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SCALE EFFECT ON CAVITATION CHARACTERISTICS OF LOW HEAD LARGE BULB TURBINE

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1. INTRODUCTION

Recent trend of developing ultra-low head hydroelectric power plants expedites the application of very high specific speed bulb turbines. Along with the decrease in net head less than 10 m or 5 m, 4-blade or 3-blade runners have been developed to increase the specific speed of bulb turbines. *Table 1* shows the Fuji Electric records of 4-blade and 3-blade bulb turbines recently supplied. Figure 1 reveals the rated head versus specific speed for the records. To develop the high specific speed bulb turbines, wide range of performance characteristics must be confirmed by model tests, as well as by the computerized flow/loss analyses and performance prediction for the optimization of hydraulic design.

In confirming the performance of an ultra-low head large bulb turbine having, for example, the head of 5 m and the runner diameter of 7.3 m by the model test, it is possible to choose the test head to the actual prototype net head. However, since the model runner diameter is 0.365 m which is only 1/20 of prototype one, scale effect must be considered to convert the model performances to the prototype ones. Recently, a standard JSME-S008-1989; "Performance Conversion Method of Hydro-Turbines and Pumps"[1] has been established and published by the Japanese Society of Mechanical Engineers. The new standard contributes to increase the reliability of converting not only the model efficiency but also the model head and discharge characteristics to the prototype performances. However, no conversion method is presented in the standard for the model cavitation characteristics to the prototype ones.

In the horizontal shaft prototype bulb turbine of the above example, elevational difference between top and

Fig. 1 Rated head versus specific speed of 4 & 3-blade bulb turbines

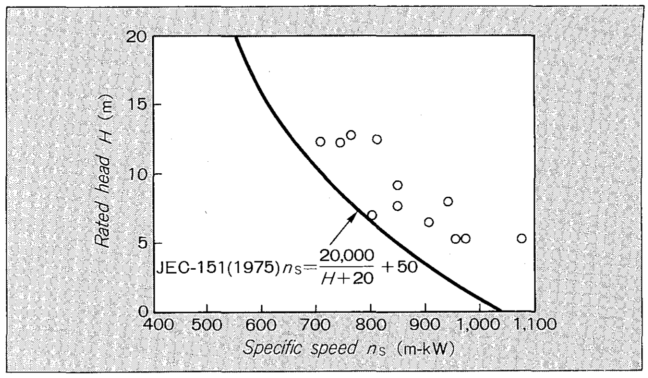


Table 1 Recent records of FUJI ELECTRIC for 4 & 3-blade bulb turbines

Power plant	Country	Unit	Output (MW)	Net head (m)	Runner dia. (m)	Specific speed (m-kW)	Operation (year)
R.B. Parker	U.S.A.	1	2.8	7.3 ~ 4.9	2.50	798	1981
Sakuma Second	Japan	2	16.8	15.5 ~ 11.3	4.49	703	1982
Dawson	U.S.A.	1	4.7	6.7 ~ 3.7	3.87	975	1983
Chungju	Korea	2	6.3	10.3 ~ 6.5	3.30	847	1985
Main Canal Headworks	U.S.A.	1	26.8	12.8 ~ 8.2	5.35	761	1986
Western Yamuna Canal	India	8	9.4	12.8 ~ 11.5	3.15	743	1986
Yuda	Japan	1	5.4	7.8	3.45	848	1986
New Martinsville	U.S.A.	2	20.0	6.4 ~ 3.4	7.30	1074	1988
Lower Mettur	India	8	17.2	9.0 ~ 3.0	6.25	902	1988
Eastern Gandak Canal	India	3	5.7	7.1 ~ 5.0	4.37	956	1990
Oya	Japan	1	3.5	9.0	2.50	810	1990
Teesta Canal	India	9	7.9	11.1 ~ 6.5	4.00	942	Manufacturing

bottom of runner is 7.3 m. There is a possibility that severe cavitations occur at the top of runner, while no cavitation at the bottom. In the model runner, on the other hand, elevational difference is negligibly small, therefore, there is almost no difference in cavitation development from top to bottom of runner. To obtain the cavitation performances of ultra-low head large bulb turbines from the model tests, a conversion method is required to establish considering the scale effect on such an altitudical distribution of cavitations.

A new conversion method of cavitation performances based on the above concept has recently proposed[2] by Fuji Electric and adopted to the JIS code JIS-B8103-1989[3] for model acceptance test of hydro-turbines and pump-turbines. Here, basic concept of the new conversion method is introduced. After summarizing the relevant problems with existing model cavitation tests, the altitudical distributions of the minimum pressure on the runner blade are computed with the three dimensional flow analysis. The distribution of cavitations from top to bottom of the prototype runner is predicted with the computed minimum pressure and the visual observations of cavitations during the model tests. Based on these data, a conversion method of cavitation performance is proposed taking the altitudical distribution of cavitations into consideration.

2. MODEL CAVITATION TEST AND SCALE EFFECT

While the net head and size of a prototype turbine vary with the civil condition and available discharge, a model turbine is limited to its size and test head in which the discharge and efficiency can be accurately measured. As a result, the model test Reynolds number Re' becomes smaller than the prototype operating Reynolds number Re. The Reynolds number is defined as

$$Re = Du/\nu \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where D, u and ν are respectively the reference (runner) diameter (m), peripheral speed at the diameter (m/s) and kinematic viscosity of water (m²/s). Under the condition of the same value for model test head as the prototype net head, the typical values for Re' and Re are respectively

$$Re' = 0.365\ \text{m} \times 24\ \text{m/s}/10^{-6}\text{m}^2/\text{s} = 8.8 \times 10^6$$

$$Re = 7.3\ \text{m} \times 24\ \text{m/s}/10^{-6}\text{m}^2/\text{s} = 1.8 \times 10^8$$

Since the Reynolds number represents the ratio of inertia force to viscous force, the lower Re' means lower model efficiency due to the higher viscous effect. This is the basis for efficiency conversion required to predict the prototype eficiency, and the new formulae are proposed in Reference (1). These formulae can be applied to the conversion of model efficiency measured during the cavitation test.

In order to predict the cavitation phenomena developed in a prototype turbine from its model test, it is necessary not only to operate the geometically homologous model under the hydrodynamically homologous condition but also to observe visually the phenomena under the same cavitation factor σ' as that for the prototype σ. The cavitation factor is

$$\sigma = NPSH/H = \{(TWL - REL + H_a + V_d^2/2g) - H_v\}/H \quad \ldots (2)$$

where $NPSH$ is the net positive suction head (m), H the net head (m), TWL the tail-water level just beyond the draft tube outlet (m), REL the reference level for cavitation factor (m), H_a the atmospheric pressure (m), V_d the areal averaged flow velocity at draft tube outlet (m/s) and H_v the vapor pressure of water (m), respectively.

The numerator of *Eq. (2)*, i.e. $NPSH$, is the difference between the total head at turbine outlet and the vapor pressure. Considering simply if cavitation occurs when the pressure reaches to the vapore pressure, σ means the ratio of head margin against cavitation inception to the net head. By adjusting σ' to coincide with σ during the model test, the similarity for the extent of margin against cavitation can be retained.

The value of σ varies with the selection of reference level REL as shown in *Eq. (2)*. In a prototype, the altitudical difference ΔREL between top and bottom of runner increases with runner diameter, while the altitudical difference per head $\Delta\sigma$ $(= \Delta REL/H)$ still more increases with decreasing net head. In a case with the head of 5 m and runner diameter of 7.3 m, $\Delta\sigma$ reaches even 7.3 m/5 m = 1.5 (the effect of altitudical difference amounts 150% of net head). The effect of altitudical difference results from the gravity and is governed by the Froude number Fr as

$$Fr = u/(g\Delta REL)^{1/2} \propto H/D \quad \ldots\ldots\ldots\ldots (3)$$

If a model test could be done under the Fr' which is the same as the Fr of prototype, $\Delta\sigma'$ could become the same as $\Delta\sigma$. Under the circumstances, the reference level for cavitation factor can be selected at any elevation (at the runner center elevation, typically). IEC supplement code for the model cavitation test[4] recommends to adjust Fr' to Fr for the prototype having the runner diameter exceeding the 25% of the net head.

In order to adjust the Fr' of model which scale ratio is 1/20 of prototype to the Fr, the test head also has to be reduced to 1/20 of 5 m, or 0.25 m. At such low test head, the accurate measurement of performance is impossible. In order to adjust the test sigma to the prototype sigma under such low test head, it is required to adjust the total pressure at the model outlet to –9.3 m, and thus the total pressure at the inlet to –9.05 m. Under the condition, the air entrained in the water will separate and become bubbles which make cavitation test difficult. In conclusion, model tests with coinciding Froude number are practicaly impossible for ultra-low head large bulb turbines.

Summarizing the problems during the model cavitation test for the ultra-low head large bulb turbines, because of unequal Froude number, the difference in sigma from top to bottom becomes almost zero in the model runner. Eventually, the altitudical distribution of cavitation development becomes different between model and prototype. As a countermeasure aginst this problem, when the cavitation phenomena at the top of prototype runner, for example, is predicted by the model test, visual observation has to be done by adjusting the test sigma to the prototype sigma at the top. In other words, the repeated model observations are necessary by adjusting the test sigma to the prototype sigma corresponding to the *REL* at each relevant altitude.

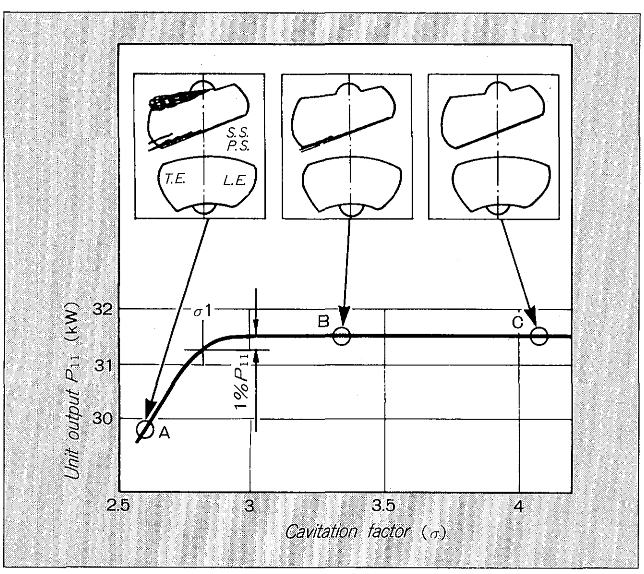
Now the open question so far is the conversion of model cavitation characteristic curve as shown in *Fig. 2* to the prototype curve. *Fig. 2* shows the model test results of a 3-blade bulb turbine around the rated operating condition, illustrating the variation of unit power output P_{11} $\{=P/(D^2H^{3/2})\}$ versus test sigma. At the point-C in the figure, no cavitation is observed in the model runner. When the test sigma is reduced to point-B, the blade tip cavitations occur on the low pressure surface of all blades as shown on the sketch in the figure, yet without effect on the power output. At point-A, cavitations develop as shown on sketch at the tip as well as at the stem of all blades, and power output deteriorates significantly.

In a prototype runner, by contrast, at an operating condition, the cavitations corresponding to the point-A develop when one blade is at the top, the cavitations are decreased to the state of point-B when the blade is turned down at the center elevation, then no cavitation like point-point-C at the bottom. In other words, it is only at the top of runner where cavitations are severe enough to affect the power output. Cavitations on each blade repeats periodically the state of point-A to -C as the runner rotates. No method had been proposed so far to convert the model curve like *Fig. 2* to the prototype performance, taking the altitudical distribution of cavitation into consideration.

3. ALTITUDICAL DISTRIBUTION OF COMPUTED MIN. PRESSURE

An example of the three-dimensional flow analysis through the distibutor and runner for ultra-low head large bulb turbines is shown in *Fig. 3*. Solid lines in the figure illustrates the streamlines solved, and dotted lines

Fig. 2 Model cavitation characteristic curve



are geometrical lines that equally divide the cross area. Since cavitations first occur on the blade surface in the area where the pressure is the lowest, the pressure distribution along the runner blade is analyzed and its lowest pressure is detected. Abscissa in *Fig. 4* shows the length L along the blade from the leading edge which is non-dimensionalized with the total length of blade. The pressure factor C_p on the ordinate expresses the static pressure at each point on the blade surface minus the total pressure at turbine outlet divided by net head as follows

$$C_p = (P/\gamma - P_{T2}/\gamma)/H \quad \ldots \ldots \ldots \ldots \ldots \quad (4)$$
$$\fallingdotseq \{P/\gamma - (TWL - REL + H_a + V_d^2/2g)\}/H \quad \ldots \ldots \quad (4')$$

where P/γ is the pressure head (m) at each point on the blade surface.

If it is supposed that when the minimum pressure P_{min} on the blade surface has dropped to the vapor pressure H_v, cavitation starts to occur at the point

$$C_{pmin} = (P_{min}/\gamma - P_{T2}/\gamma)/H \quad \ldots \ldots \ldots \ldots \quad (5)$$
$$\fallingdotseq \{H_v - (TWL - REL + H_a + V_d^2/2g)\}/H \quad \ldots \ldots \quad (5')$$
$$= -\sigma_{beg} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (5'')$$

By detecting the minimum pressure factor C_{pmin} from *Fig. 4* and changing the sign, the incipient sigma σ_{beg} can be obtained.

As a typical example, *Fig. 4* shows the pressure distribution along three streamlined profiles – hub, middle and tip of a blade. The pressure on the suction surface marked with Δ in the figure becomes the lowest at around the outlet of tip profile, and cavitations start to occur first from this point when the sigma is lowered. With further decreasing sigma, cavitations begin to appear also at the mid-length along the hub profile, and finally, cavitations occur on the middle area of the low pressure surface of blade.

When the flow anlysis as shown in *Fig. 4* is made on many streamlined profiles and the minimum pressure of each profile is drawn against each radial position, *Fig. 5* is obtained. Dashed lines show the altitudical distribution of minimum pressure when the *REL* in *Eq. (5')* is fixed to the runner top. The distribution is symmetric to the rotating axis, and corresponds to the distribution of model

Fig. 3 Computed meridional steamlines through wicket gates and runner

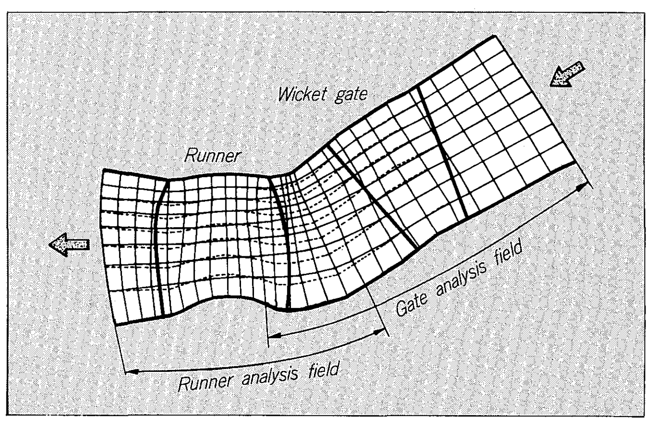


Fig. 4 Pressure distribution along blade profile

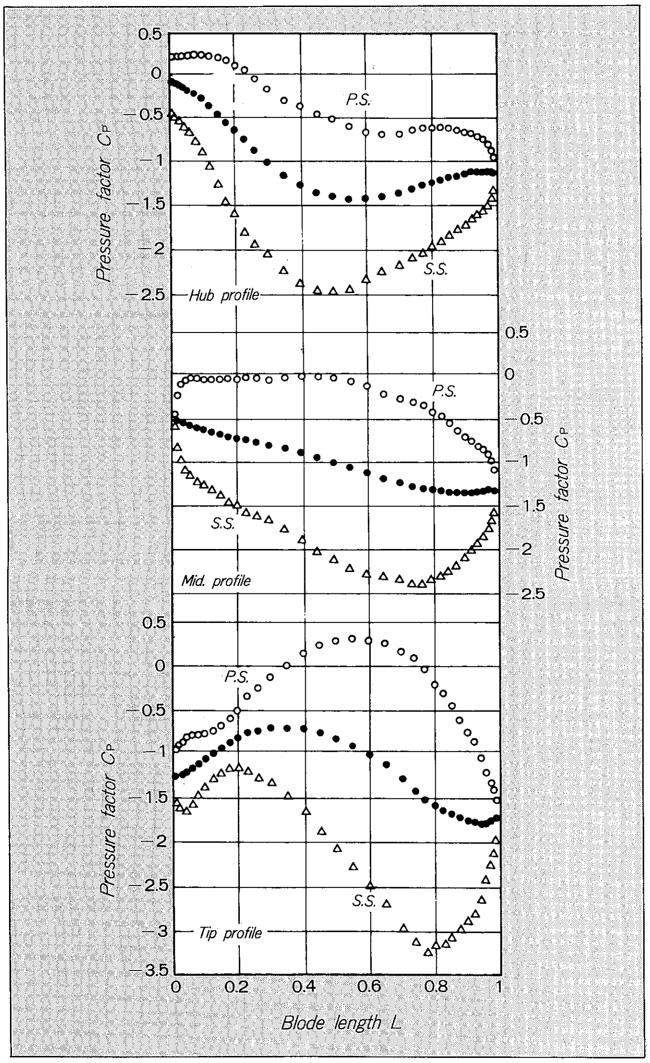
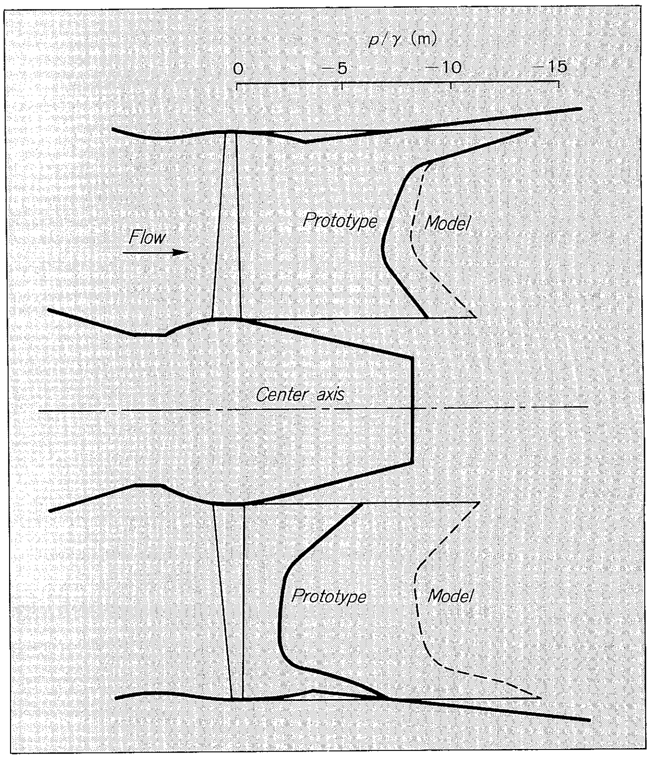


Fig. 5 Altitudical distribution of minimum pressure on blade



runner. On the other hand, the solid lines indicate the distribution of minimum pressure obtained from *Eq. (5')* using the *REL* at the respective position of prototype runner. The prototype distribution is distorted by the effect of altitudical distance. From *Fig. 5*, it can be quantitatively understood that the prototype pressure becomes the lowest at the top of runner tip, followed by the top of hub, and when blades turn downward, the minimum pressure can not be so low.

4. ALTITUDICAL DISTRIBUTION OF CAVITATIONS

By connecting the visual observation results of cavitation phenomena in the model runner at the various sigma as shown in *Fig. 2* to the alititudical distribution of minimum pressure in the prototype runner as shown in *Fig. 5*, the altitudical distribution of cavitation phenomena in the prototype runner can be produced as shown in *Fig. 6*. The figure illustrates the cavitation phenomena at 5 altitudes that are obtained by equally dividing the outer periphery of runner into eight pieces, for example. The prototype cavitation phenomena with local blade performance can be assumed from the model test results corresponding to the respective sigma.

By applying this concept to the model cavitation characteristic curves, a new conversion method to the prototype cavitation performance curves can be derived taking the effect of altitudical distribution as shown in *Fig. 6* into consideration.

5. CONVERSION OF CAVITATION CURVE

A new method to convert the diagram of model cavitation characteristics into that of the prototype can be proposed as follows: In *Eq. (2)*, *TWL*, V_d and *H* are given as constant values under the specific operating condition. Therefore, the effect of a variable *REL* must be considered in cavitation conversion.

(1) Divide equally the outer periphery of prototype runner into eight pieces, for example, as shown in *Fig. 6*, and set the values of *REL* at five altitudes.

(2) Calculate the values of cavitation factor σ at these five *REL*s from *Eq. (2)*.

(3) Read the model performances P_{11}, Q_{11} and η at these five sigmas from the model cavitation characteristic curves.

(4) Convert the model performances to the prototype ones with the methods described in References (1) or (3).

(5) Each blade of prototype runner can be supposed to generate repeatedly with its rotation the respective local performance corresponding to the each sigma. Therefore, the overall performance of prototype runner can be expressed as the mean value of these converted local performances.

Fig. 6 Altitudical distribution of cavitations in prototype runner

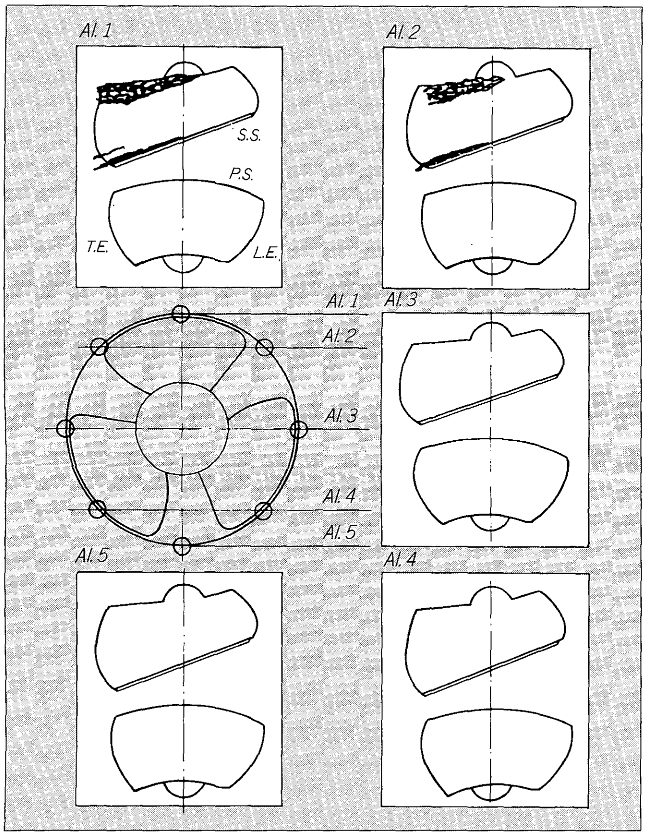
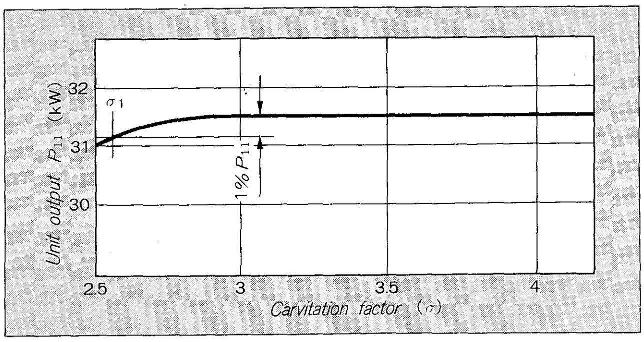


Fig. 7 Converted prototype cavitation characteristic curve



(6) A single blade passes on the altitude #1 and #5 once a rotation, while it passes through the altitudes #2 to #4 twice a rotation. It is reasonable to consider the weight of 1/8 for the performances at the altitude #1 and #5, as well as the weight of 2/8 for those at the altitudes #2 to #4 to obtain the weighted mean performance.

Fig. 7 is the prototype cavitation characteristic curve converted from the model test results shown in *Fig. 2* using the above new method. It is clear from the figure that the deterioration of prototype performance due to the development of cavitations is much smaller than the model curve in the region below the critical sigma where the performance begins to change. For example, the value of σ_1 which is defined as the sigma where the power output deteriorates by 1%, is much smaller in prototype than the σ_1' for model. This tendency can be understood by the fact that the development of cavitations is limited to the upper part of runner due to the altitudical distribution of pressures, and the prototype overall performance is expressed with the weighted mean value of the respective local performance at each altitude.

6. CONCLUSION

A method is proposed which can predict the cavitation phenomena and its altitudical distribution in the ultra-low head large bulb turbine whose runner diameter exceeds the net head, using the model test results and the three-dimensional flow analysis through the runner. Also, a new conversion method of cavitation performances is proposed taking the altitudical distribution of cavitations into consideration. This method is adopted to the JIS code[3] recently revised for model acceptance test.

Reliable prediction of cavitation characteristics is now possible from the model test for ultra-low head large bulb turbines which was hardly discussed before. The concept of this conversion method taking the altitudical distribution of cavitation development into account can be applied to the conversion of cavitating runaway speed characteristics[5] of ultra-low head large bulb turbines which will be explained separately.

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