700-m 400-MW Class Ultrahigh-head Pump Turbine

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OVERVIEW: The first unit of the world-leading ultrahigh-head pump turbine was commissioned in December 1999 at the Tokyo Electric Power Company (TEPCO)'s Kazunogawa pumped-storage power plant in Japan. This paper describes several state-of-the-art technologies applied to the hydraulic and mechanical designs, which ensure the pump turbine's high reliability and stable operation. Kazunogawa; vertical Francis-type single-stage reversible pump turbine, turbine head 728 m, output 412 MW, maximum pumping head 779 m, speed 500 min^{-1}.

INTRODUCTION

ELECTRICITY is mostly produced by thermal and nuclear power in Japan. For the purpose of shaving the peak and the bottom load in the power grid, over 40 pumped-storage power plants have been built in Japan since the first reversible pump turbine was put into operation at the Oomorigawa Power Station in 1959. The higher the pumping head, the smaller the discharge. Small reservoirs and compact pump turbines save on construction costs of high-head power plants.

Since the situation described above the head and unit capacity of pump turbines have been increasing year by year. Since the development of 500-m, 300-MW class pump turbines in the 1970s, advances in technology have led to a major breakthrough with the development of the 700-m, 400-MW class pump turbine at the end of this century.

One of the problems with the ultrahigh-head pump turbine is that its runner is exposed to the cyclic high-pressure pulsation caused in the runner blades and wicket gate cascade, where the flow velocity of the highly pressurized water is as much as 150 m/s. The runner, which is the most important component of the pump turbine, can reduce the reliability of the turbine if its resonance increases the alternating stress. This is why Hitachi, in collaboration with the Tokyo Electric Power Company (TEPCO), started a two-year research program in 1989 to study the runner resonance phenomena and technique for avoiding the resonance. The ultrahigh-head pump turbine has been designed utilizing the latest technology such as computational fluid dynamics (CFD) and three-dimensional computer-aided design and associated structure analysis (3D-CAD/CAE) to ensure that it has high strength reliability and high hydraulic performance. Furthermore, several tests were conducted with actual size components to confirm the performance and durability of the components under severe conditions.

Field tests of the No. 1 Unit of Kazunogawa Power Station verified that the sound and vibration levels are lower than those of the conventional 500-m, 300-MW class pump turbines.

TRENDS IN PUMP TURBINE TECHNOLOGY

The history of the maximum pumping head of pump turbine is shown in Fig. 1. The first domestic pump turbine with a pumping head of 130 m started operation about 40 years ago. Within 15 years, the pumping head had exceeded 500 m for the first time at Numappara (Electric Power Development Company, 230 MW). At the Kazunogawa Power Station (TEPCO, 412 MW), which started commercial operation in December 1999, the pumping head exceeded 700 m. The development of the pump turbine is a result of the on-going research and development and advances in CFD and 3D-CAD/CAE technology.

FLOW ANALYSIS

Flow analysis was applied to the fluid design of all of the flow passages from the spiral case to the draft tube of the ultrahigh-head pump turbine. This analysis provides design engineers with the pressure and velocity distribution of the flow field, the submergence and location of cavitation incipience, the pumping head at which reverse flow and severe vibration occur, and the hydraulic loss and its distribution. The analysis also makes it possible to more precisely predict

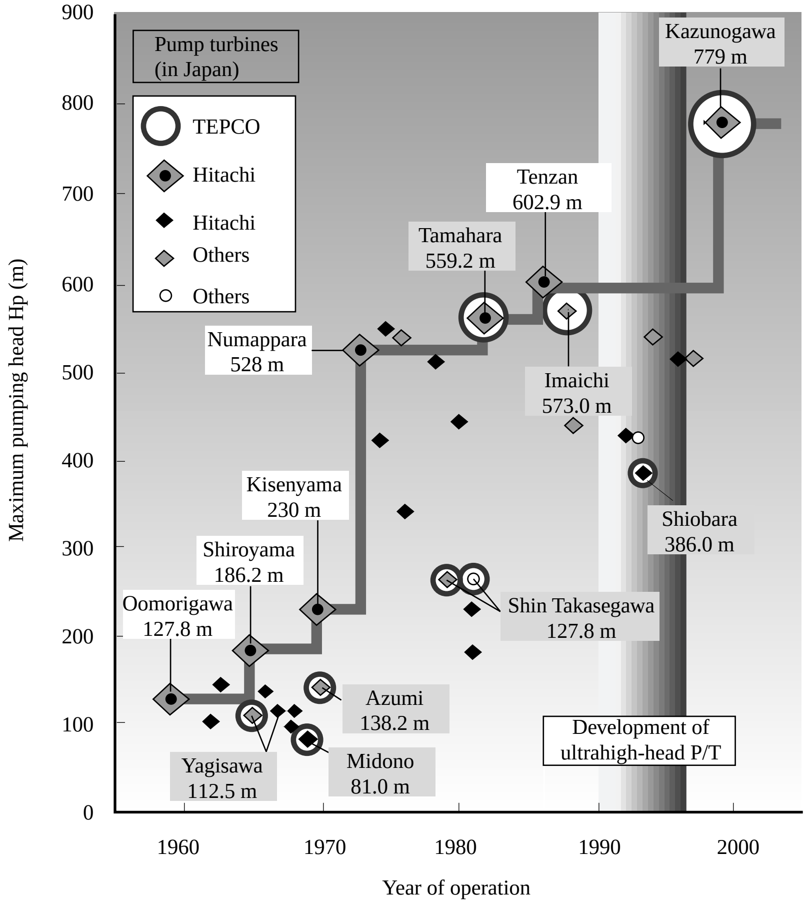


Fig. 1— Trend of Maximum Pumping Head.

hydraulic performance than before.

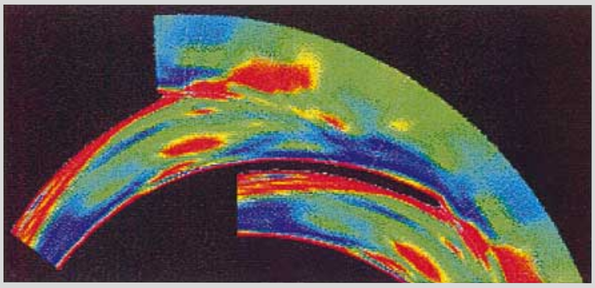
In the past, it took a long time to design the flow passage because a physical model test with a Pitot tube and/or the use of an oil film method was necessary to review flow conditions. At present, however, it has become possible to quickly review flow conditions of the preliminary design and improve the design using a numerical model. The present fluid design process has changed from the conventional cut-and-try method to the present CFD-based method.

For the final fluid design, a model test was carried out to confirm that the hydraulic performance (efficiency, cavitation, pressure pulsation, and hydraulic thrust) satisfies the required specifications for the design to obtain the wicket gate torque and four quadrant characteristics.

A vane profile design tool was applied to the fluid design. The tool can easily create any smooth vane profile and maintain the inlet and outlet angles and vane clearance that the design engineer intends. It runs on a personal computer and quickly generates a grid model for flow analysis. It provides numerical data for manufacturing a prototype of the model and the model itself.

Hitachi has a quasi three-dimensional non-viscous steady flow analysis that calculates velocity and pressure distributions within several minutes on a work station, and a three-dimensional k-ε turbulent flow analysis and unsteady turbulent flow analysis that runs on a super computer. Utilizing an appropriate analysis code for each stage from outline to the precise design stage, Hitachi was able to develop an efficient and firm fluid design. The loss distribution of normal operation and the velocity distribution of reverse flow are shown in Fig. 2 (a) and (b), respectively. These two samples, showing the vortex flowing and the velocity fluctuating on the visualization tool, are the results of the unsteady flow. It is possible to visualize the distribution of any numerical value that is defined optionally to have physical meaning and to ease checking our vane profile design.

The fixed vane of the pump turbine runner operating in the off-design region can cause cavitation, flow separation, and swirl flows, which affect the performance and increase sound and vibration. A flow analysis of these phenomena was therefore conducted



(a) Loss distribution (pumping mode)

(b) Velocity distribution (reverse flow)

Fig. 2— Flow Analysis (Runner)

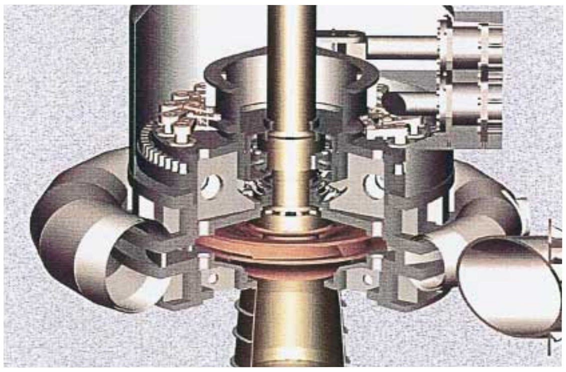
for the off-design region to improve performance.

In the case of the ultrahigh-head pump turbine, hydraulic efficiency is apt to be low because leakage through the runner seal becomes large and the discharge becomes small relative to the output.

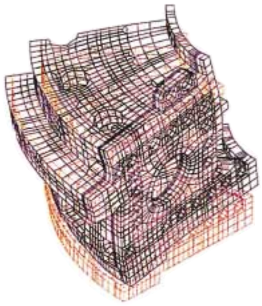
Hydraulic efficiency was improved by applying a labyrinth seal with four stages to the crown and a labyrinth seal with five stages for the band of the runner, whereas for conventional pump turbines, labyrinth seals with three and four stages have been used.

STRUCTURE DESIGN

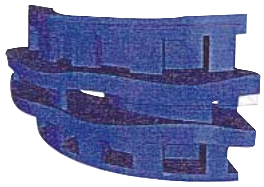
In the 1990s, the use of CAD/CAE spread rapidly into wide use, and made it easy to analyze the stress/ deformation and vibration in complex structures. We used a 3D-CAD/CAE system to design the structure of the ultrahigh-head pump turbine. Once a three-dimensional model of the structure was made, it was easy to make modifications and attain the optimum design, and to take various structural analysis results under various boundary conditions into consideration. The following points were analyzed; the natural frequency of the runner, the deformation of the flange gap at the packing rubber strings between the head cover and the speed ring, the stress and deformation of the stay vanes, the contact pressure or deformation gap at the flanges of the split head cover, the deformation of the main shaft bearing related to the metal gap, and the stress and deformation of the inlet valve. A 3D-CAD drawing of the pump turbine assembly is shown in Fig. 3 (a) and sample outputs of the head cover deformation and speed ring are shown respectively in Fig. 3 (b) and (c).



(a) 3D-CAD assembly drawing



(b) Deformation of head cover



(c) Deformation of speed ring

Fig. 3— 3D-CAD Assembly and Deformation.

The dimensions of the design components and possible interference between them were checked by simulating the actions of the components on the computer. In addition, accessibility of the narrow head cover was checked in consideration of the installation and maintenance of the pump turbine, after modeling the pipe arrangements provided for the guide bearing and the shaft seal coolings, thrust equalizing, and air supply, in addition to the stiffener of the head cover and the wicket gates.

The structural features of the ultrahigh-head pump turbine are compared below with the conventional 500-m, 300-MW class pump turbine.

(1) The spiral cases and runners of both pump turbines are almost of the same size. The rotational speed, however, was increased from 429 rpm to 500 rpm. The design pressure is as high as 1,200 m, which was determined by studying the hydraulic transient and pumping shut-off pressure. The components and assembled structure were designed so that deformation

does not exceed that of the conventional pump turbine, in order to prevent water from leaking through the packing over long periods of operation, and excessive friction of the movable components such as the wicket gate and the gate operating mechanism. Consequently, all of the components are thicker and stiffer than in the conventional turbine.

(2) The speed ring's twenty thick stay vanes designed to withstand the design pressure hinder accessibility for inspection and maintenance. A speed ring with ten long stay vanes has therefore been adopted to provide easier access and improve hydraulic performance.

(3) An actual head model test revealed that a runner with six vanes causes a higher stress amplitude than a runner with seven vanes, so the seven-vane was adopted. Accessibility was checked in a comparison of six- and seven-vane runner mockups.

(4) A stainless steel lining was attached with numerous screws to the wetted surface of the head and bottom covers of the conventional pump turbine. For the ultrahigh-head pump turbine, however, rolled stainless plates (13Cr-5Ni) were welded on to prevent their removal by the pressure of water seeping into the gap between the lining plates and the covers.

(5) The lubrication oil of the main shaft guide bearing is cooled by a heat exchanger located outside of the turbine pit. The oil is supplied to the oil dam in a forced oil circulation system, thus creating more space for inspections and maintenance to be carried out inside the small head cover.

(6) Because the pipes inside the head cover suffer severe vibration from the head cover, block supports were added. The proper location and distance of the supports were investigated considering the pipe diameter and thickness.

(7) Friction devices were installed on the gate operating mechanism to prevent sequential gate fractures triggered by shear pin failures of the wicket gate vane.

(8) Deformation of embedded parts such as the bottom cover and speed ring is inevitable in the field welding and concrete pouring process. After the embedded components were installed, field machining was carried out to eliminate the waviness of the sealing planes of the head cover.

COMPONENT TESTS

Several tests of the important components were carried out in the shop and laboratory to confirm their performance and durability under severer conditions than those the conventional pump turbine normally operates under.

Actual Head Model Test

Pressure fluctuation is caused by flow interference between the runner vanes and the wicket gate vanes around the runner. Resonance occurs when the natural frequency and vibration mode of the runner in the water match, respectively, the frequency and excite mode of the pressure fluctuation around the runner. This resonance can reduce the strength reliability, especially as the head of the pump turbine becomes high. It is very important to prevent resonance at the rated speed because the stress cycle of the normal operation primarily affects the cumulative fatigue of the runner. Therefore, the natural frequency of the runner should be tuned so that it does not match the frequency of the pressure fluctuation. Unfortunately, the vibration of the runner in a narrow water chamber is much too complex to analyze numerically. Consequently, an actual head model test was conducted to establish a technique for preventing resonance of the runner in water.

The actual head model test satisfies the similarity conditions of both the hydro dynamics and vibration dynamics. The model test equipment is shown in Fig. 4. The former concerns the hydraulic exciting forces and distribution; the geometry of the flow passage from the spiral case to the draft tube is made similar to the prototype likely in the hydraulic performance model. The latter concerns vibration with virtual mass effect; the model runner is made of the same material as the

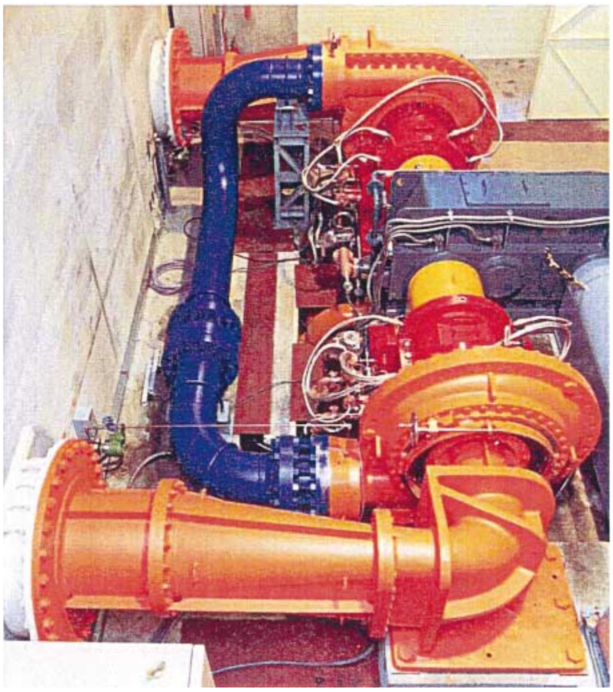


Fig. 4— Actual Head Model Test Equipment.

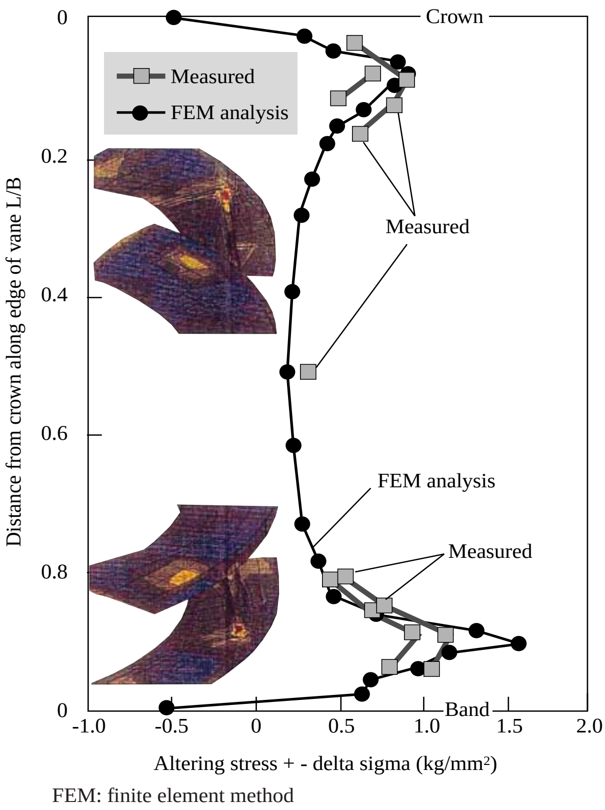


Fig. 5— Alternating Stress on Runner.

prototype and the whole geometry of the runner is precisely similar to the prototype, so that the natural frequency in water of the model is MR (model ratio) times of the prototype.

When the model speed is increased to the MR times of the prototype, the test head becomes the same as the prototype head. Changing the speed, we observed the resonance curve of the stress amplitude. The test showed that the natural frequency in water is approximately 0.5 times that in air.

The measured stress amplitude of the runner agrees well with the stress amplitude of the analysis as shown in Fig. 5. The resonance can be prevented by modifying the stiffness of the runner crown and band and modifying the gap between the head cover and runner crown. Fracture dynamics were applied to determine the allowable defect size for manufacturing highly reliable runners, considering the alternating stress and cycle number when starting and stopping operation as well as under normal operation.

In the manufacturing process of the runner we paid special attention to controlling the stiffness of the runner in order to prevent the resonance.

Main Shaft Seal

The operating conditions of the main shaft seal also become severer when the pumping head and rotational speed increase. The sealing performance and the durability under these severe conditions were tested in the laboratory with an actual size test unit of the main shaft seal. The shaft seal was composed of two carbon seal rings and one resin ring, basically the same as the conventional pump turbine. The carbon seal rings were designed so that at least one of the two rings will perform reliably even if the other ring does not work. A drawing of the test equipment is shown in Fig. 6 (a). It was confirmed that the seal ring can fit to the severely vibrating main turbine, even in load rejection, and performs well.

The shaft seal requires a supply of cooling water when the water is forced below the runner by compressed air in the pump start-up and condenser operation modes. It was also confirmed that there were no problems with the shaft seal when cooling water was not supplied for the duration that the water remained in the cooling water supply pipe.

Wicket Gate Stem Seal

The seal of the wicket gate stem loads the high

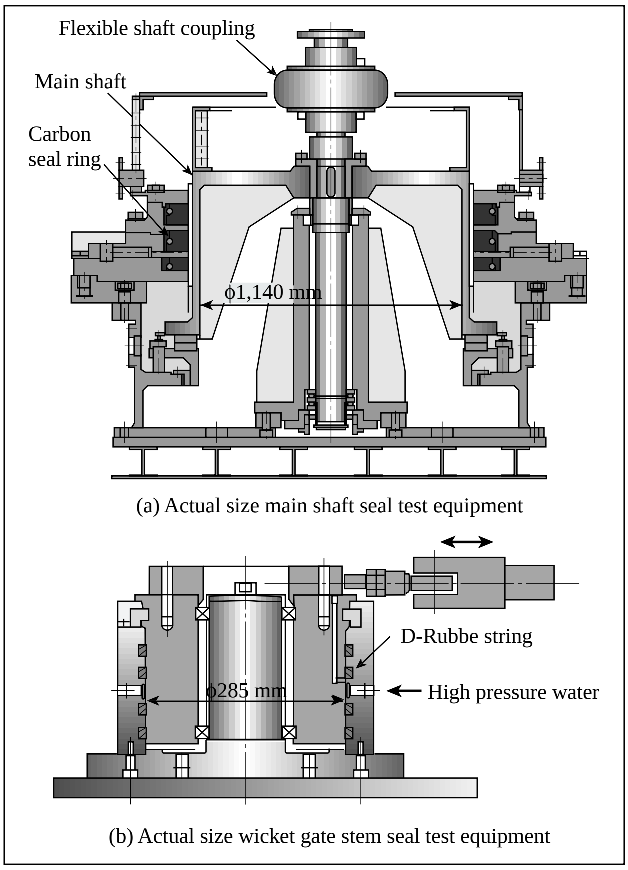


Fig. 6— Actual Size Seal Test Equipment.

water pressure of the load rejection and the pump shut off, and, moreover, that moves as the output power changes. The seal can be omitted if the lower stem of the wicket gate is mounted in the closed pocket of the lower bearing. In this case, however, it seems difficult to make a sturdy stem support in the narrow head cover that can withstand the enormous up thrust and rotational movement of the wicket gate. Therefore, the lower stem is mounted in the open pocket of the lower bearing with the same stem seal as the upper stem of the wicket gate.

In our shop we tested sealing performance and the durability under high pressure and moving conditions of various seal string materials and hardnesses. A drawing of the test equipment is shown in Fig. 6 (b). The D-shaped synthetic rubber string ring with the proper hardness was applied to the packing of the prototype stem seal.

Head Cover - Speed Ring Seal

The seal between the head cover and the speed ring loads the high water pressure and moreover encounters the gap caused beside the groove for O-rubber ring. The gap increases when the operation starts and decreases when the operation stops. If the gap is large, the rubber string can easily move out of the groove during operation, but can not return easily when operation stops. This cycle quickly degrades the sealing performance of the rubber string.

Simulating the actual change and movement of the gap caused at the occasion of starting and stopping the turbine, we tested in the shop the durability of the O-rubber rings made of various rubber materials and having various hardnesses, and plugging rates. The best combination was selected from the sealing performance and the observation of the rubber ring after the durability test.

Also, the maximum allowable gap maintaining the seal over an extended period of operation was determined. Taking into account the gap increase in load rejection, we determined the allowable initial gap of the installation. Therefore, after installing the embedded components, we decided to use machining equipment at the site to remove the waviness and deformation that degrades the sealing performance. Machining at the site ensured that the initial gap at the sealing portion was less than 0.1 mm.

Air Leakage in Condenser Operation

The submergence of the ultrahigh-head pump turbine exceeds 100 m. The air density depressing the water in the upper draft tube during condenser operation exceeds that of the atmosphere by a factor of ten. The runner rotating in high density air induces hard swirls like a storm in the depressed water, which makes numerous air bubbles. These bubbles flow down through the bend of the draft tube driven by the secondary flow of the swirl flow. This is the air leakage in condenser operation. If the rate of air leakage is larger than the air compressor rate, the condenser operation can not continue very long.

The relative rate of the air leakage was calculated for several shapes of the draft tube utilizing the two-phase flow analysis.

Furthermore, an air leakage model test was conducted. The air pressure was the same as in the prototype so that the runner actually made the air leakage with the same mechanism as that of the prototype. The shape of the draft tube with low air leakage was determined, and the rate of air leakage was converted from the model air leakage.

FIELD TEST RESULTS

We conducted a field test, which indicated that the sound level of the developed ultrahigh head pump turbine was lower than that of the conventional one as shown in Fig. 7. The latest technology was therefore considered to have been successfully applied to the fluid and structural designs.

The basic hydraulic performance, such as the turbine output and pump input, agrees well with the converted characteristics from the hydraulic performance model tests. The transient phenomena, such as the change of the speed and the pressure of

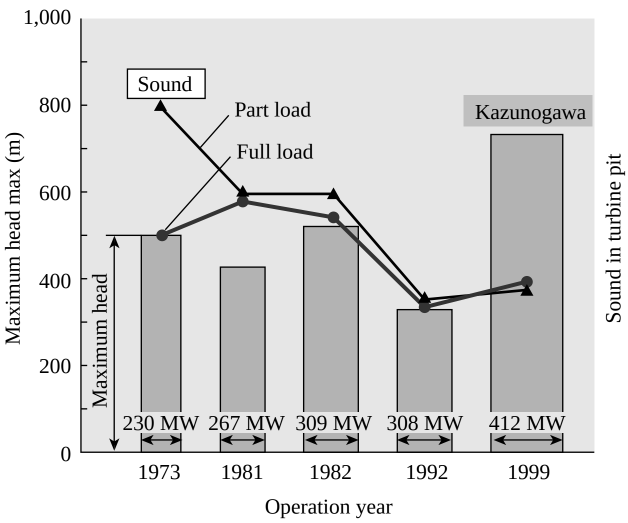


Fig. 7— Head vs. Unit Capacity of Pump Turbines and Sound Level.

the spiral case and draft tube, agree well with the simulation results calculated taking into consideration the four quadrant characteristics of the pump turbine as well as the water way system of the power station.

The rotational speed of the runner resonance (419 rpm), where a very small peak of vibration was observed in the acceleration of the head cover, agrees well with the resonance speed expected from the actual head model test (428 rpm).

The water leakage from the main shaft seal was small. The water leakage from the wicket gate stem seal and the head cover were negligible. The air leakage through the draft tube was so small as to require depressing the air supply for an hour. The input power in the condenser operation was lower than expected. The vibration of the pipe arrangements in the head cover was also very low. These field test results verify the excellence of our design of the ultrahigh-head pump turbine.

CONCLUSIONS

This paper outlines the latest pump turbine technology. It discusses the fluid design, the structural design, and the component tests applied to the development of the ultrahigh-head large-capacity pump turbine. After successfully completing the first unit, we will further improve the hydraulic performance, the performance of the main components and their maintenance and ease of inspection. The results of this development work will be applied to conventional turbines as well as pump turbines.

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