane inlet low pressure surface. Vibration and noise were both small and operation at high heads was smooth.

2) Vibration characteristics during low head generating

As partial load vibration was large at heads of less than 70m with the old runner, generating operation had to be restricted to only near full gate opening. Since the new runner was designed with special emphasis on vibrational improvements at higher heads, the head at which high pressure surface flow separation starts at runner vane inlet was predicted to shift rather higher head from model test as can be seen in *Fig. 10.* Therefore, careful field tests were performed to determine the stable generating operation range at gross heads of 70m and 65m.

*Fig. 12* shows the vibration characteristics at a gross head of 65m. There was flow separation from the runner vane inlet high pressure surface in whole gate opening ranges and both noise and priming pressure fluctuation were high. However, at gate opening of over 70%, the pressure fluctuations at spiral casing inlet and upper draft tube and the vertical vibration of bearing support showed almost the same low values as a high head operation when there was no separation of flow at runner vane inlet. As can be seen in *Fig. 13* (a), priming pressure fluctuation is only value which detects the local pressure reduction due to the

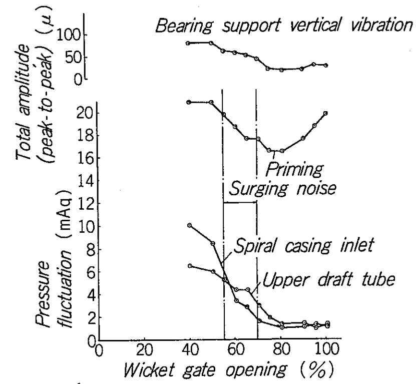
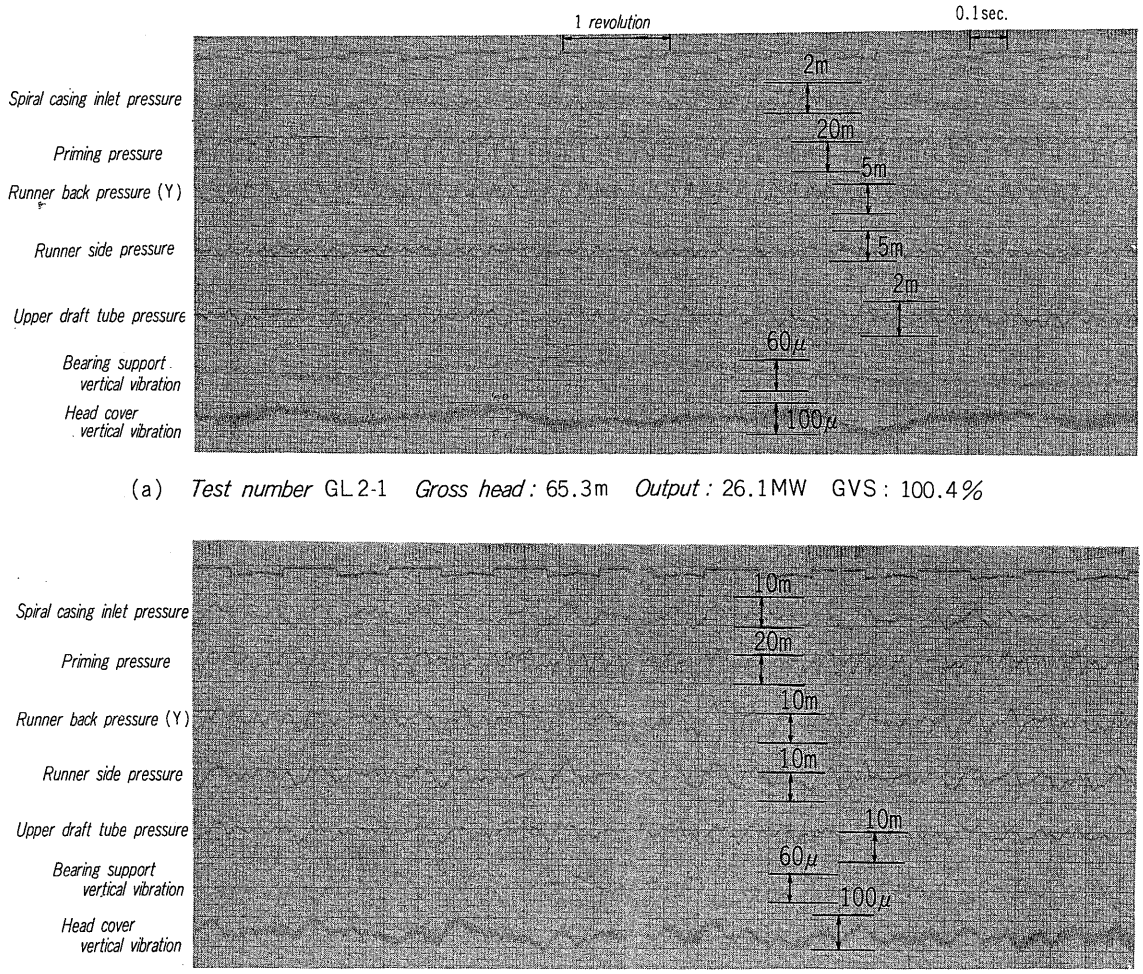



***Fig. 12.*** Pressure fluctuation and vibration at low head generating mode (gross head: 65m)




(a) *Test number* GL 2-1 *Gross head:* 65.3m *Output:* 26.1MW GVS: 100.4%

(b) *Test number:* GL 2-10 *Gross head:* 65.1m *Output:* 8.4MW GVS: 50.4%

***Fig. 13.*** Examples of vibration oscillogram at generating mode

flow separation on the runner vane inlet high pressure surface which passes round near the priming pressure measuring position. As priming pressure fluctuation occurs uniformly and continuously around the runner, there are few effects on the main flow. It is concluded that vibration energy given to the pump-turbine is small even when priming pressure fluctuation amplitude is large at the gate oepning of over 70%.

However, at an gate opening of lss than 60%, the flow disturbance occurs to the whole runner with the large pressure fluctuation of frequency 2-3 times that of the rotating speed of runner as shown in *Fig. 13* (b). The hydraulic pressure above crown and below band of runner changes at the same phase irrespective of circumferential position. Therefore, the hydraulic pressures at spiral casing inlet and upper draft tube change at almost the same frequency and the waveforms of those including priming pressure fluctuation are distorted. Under these rough generating operational conditions, the vibration energy to the pump-turbine is so large that mechanical vibration increases. As a result, it has been concluded that the continuous generating operation should be limited within short hours at the gate opening of less than 60%.

According to results from model test, the gate opening at which this type of pressure fluctuation starts to become large coincide well with the folded point of the unit flow characteristic curve. Therefore, it is evident from *Fig. 10* that the lower limited head at which smooth generating operation is possible at the maximum gate opening is approximately 53m.

3) Generating output versus gate opening

*Fig. 14* shows the test result of generating output versus gate opening for the new runner. At the time of low heads and partial loads, a generating output somewhat higher than the value predicted from model test.

*Fig. 15* shows a comparison of new and old runner outputs at the net head of 90m. The new runner output showed to exceed over the full range of gate opening and it was proven that turbine characteristics were improved.

### 3. Head Cover Deflection

*Fig. 16* shows a comparison of the deflection in bearing support of old and new head covers measured at a gross head of 90m. The deflection was reduced to less than half of that before improvement. There was also close agreement between measured deflection and values calculated by the finite element method.

The measurement results during generating operation clearly indicated a tendency for a smaller deflection when the gate opening became small since the static pressure at the wicket gates outlet decreased. Similarly, when the gate opening decreased under a constant gross head during the pumping operation, the total dynamic pressure at impeller outlet became large and the static pressure inside distributor increased which means that the deflection of head cover became large.

The effect of the head cover stiffness increase on the decrease in vertical vibration of bearing support can not be clearly compared, since the vibromotive force of pressure fluctuations, etc. are not the same in old and new impeller-runners. The decrease in the bearing support vibration can be evaluated to be proportional to or more

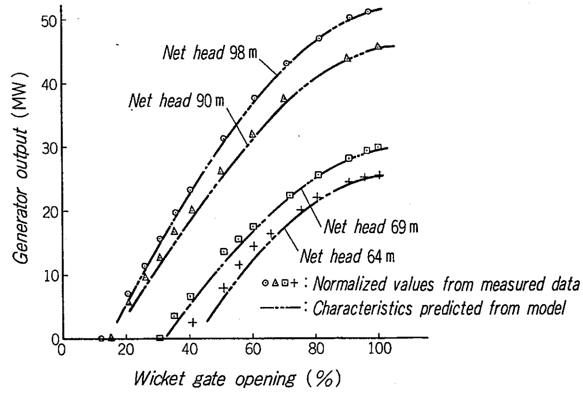



*Fig. 14.* Results of generating load test on the new runner

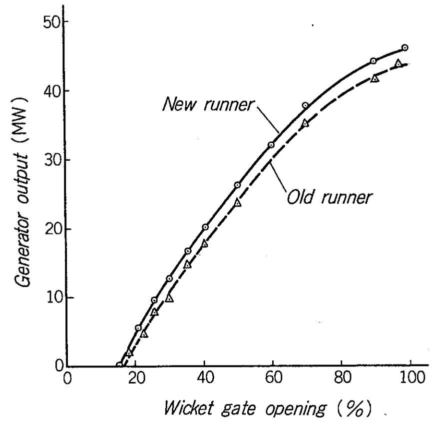



*Fig. 15.* Comparative results of generating load test (net head: 90m)

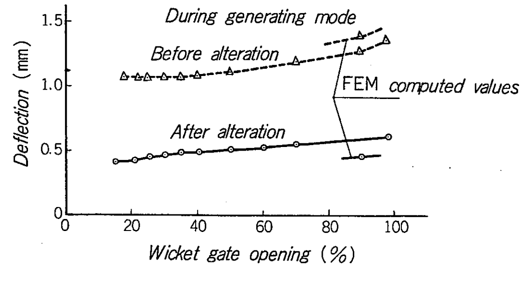



*Fig. 16* Results of deflection at bearing housing (gross head: 90m)

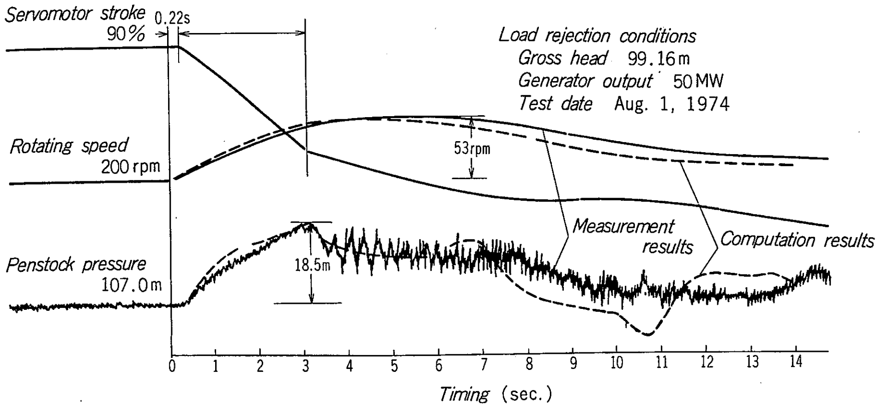



*Fig. 17.* Results of generating load rejection test

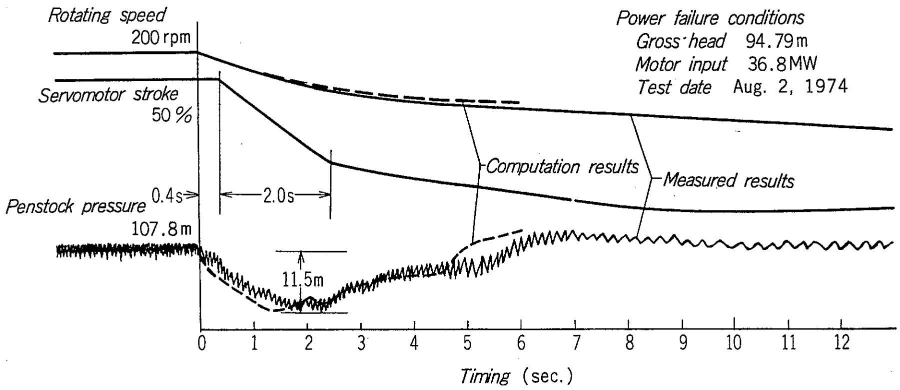



*Fig. 18.* Results of pump power failure test

than the decrease in static deflection.

### 4. Generating Load Rejection Characteristics

With the old runner, wicket gate servomotors were closed with the sequence which has single cushionning time. Therefore, the unit ran deeply into the reverse-RPM pump region (the fourth quadrant) with high vibration after the load rejected. With the new runner, servomotors closing sequence which has the double cushioning times has been used to suppress the deep intrusion into the reverse-RPM pump range. A thorough computer simulation to the load rejection under various generating conditions was performed using the four quadrant characteristics of new impeller-runner. As a result, the most suitable closing time was decided. *Fig. 17* shows a comparison of the load rejection result at field test with computer result for the above mentioned optimum closing time. There was good agreement between computer simulation and measured values.

### 5. Pump Power Failure Characteristics

*Fig. 18* shows a comparison of pump power failure test result at gross head of 94.8m with computer analysis. As in case of generating load rejection, there was good agreement between measured values and computer results.

## IV. CONCLUSION

The paper has given the field test results over a wide head range from 98m to 65m, in order to investigate the limit of application to the large head variation of Francis type reversible pump-turbine. The conclusions obtained are summarized as follows;

(1) In high head pumping mode, smooth operation can be attained by adopting the impeller designed as shift back-flow starting point to higher head (lower discharge) region.

(2) The operational limit due to back-flow in high head pumping mode can be predicted from the priming pressure fluctuation measurd at model test.

(3) In low head generating mode, even if there were large priming pressure fluctuation, stable operation can be expected as far as flow disturbance has no influence on the whole runner.

(4) The operational limit due to main flow disturbance in low head generating mode can be predicted from the folded point of unit discharge versus unit speed curve at constant gate opening obtained by model test.

There are few examples of detailed field test date together with model test results as to the application limits of high head pumping and low head generating operations where problems arise in smooth operation of Francis type reversible pump-turbines with such a wide head range. The paper would contribute the development of the subject.

### Reference:

(1) T. Ueda et al.: Vibration reduction of reversible pump-turbines, Fuji Electric Review 19 No. 4, 1973