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IMPROVEMENT OF HYDRAULIC TURBINE EFFICIENCY

Tsuneo Ueda

I. FOREWORD

The sharp increase in the cost of energy in recent years has led to a strong demand for more efficient electric generation and the evaluation of 1% of efficiency has become higher than a half of the price of generating equipment.

Therefore, efforts to make hydroelectric equipment more efficient are continuing. However, since the hydraulic turbine loss is triple the loss of a generator, the demand for better hydraulic turbine efficiency is especially strong.

Whereas the losses in a generator are clearly divided into copper loss, iron loss, stray load loss, and mechanical loss, and techniques for reducing these losses have been established, grasping the flow in the water passage of a hydraulic turbine is difficult and, therefore, separation of the loss of each element is insufficient and efficiency improvement techniques have been developed through the accumulation of experiences based mainly on model tests.

However, grasping of the loss of each element and their reduction technology are being developed through recent advances made in numerical analysis of flow using a large computer and advances in model testing technology, and improved efficiency is expected in the future.

Making turbines and pump turbines more efficient is widely interpreted to mean "effective use of hydraulic energy". Its techniques can be classified as follows:

(1) Making conventional turbines (Pelton, Francis, Kaplan etc.) more efficient (mainly by improving the maximum efficiency).
(2) Using hydraulic energy efficiently by improving efficiency in the off-design points (partial load operation, low head operation), etc. and widening the operatable range.
(3) Increasing efficiency by developing new types of turbines (bulb turbine, etc.).
(4) Use of unused hydraulic energy by developing economical small turbines, generators, and developing extremely low head turbines, generators.
(5) Improving efficiency by increasing the specific speed of high head turbines (especially high head pump-turbines).
(6) Improving efficiency by increasing the unit capacity.
(7) Improving the utilization rate by improving the reliability of the machine.

The present state of turbine and pump-turbine efficiency improvement technology and future topics for (1) through (4) above are discussed.

II. MODEL TEST

Since the internal flow of turbines and pump-turbines is complex, experimantal design methods and repeated model tests with models homologous with the prototype machine have made a substantial contribution to the improvement of efficiency.

Even advances made in numerical analysis of internal flow by large computer and performance prediction technology have not changed the importance of model tests as a means of improving efficiency, especially the efficiency of off-design points.

In the past, the hydraulic turbine has been a high efficiency prime mover, and its efficiency was estimated to have already reached 90% by about 1930. However, since turbine efficiency has reached a maximum of 95% in recent years, the average improvement in efficiency in the past was on the order of 0.1% per year.

To verify this small amount of efficiency improvement, the efficiency measureing accuracy of the model test itself must be on the order of 0.1%.

Therefore, following countermeasures have been taken.

(1) Improvement of the head, flow, output, measureing accuracy of the measurement system.
(2) Stabilization of the test conditions.
(3) Automation of measurement and statistical treatment of data by introducing the computer.
(4) Improvement of model manufacturing accuracy by the introduction of NC machining.

To make the error in scale-up of performance from model to prototype machine as small as possible, the model is made larger, the test head is made higher and the Reynolds number $Re = \frac{D\sqrt{2gH}}{\nu}$ is generally made 1×10^7 or greater (D: Runner exit diameter, H: net head, ν: kinematic viscosity of water)

In the past, model tests were used mainly to verify if the overall efficiency reached the expected value or to check the effect of partial design improvements, that is, as a try and cut tool. Today, however, model tests are used to measure the flow in the water passage and grasp the each loss component to improve the performance prediction accuracy by numerical analysis, and the following special measurement techniques are introduced:

(1) Measurement of the flow distribution, pressure distribution, and loss around the guide vane and measurement of the inlet head of the runner.
(2) Measurement of the flow distribution, pressure distribution at the runner outlet, and draft tube loss by the pitot tube for three dimensional fluctuating flow measurement.
(3) Measurement of the pressure and flow distribution on the runner blade.
(4) Measurement of the internal flow by the tuft method, oil film method, high-speed camera, and other flow visualization techniques.

III. NUMERICAL ANALYSIS OF INTERNAL FLOW AND PERFORMANCE PREDICTION

To further improve the performances of a turbine and pump-turbine, it is necessary to know the internal flow of the components (spiral casing, stay vane, guide vane, runner, draft tube, etc.) and the loss of each component accurately, and then to predict the performances correctly in the design stage. Recently, however, turbine and pump-turbine performance prediction by combining the results of numerical analysis of flow by large computer and experimental measurement of flow and measurement of losses has become possible.[1), (2)]

Numerical analysis of the internal flow and performance prediction of turbines and turbine operation of pump-turbines is performed by the following methods:

1) Runner head

To predict the turbine characteristics of turbines and pump-turbines, first the difference of the runner inlet Euler head Hr_1 and runner outlet Euler head Hr_2 (called runner head Hr hereinafter) must be accurately found. (See *Fig. 1* for a definition of the symbols.)

$$Hr = Hr_1 - Hr_2$$
$$= \frac{1}{gQr}\int_c^b Ur_1 V\theta r_1 dQr - \frac{1}{gQr}\int_c^b Ur_2 V\theta r_2 dQr$$

To accurately find Hr_1, the flow passing through the double circular cascade consisting of the stay vanes and guide vanes is analized quasi-three-dimensionally as a combination of the meridional plane stream and blade-to-blade stream. In this case, the stream line curvature method[(3)] is used for the meridional plane stream line analysis and, since the blades are thick, the singularity method which distributes the vortices on the blade contour[(4)] is used for the blade to blade stream line analysis.

Of course, since the actual flow is different from the value calculated as the potential flow because of the affect of the boundary layer and secondary flow, correction is necessary.

The accurate runner inlet Euler head can be calculated using correction factors.

These correction factors can be obtained from the accumulation of comparisons between calculated value and model test data performed in the past, refering the results of the excellent researches.[(4)]

Fig. 2 shwos the calculated results of the flow at the guide vane outlet of a Fransis turbine. It can be seen from this figure that to correctly find the Euler head of the runner inlet, the flow around the guide vane must be calculated three-dimensionally.

To find Hr_2, the flow in the runner must be analyzed quasi-three-dimensionally[(5),(6)] and the obtained runner outlet flow must be corrected by considering the boundary

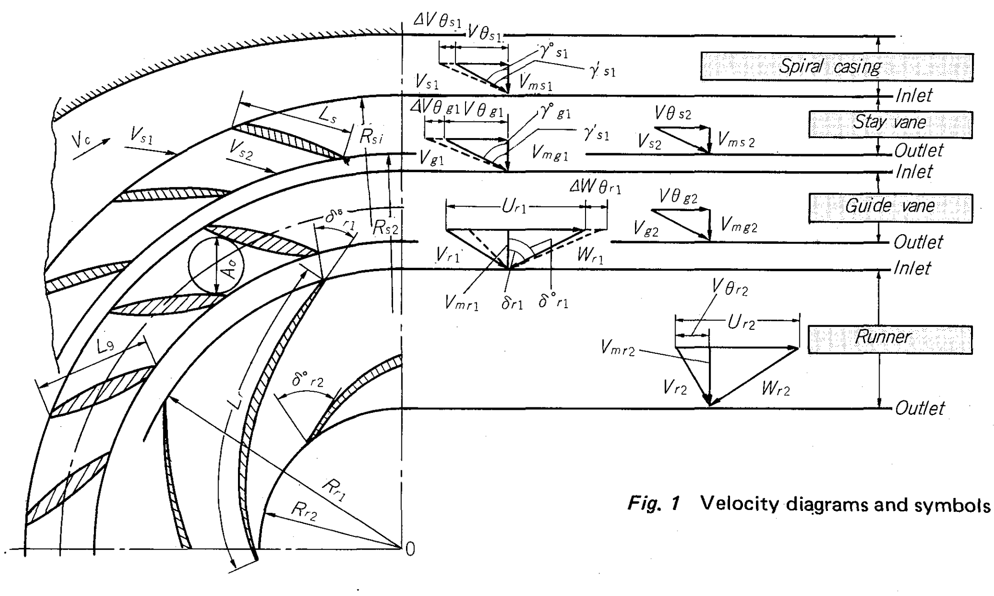

Fig. 1 Velocity diagrams and symbols

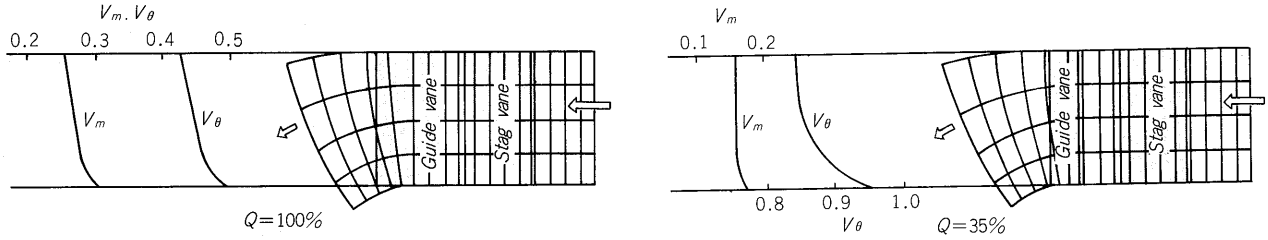

Fig. 2 Flow at guide vane outlet of Francis turbine

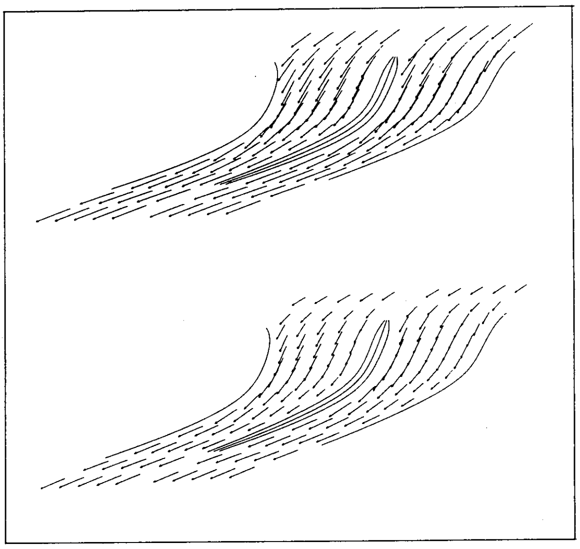

Fig. 3 Flow in runner of Francis turbine

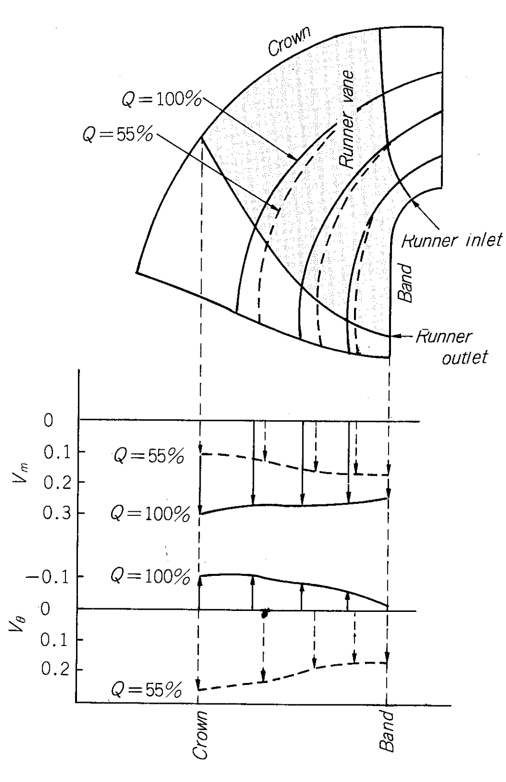

Fig. 4 Flow at runner outlet of Francis turbine

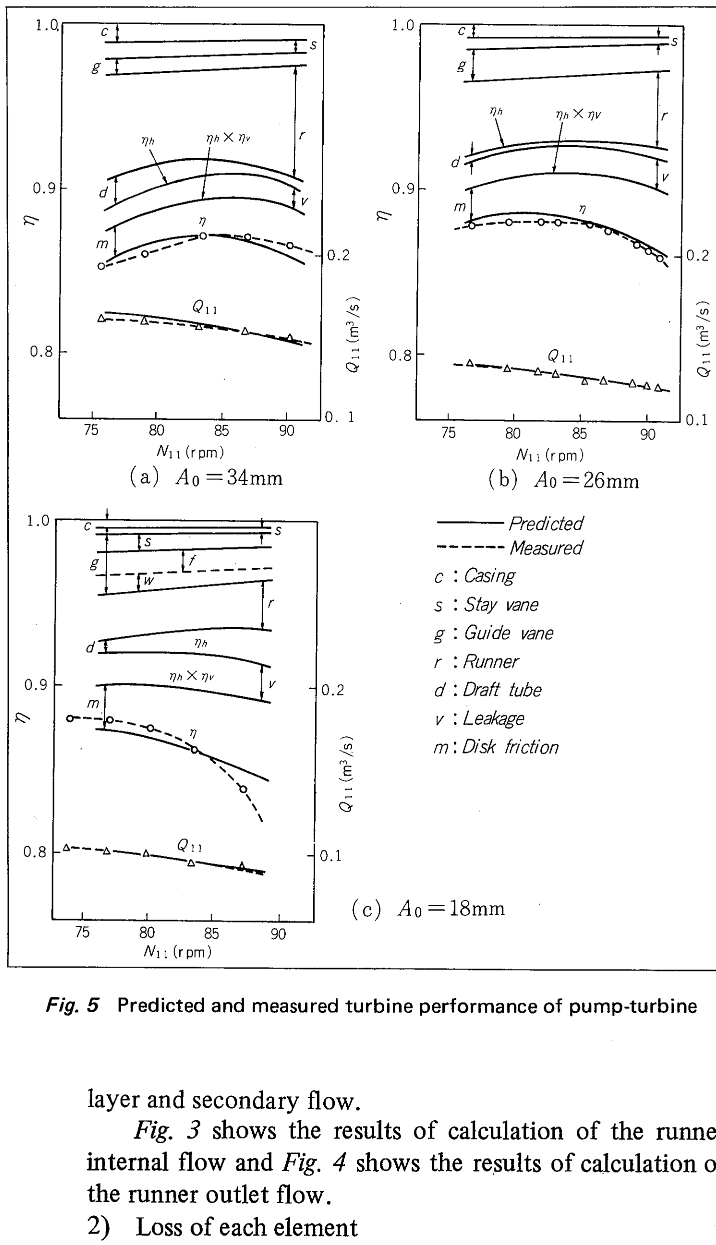

Fig. 5 Predicted and measured turbine performance of pump-turbine

layer and secondary flow.

Fig. 3 shows the results of calculation of the runner internal flow and *Fig. 4* shows the results of calculation of the runner outlet flow.

2) Loss of each element

Various methods of classifying turbine and pump turbine flow losses can be considered.[(1)~(3)] A typical example is described below.

(1) Spiral casing internal friction loss and band loss: ΔHc

(2) Stay vane inlet shock loss, blade surface and wall surface friction loss, secondary flow loss and wake mixing loss: Hs

(3) Guide vane inlet shock loss, vane surface, upper and

lower wall surface friction loss, secondary flow loss and wake mixing loss: ΔHg

(4) Runner vane inlet shock loss, blade surface, crown and band internal surface friction loss, secondary flow loss and wake mixing loss: ΔHr.

(5) Draft tube swirl flow loss, diffuser loss and bend loss: ΔHd

To calculate the stay vane, guide vane, and runner vane loss, the the boundary layer found from the blade and wall pressure distribution and velocity distribution obtained from the results of numerical analysis must be considered.(7), (8)

3) Turbine efficiency

The effective head of a turbine and pump-turbine is given by the following equation:

$$H = H_r + (\Delta H_c + \Delta H_s + \Delta H_g + \Delta H_r + \Delta H_d)$$

Therefore,

$$\eta_h = 1-(\Delta\eta_c + \Delta\eta_s + \Delta\eta_g + \Delta\eta_r + \Delta\eta_d)$$

where, $\eta_h = H_r/H$: hydraulic efficiency

$\Delta\eta = \Delta H/H$: loss head ratio in each component.

Therefore, turbine efficiency is defined as follows:

$$\begin{aligned}\eta &= (\rho \cdot g \cdot H_r \cdot Q_r - \Delta P)/\rho \cdot g \cdot H \cdot Q \\ &= H_r/H - (H_r/H) \cdot (\Delta Q_c + \Delta Q_b)/Q - \Delta P/\rho \cdot gH \cdot Q \\ &= \eta_h - \Delta\eta_v - \Delta\eta_m\end{aligned}$$

Where Q_r is the flow through the runner, ΔP is the friction loss of the runner outer wall, ΔQ_c and ΔQ_b are the leakages from the runner crown seal and band seal, Q is the turbine or pump-turbine flow, $\Delta\eta_v$ is the leakage loss, and $\Delta\eta_m$ is the mechanical loss.

Fig. 5 shows the predicted and measured turbine performance of a pump-turbine at three guide vane openings.

This figure shows that the loss of each component changes with operating condition.(1)

Since prediction of pump performance of a pump-turbine is performed by the same technique as turbine performance, it is not described here.

IV. FLOW NUMERICAL ANALYSIS TECHNIQUE AND PERFORMANCE IMPROVEMENT

Since advances in numerical analysis of flow by large computer and advances in model test technology have made it possible to predict the performances of turbines and pump, that is, efficiency characteristic, discharge characteristics, runner inlet and outlet cavitation characteristics, flow separation, draft tube surging, vibration characteristics, etc., in the design stage, improvement of performance by using these techniques is expected in the future.

1. Optimization of design by numerical experiment

In the past, turbine and pump-turbine performance improvement and design optimization were performed by try and cut method using the model and required considerable development expense and time.

Eliminating repetition of such model tests was the desier of designers for a long time.

Performance prediction by computer answers this desier.

That is, the computer-aided design → flow analysis →

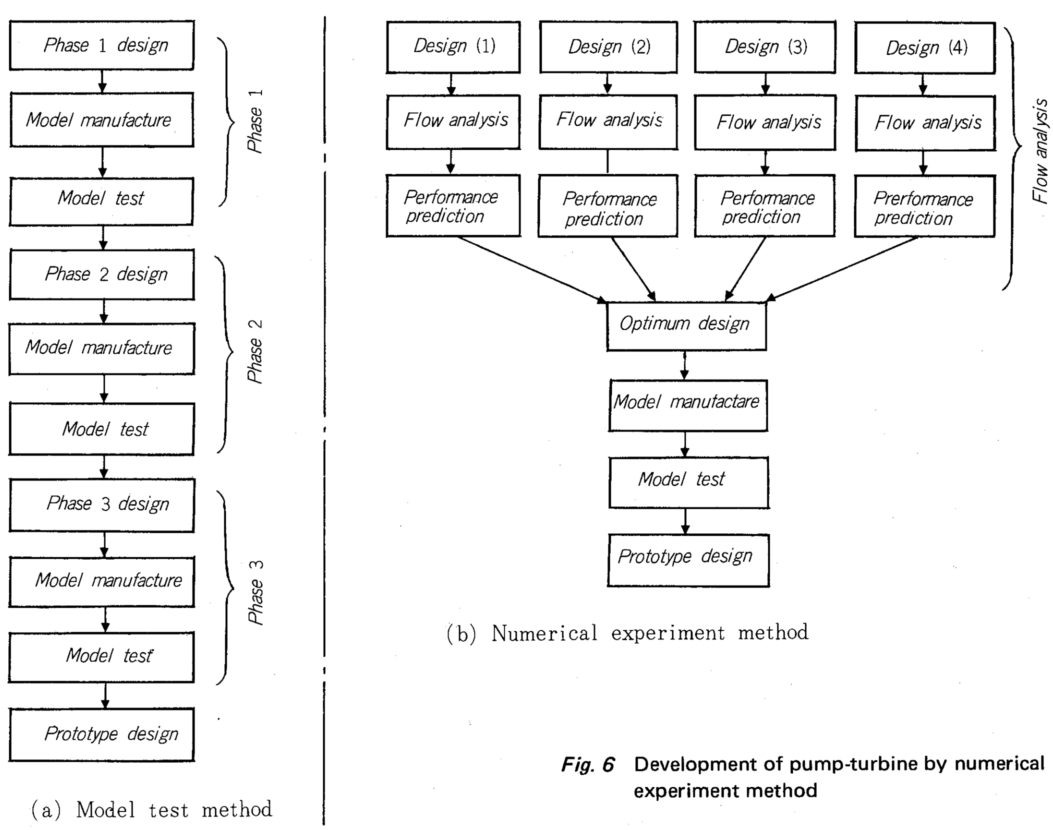

Fig. 6 Development of pump-turbine by numerical experiment method

performance prediction process can be called numerical experiment in the sense that it partially replaces model testing.

In the design stage, many design parameters (runner inlet diameter, outlet diameter, inlet height, runner vane load distribution, etc.) can be changed systematically and performance improvement and design optimization can be performed by numerical experiments.

Of course, after selecting the optimum design by numerical experiment, the performances of the operating region far from the design point that cannot be forecast with present technology must be verified by model tests.

Fig. 6 compares the optimization by numerical experiment method and conventional optimization method by model test.

2. Off-design point performance improvement

In the past, improvement of the efficiency of the design point, that is, maximum efficiency, of turbines and pump-turbines was stressed. However, recently requirement for turbine and pump-turbine performance have become more diverse because of the special character of hydroelectric power generation.

(1) Maximum efficiency is stressed in large capacity generating plants with many units.
(2) Since especially part load operationis frequently performed at run-off river power plants with one unit and power plants that perform AFC operation, part load efficiency is stressed.
(3) When the power plant has a deep flood control storage reservoir, efficiency at low head operationis also stressed.
(4) For pumping-up power plant, balance of turbine performance and pump performance of the pump-turbine and overall efficiency are stressed.

Numerical analysis of flow and performance prediction techniques at the off-design point are very effective in meeting these demands.

For example, as can be seen from *Fig. 7*, since the guide vane secondary flow loss, runner loss, and draft tube swirl flow loss increase at Francis turbine part load, measures to reduce these losses are extremely important in improving the part load efficiency.

Since the flow at the runner inlet and inside the runner is different at full load, maximum efficiency point, partial load, and other operating conditions, and the places where losses are generated and their causes can now be predicted, the turbine efficiency at partial load can be improved without sacrificing efficiency at full load by adopting suitable countermeasures in the funner profile and runner vane design.

Fig. 8 compares the performance of a turbine with improved partial load performance by this method and the performance of a conventionally designed turbine.

Since the efficiency of Francis turbine dropped suddenly and cavitation pitting due to channel vortices appear at part load its operating range was normally limited to 40% or higher output. However, it has been confirmed that the

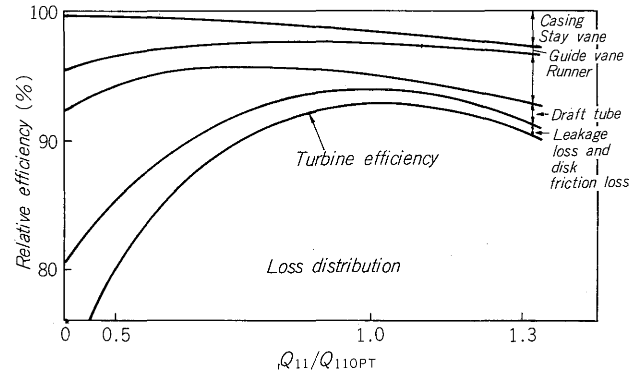

Fig. 7 Loss distribution of Francis turbine

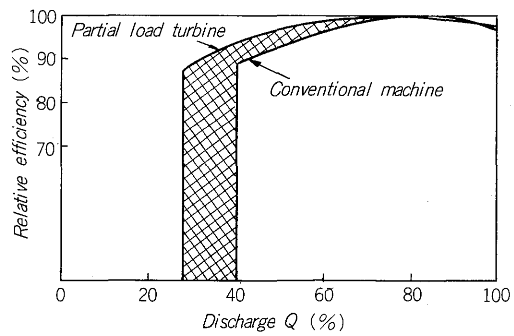

Fig. 8 Improvement of partial load performance of of Francis turbine

new turbines have smaller efficiency drop and there are no harmful cavitations above 25% output.

V. NEW TURBIN DEVELOPMENT

While efforts to improve efficiency of conventional turbines by partial improvement maintaining the basic form is continuing, substantial improvements in efficiency and the use of hydraulic energy not used in the past for economical reasons have been made by developing new types of turbines.

An example of the former is the recent development of the high capacity bulb turbine and an example of the latter is the development of the S-type tubular turbine.

1. Development of the bulb turbine

Since the vertical shaft Kaplan turbine has a considerable bend loss in the water passage, its improvement in efficiency is limited and this trend is more pronounced, for high specific speed Kaplan turbine having a 20 m or lower head, since the velocity per unit-head becomes very high.

The bulb turbine was developed to smash through this limit. However, recent advances in bulb turbine generator have made the generator smaller and the shape of the water passage has become a linear shape close to the ideal and the efficiency far exceeds that of the vertical Kapaln turbine.

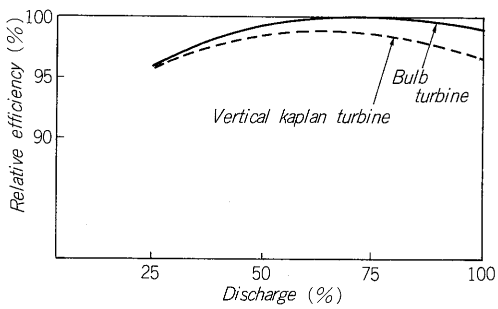

Fig. 9 Comparison of bulb turbine efficiency

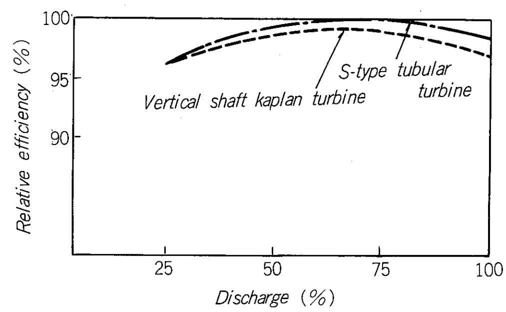

Fig. 10 Comparison of S-type tubular turbine efficiency

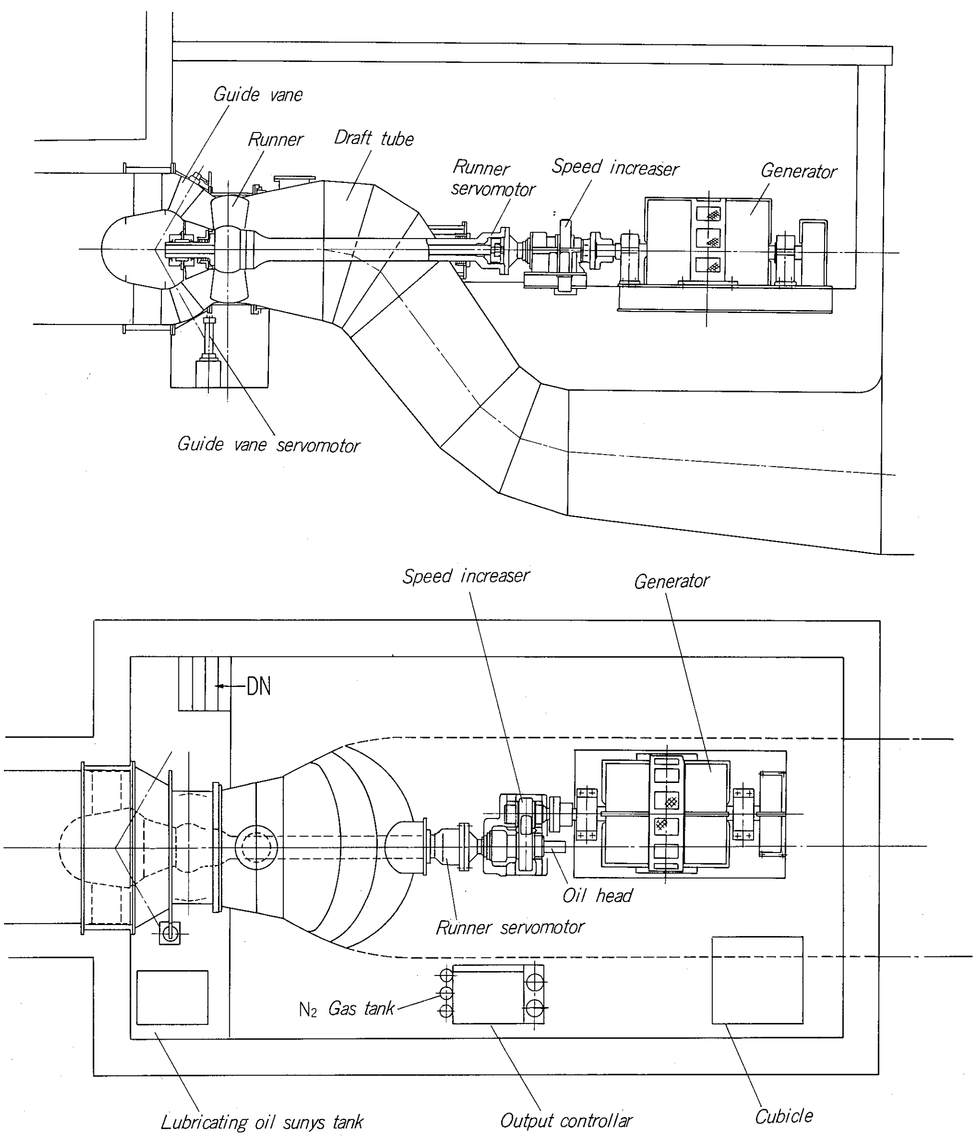

Fig. 11 Layout of S-type tubular turbine

Fig. 9 compares the efficiency of the bulb turbine to the efficiency of the vertical shaft Kaplan turbine. It can be seen that the efficiency of the bulb turbine is superior to that of the Kaplan turbine, especially at high flow.

Since the velocity around the guide vane and runner vane must be made as high as possible to make a bulb turbine smaller, accurate analysis of the flow at these parts is indispensable.

2. S-type tubular turbine

The generator of an S-type tubular turbine is placed outside the water passage to lower costs and facilitate maintenance. The efficiency of the S-type tubular turbine is somewhat lower than that of the bulb turbine, but better than that of the vertical shaft Kaplan turbine and it will probably be used in small capacity hydraulic power generation where the head is 20 m or less. *Fig. 10* compares the efficiency of the S-type tubular turbine and vertical shaft Kaplan turbine. *Fig. 11* shows the layout of the S-type turbine.

VI. REDUCTION OF THE LOSS AT EACH COMPONENT

1. Runner

In the past, runner design was stressed in improving the efficiency of the turbine and pump-turbine. However, since the precision of analysis of the flow at the runner inlet and in the runner has been improved, the runner losses have been reduced.

There is little margin for improvement of efficiency of design points by runner improvement only. However, a large margin for improvement of the efficiency of the off-design points is expected in the future.

2. Spiral casing and stay ring.

The casing and stay ring losses of high specific speed turbines is especially high.(7) Therefore, research is being performed on the flow in the spiral casing(8) and on the shape of the casing inlet to improve the flow.

Fuji Electric was one of the first to focus its attention on the low-loss parallel type stay ring and has used it instead of the conventional bell mouth type stay ring ever since 1965.

However, the loss at the parallel stay ring also differs with the shape of the inlet.

3. Guide vane

Since the the loss of the double circular cascade consisting of stay vane and guide vane is large at turbine partial load, especially at partial loads on high head pump-turbines having thick blades for strength, research is being conducted on optimization of the shape and angle of the guide vane and optimization of the circumferential relative position of the stay vane and guide vane.(4)

4. Draft tube

Most of the loss of the elbow type draft tube is caused by the upper draft tube and bend part. Therefore, from the standpoint of efficiency, the inlet velocity of the bend should be reduced by making the inlet cone longer through the use of a upper draft tube having the optimum cone diffuser angle (8 to 10°). The cone angle of the upper draft tube should be made larger to weaken the draft tube surge. Research is being conducted on eliminating the drop in efficiency up to a cone angle of about 15° when there is a swirl flow.(12)

Since efficiency, excavation, and draft tube surging are closely associated and the relationship between efficiency and operating conditions changes with the draft tube shape, the optimum shape of the draft tube is different for each power generating plant.

Since the efficiency can be improved more when a cone type draft tube is used than for an elbow type draft tube, there are also examples where a horizontal shaft Francis turbine was used.

VII. CONCLUSION

To the hydraulic designer, of the manufacturer runner design has been more an art than a science. The elements of science have become more numerous with recent advances in flow analysis technology.

The authors wish to thank engineers of the Kansai Electric Power Co. for there assistance regarding the development of flow numerical analysis and performance prediction technology, and the development of the design optimization technique by computer, and the engineers of Power Development Corp. and New Energy Foundation for the development of the partial load Francis turbine.

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