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DERIAZ TYPE PUMP-TURBINE FOR KUROMATAGAWA No. 2 POWER STATION, ELECTRIC POWER DEVELOPMENT CO., LTD.

By Tsuneo Ueda,
Shigeo Kuroyanagi and
Suminori Yamazaki
Design Dep't., Kawasaki Factory

I. PREFACE

The Deriaz type pump-turbines have been adopted for the Kuromatagawa No. 2 Power Station, Electric Power Development Company, because of a pumped storage power station having such great head variation that the minimum head becomes 50% of the maximum head and also because of the high efficiency required for partial loads. The plant is now being erected with utmost effort being exerted to put it into operation in November, 1963. Since these are record making machines in Japan and in the world for their great output and for their high head, the Deriaz type pump-turbine with its excellent performance has been successfully developed through much research and many tests. The machines have many special features in construction and control, adopting 2 speed operation, coupled with a pole change generator-motor in order to obtain more superior performance. We wish to introduce the outline of the equipment.

II. OUTLINE OF POWER STATION AND WATER TURBINE

The Kuromatagawa No. 2 power station is a dam and conduit type power station having a maximum output 16,100 kw, which takes the water from the Kuromatagawa No. 2 reservoir through a circular pressure tunnel and penstock of 3.3 m inner dia. about 200 m length and discharges the water into the Kuromatagawa No. 1 Dam through a tail race pressure tunnel about 140 m length.

Further, from the view point of increasing the total generated energy and utility of the equipment during the dry season etc., this is used as a pumping-up power station.

Because of the draft head attained −9.5 m. the building is made in circular form having an inner dia. 14 m below the floor of the generator air duct in order to minimize the excavation.

The specifications of the pum-pturbine are as follows.

Type	Vertical shaft movable vane Deriaz type pump-turbine
Turbine operation	
Net head	78/73/39 m
Discharge	28/28/20.7 m^3/s
Output	19,200/18,000/6500 kw
Rotating speed	300 rpm
Specific speed	189 m-kw
Pump operation	
Net pump head	80/75/41 m
Pump discharge	21.7/23.9/16.5 m^3/s
Pump input	19,900/20,000/7900 kw
Rotating speed	333/300 rpm
Specific speed	63.5 m-m^3/s

III. PERFORMANCE

1. Model Test

Before manufacturing the actual turbine, various kinds of model tests were carried out to obtain a pump-turbine with a revolution as high as possible

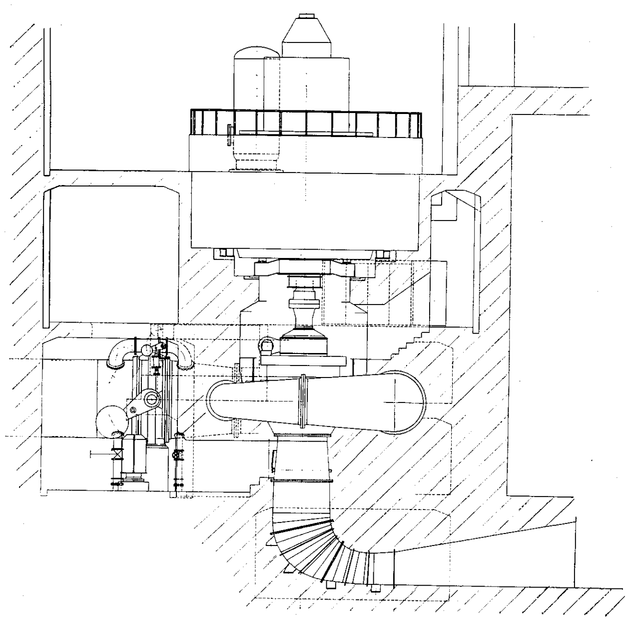


Fig. 1 Cross section of power house

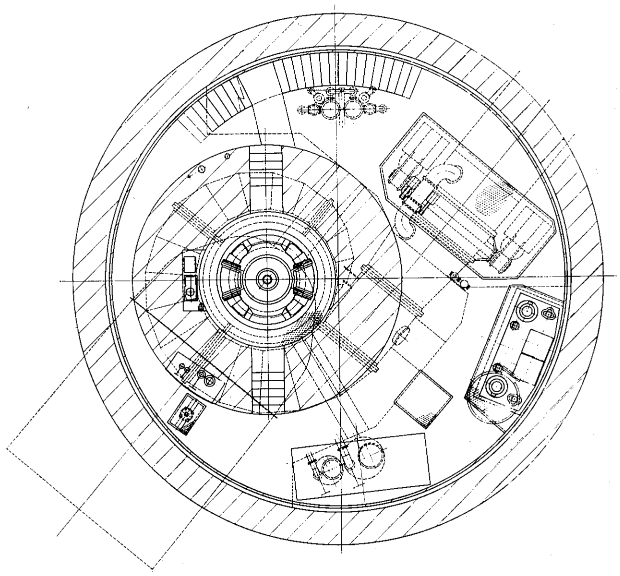


Fig. 2 Plane of power house

to satisfy the following conditions.

(1) The normal head weighted mean efficiency should be higher than that of the Francis turbine.

(2) The draft head should be within −9.5 m.

(3) The efficiency should be sufficiently high even at partial loads.

(4) The capacity of generator-motor should be determined by the requirement of the turbine.

The results of model tests exercised will be described briefly as follows:

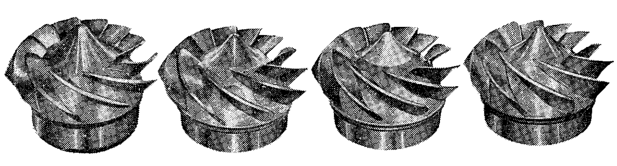
1) Casing

During the period of design, discussions were held as to whether the casing should adopt diagonal flow type like the Sir Adam Beck-Niagara Power Station or radial flow type, and finally the radial flow type was adopted because of its simple construction and easy assembly though it was hydraulically disadvantageous for its slight decrease of efficiency at a large flow when operated as a turbine.[1) 2)]

2) Specifications of model tests

The model tests were carried out for 4 kinds of runners *A, B, C* and *D*, each having a different conical angle, boss ratio and blade form. The numbers of blade were 10 and runner basic dia. was 450 mm. *Fig. 3* shows the model runners.

The turbine test was carried out at a head of 12 m and the pump test at a normal head of about 40 m.



(A) (B) (C) (D)

Fig. 3 Model runners

Table 1 **Comparison of the greatest efficiencies for 4 kinds of runners**

		A	B	C	D
Turbine operation	Efficiency (%)	88.7	89.2	89.7	89.3
	Unit revolution (%)	97.6	100	100	101.3
	Unit discharge (%)	92.3	99	100	95.1
Pump operation	Efficiency (%)	88.7	88.7	88.2	88.3
	Pump head (%)	110	111	100	114
	Pump discharge (%)	108	99	100	99

Table 1 shows the results of efficiency tests for the turbine and pump. After discussions, runner *C* was finally adopted.

3) Turbine Characteristics

Fig. 4 shows turbine characteristics of runner *C*, taking unit speed as abscissa and unit discharge as ordinate.

Fig. 5 shows turbine efficiency for various discharges and *Fig. 6* shows the same for various heads,

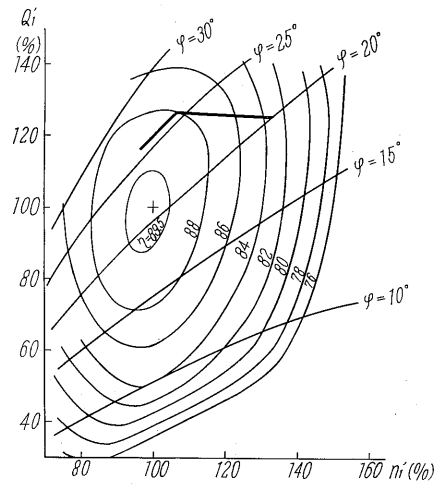


Fig. 4 Turbine characteristics of model C

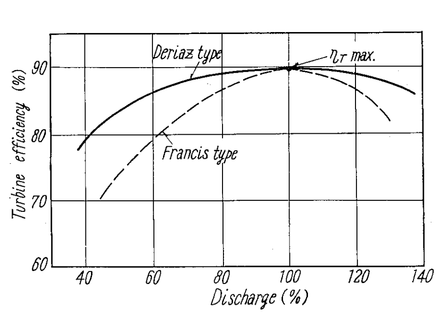


Fig. 5 Turbine efficiency of Deriaz type pump-turbine compared with Francis type pump-turbine

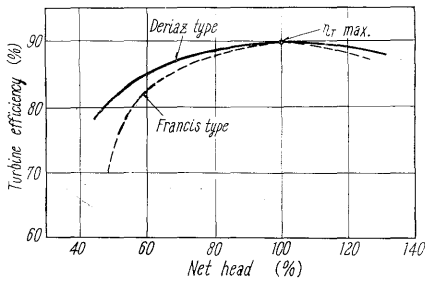


Fig. 6 **Turbine efficiency of Deriaz type pump-turbine compared with Francis type pump-turbine**

both abscissa being compared with the Francis type pump-turbine.

4) Pump Characteristics

Fig. 7 shows pump characteristics of runner *C*. The characteristics, in the case of a constant angle of runner blade, are quite similar to the ordinary Francis type pump turbine and the maximum efficiency at each runner blade angle and minimum cavitation coefficient are nearly the same, while the discharge and head decrease according to decrease of the blade angle. Accordingly the efficiency can be shown by the full lines in the drawing and the cavitation coefficient can be illustrated by the dotted line as a contour line. The cavitation coefficient σ in the drawing is converted into that at the normal head. In the drawing, within a boundary at the left side above σ min. line, cavitations occur from the back surface of blade edge and within a boundary at the right side below σ min. line, cavitations occur from the front surface of the blade edge.

With variable head if the turbine operates close to σ min. line by controlling suitably the angle of the runner blade, change of σ is very small, which is smaller in case of the low head than in case of the high head. Also differing from the Francis type pump turbine, the pump discharge can be widely changed by suitable control of the runner blade angle.

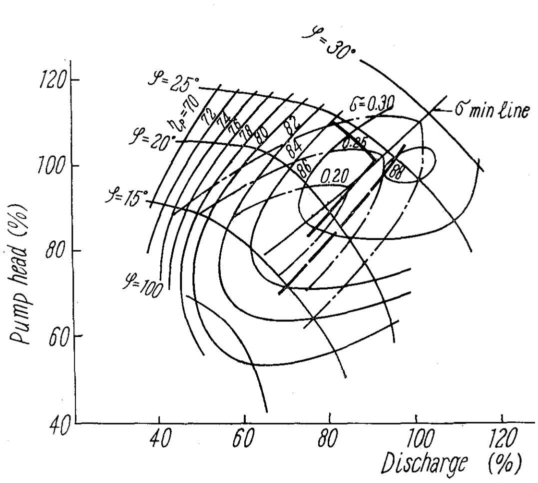


Fig. 7 **Pump characteristics of model C**

5) Special tests

(1) The tests on efficiencies affected by runner gaps were carried out. *Fig. 8* is the test result, which shows that the influence due to the runner gap is great, so that it must be seriously discussed to decide the runner gap.

(2) With regard to the hydraulic thrust that has a great effect on the value of runner gap, detail tests were carried out for both turbine operation and pump operation.

(3) For the purpose of capacity determination of the servo-motor, a measurement for moments of the runner blade and guide vane in turbine operation and pump operation was carried out, thus it was confirmed that due to a very small difference between opening and closing moment, the axis positions of the blades were suitable to be selected. *Fig. 9* shows

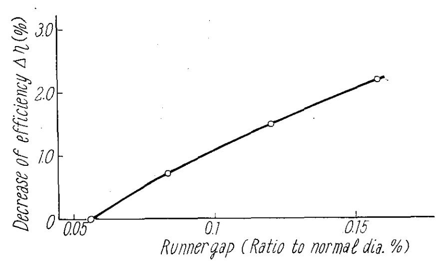


Fig. 8 **Efficiency drop versus runner gap**

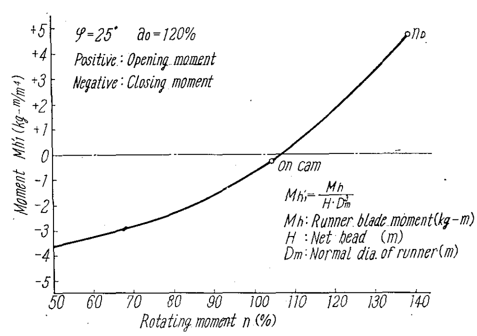


Fig. 9 **Runner vane moment in turbine operation**

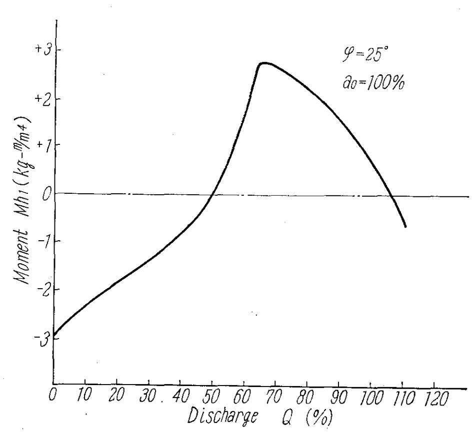


Fig. 10 **Runner vane moment in pump operation**

one example of the moment of the runner blade at the time of turbine operation, *Fig. 10* at the time of pump operation.

2. Characteristics of Actual Machine

(1) Unit discharge at the maximum efficiency of the Deriaz type pump turbine during pump opepation is, in general, larger than that during turbine operation, so the pump input has a tendency to become larger than the turbine output. Therefore after analytical discussion on cavitation, efficiency etc., the motor capacity shows a some what larger value than that determined by turbine output.[3]

(2) The ratio of unit speeds at maximum efficiency for turbine operation and pump operation is about 1.1, the same as the Francis type pump turbine. However 2 speed being adopted in order to improve the efficiency at the time of low head of turbine, it was decided to operate at 333 rpm at the time of pump high head, 300 rpm at the time of turbine operation, and 300 rpm at the time of pump low head.

(3) *Fig. 11* shows the turbine characteristics.

The thick line in *Fig. 4* is a line corresponding to the turbine maximum discharge. Also the pump characteristics are shown in *Fig. 12.* The continuous thick line in *Fig. 7* is a line corresponding to 333 rpm operation and the thick dotted line 300 rpm operation.

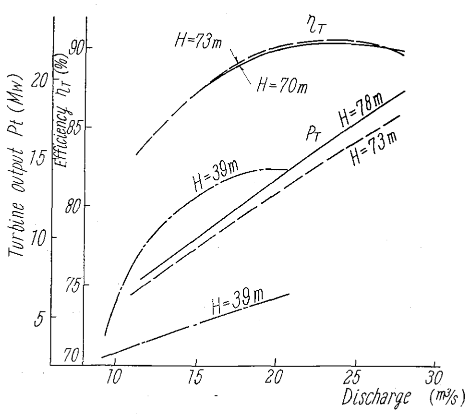


Fig. 11 Turbine performance

3. Operation System

1) Because the pull-in-torque can be made less than 10% of the rated value by fully closing the runner blade, no water level depressor like that of the Francis type pump-turbine is necessary.

The starting shall be effected by the following process.

(1) The runner blade is fully closed.

(2) By lifting main of rotating weight by the magnetic thrust bearing, the static friction of thrust being is reduced.

(3) 1/2 of the rated voltage is applied in order to limit the rush current within the rated kva.

(4) After acceleration, it is changed over to the full voltage during asynchronous operation.

(5) The generator is pulled in synchronism by applying dc excitation.

(6) The runner blade is opened and then the guide vane is opened.

2) During pump operation, it is operated at 333 rpm above a head of 60 m and 300 rpm below a head of 60 m. However the variation of upper water level depends upon the season and the variation range of tail race head being 2 m., change over of speed is not carried out during operation, but the speed is automatically changed over by detection of a difference between upper and lower level at the time of starting the pump. At the same time, the pump operation cam being changed over, the openings of the runner blade and guide vane are automatically settled in the most appropriate condition for the head.

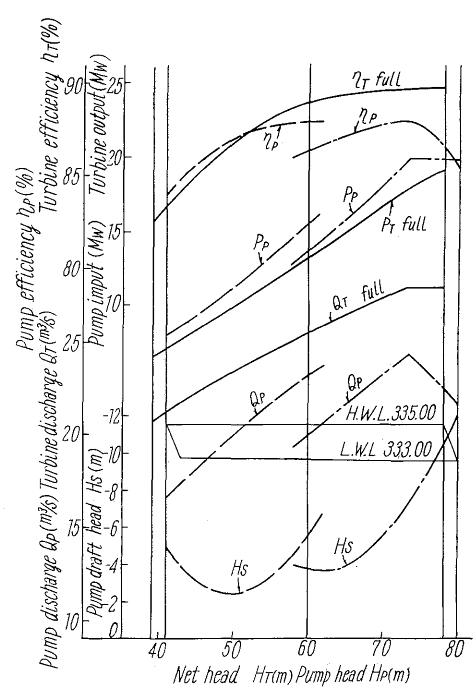


Fig. 12 Turbine and pump performance

4. Transient Phenomena of Pump

By discussion of the transient phenomena at the time of motor trip, it was confirmed that there was found no danger of water column separation of penstack even in the case of non-operation of the runner blade and guide vane. The relation between the guide vane closing time at the maximum head and minimum head and the transient phenomena were also discussed for determination of guide vane closing speed.

Since it is, in general, considerably complicated to obtain graphically the transient phenomena at the time of motor trip, especially in this power station, where the penstock reflection time is very small compared with the time constant of generator-motor, the transient phenomena in this power station were obtained by solution of the following simultaneous equation by means of the digital computer.

$$T_1'=f_1(Q_1') \quad \cdots\cdots\cdots\cdots\cdots\cdots (1)$$

$$M=h\cdot T_1' \quad \cdots\cdots\cdots\cdots\cdots\cdots (2)$$

Equation of moment of inertia,

$$\Delta n=-374\frac{M_n}{N_n\cdot GD^2}\cdot m\cdot \Delta t \quad \cdots\cdots\cdots (3)$$

Equation of water hammer $(t=itr)$

$$q_i=\frac{g\cdot H_n}{a\cdot v_n}\left\{h_i-h_0+2\sum_1^{i-1}(h_i-h_0)\right\}+q_0 \quad \cdot\cdot (4)$$

$$Q_1'=\frac{q}{\sqrt{h}} \quad \cdots\cdots\cdots\cdots\cdots (5)$$

$$n_1'=f_2(Q_1') \quad \cdots\cdots\cdots\cdots\cdots (6)$$

$$h=\left(\frac{n}{n_1'}\right)^2 \quad \cdots\cdots\cdots\cdots\cdots (7)$$

Where,

H=Total head (m), n_1'=unit speed

Q=pump discharge (m^3/s)

Q_1'=unit discharge, N=speed (rpm)

T_1'=unit torque, M=Torque (t-m).

$h=\frac{H}{H_n}$, $q=\frac{Q}{Q_n}$, $m=\frac{M}{M_n}$

a=Velocity of pressure wave (m/s)

tr=Penstock reflection time (s)

Suffix n denotes a basic condition

In the above formulae (1), (6),

$T_1'=f_1(Q_1')$, $n_1'=f_2(Q_1')$

are complete pump characteristics and obtainable by experiment. For example it is expressed by the curve as shown in *Fig. 13* for openings of the guide vane and runner blade at the minimum head. Because of difficulty in function approximation, the precise piece-wise linear approximation was made. In the part below point p in *Fig. 13*, T_1', n_1' become many-valued functions, so that the above calculation process must

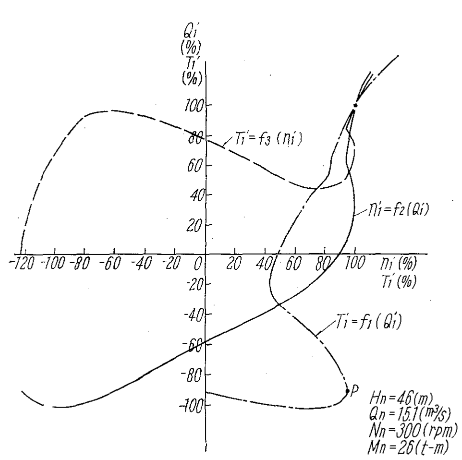


Fig. 13 Completed characteristics at minimum pump head

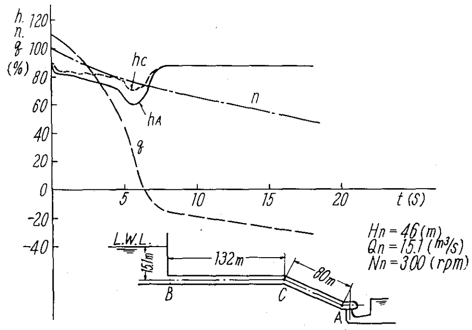


Fig. 14 Transient phenomena in case of pump trip

be changed, using the relation $T_1'=f_3(n_1')$, $Q_1'=f_2^{-1}(n_1')$ in *Fig. 13*.

Fig. 14 is a result calculated when the guide vane does not function at the time of minimum head. The pressure drop at point C, where water column separation is most likely to occur in the penstock, is 19%, that is 8.8 m, which shows no danger of water column separation.

IV. CONSTRUCTION

The present pump turbine has the largest capacity of any ever manufactured in Japan as a Deriaz type pump turbine. Therefore special consideration has been given to the constructional design.

1. Pump-turbine

The construction of the pump-turbine is shown in the sectional drawing in *Fig. 15*.

Since the runner is the most important item in a Deriaz type pump-turbine, from the point of view of construction which includes many new problems, great attention was paid to the construction and materials used, so that all important problems were successfully solved by the experiments. The runner blade and runner boss have been made of 13% chromium stainless steel. The runner blade, as the shaft is inclined, gets bending stress due to centrifugal force during rotation besides axial directional tensile stress, compared with the runner blade of an ordinary Kaplan type turbine, to which special consideration must be paid in design. Hence, in order to know the actual stress during operation, the stresses on various parts of the rotating blades were measured by the strain meter. Moreover, after the actual runner was manufactured, by rotating it at above the maximum speed, the stresses of each part were measured. Thus its safety was assured.

The servo-motor for runner blade was adopted, the same type (up and down system) used in the ordinary Kaplan type turbine. The operating mechanism of the runner blade provides, as shown in *Fig. 18*, one more angle lever between the operating lever mounted on the blade stem and an operating disc mounted

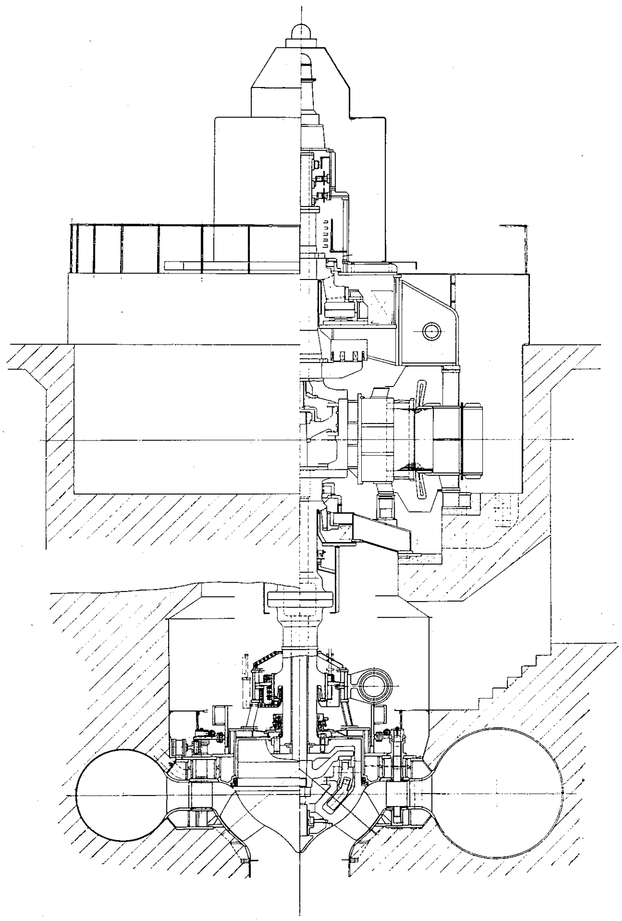


Fig. 15 Sectional view of Deriaz type pump-turbine

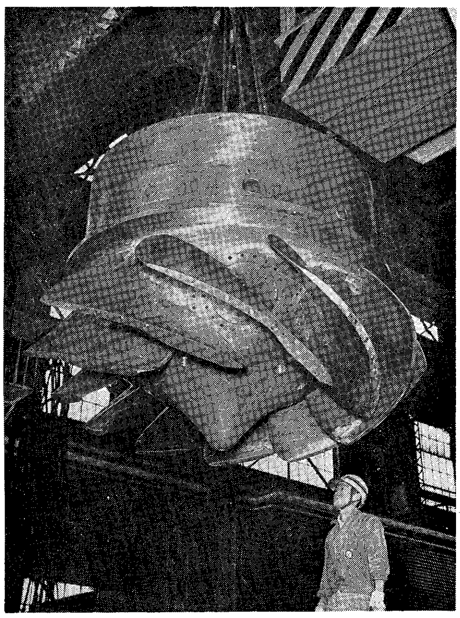


Fig. 16 Deriaz type pump-turbine runner

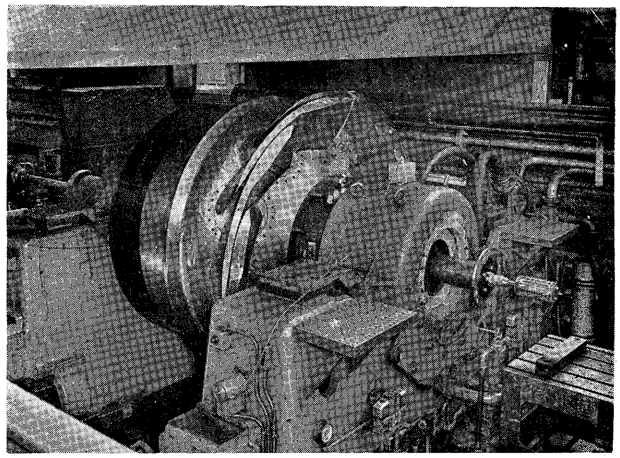


Fig. 17 Rotating test of runner

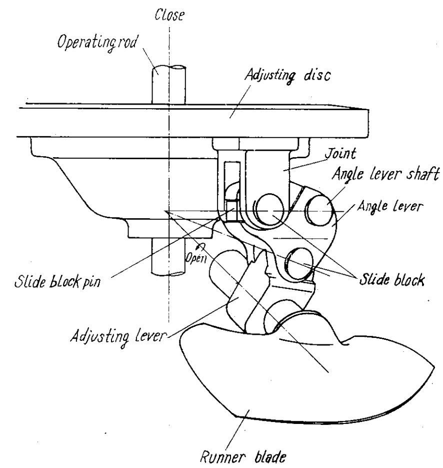


Fig. 18 Operating mechanism of runner vane

on the operating rod, and the respective lever is equipped with a slide block, enabling three dimensional motion. It is well understood that this mechanism makes a very smooth motion geometrically, if the shaft center lines for angle lever and slider block pin are made to cross at one point. However, further, a model of reduced scale was made to assure the motion of this mechanism. At the same time the assembling and disassembling process was studied. Because this mechanism transmits power through the slider block, a slip is always produced between the slider block and its fork formed lever. If any clearance should occur due to friction between them, the runner blade is likely not to move smoothly; therefore the material must be carefully selected. For this purpose, a model of link mechanism having the same size and the same material as the actual one was made and connected to the oil pressure servomotor to bear the same load as the maximum load during the actual operation. Then the withstand operation test of 20,000 strokes was undertaken, in which the condition of friction was inspected every 5000 strokes and the result proved quite scccessful, and even after a 20,000 strokes test, no abnormality was found. Also it was ascertained that friction was intimated with the initial slip, abrasion developed very little later.

The packing device for the shaft of the runner blade used the same V formed rubber packing as the ordinary Kaplan type turbine and the pressure device of lubricating oil inside the boss adopted the same standard system as the Kaplan type turbine, in which a constant oil pressure, higher than the external water pressure, is always maintained by the operating oil pressure of the servo-motor of the runner blade. Besides the runner blade operating mechanism, various parts such as runner boss, runner boss cover, operating disc etc. were also carefully considered in design and at the same time they were applied with a maximum load practically encountered under manufacturing or after completion. Thus the stress and strain were measured to assure safety in use.

Since the guide vane is used for both turbine and pump, special consideration was paid to its form. It is equipped with a guide vane braking device so as not to cause vibration of the guide vane due to pulsation of hydraulic pressure at the time of pump operation. This device, as shown in *Fig. 20*, provides one oil pressure servo-motor for every two guide vanes, the force of which is amplified by the link mechanism to push the guide vane shaft. This guide vane braking device is put into operation after the opening of the guide vane is settled at a specified value during pump operation and when changing the opening of the guide vane, at first, the braking device is released and then the opening of the guide vane will be changed. Accordingly special consideration is paid to the guide vane braking device in order to release the brake quickly when it is needed to close the guide vane suddenly.

The servomotor of the guide vane adopted is a ring servomotor used in all our Kaplan type turbines and some of large output Francis type turbines and is mounted on the main bearing stand arranged in a very compact space.

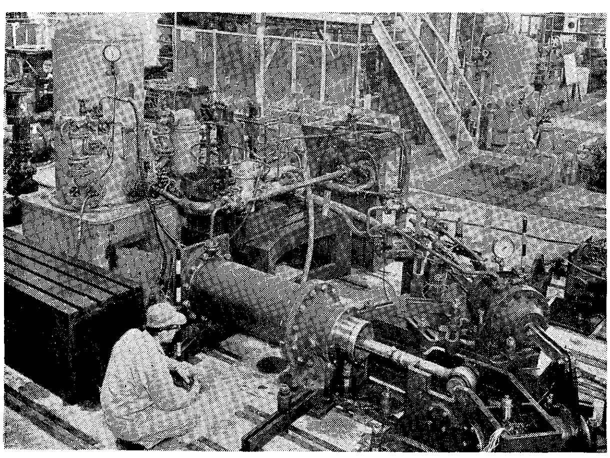


Fig. 19 Wear test of operating mechanism of runner vane

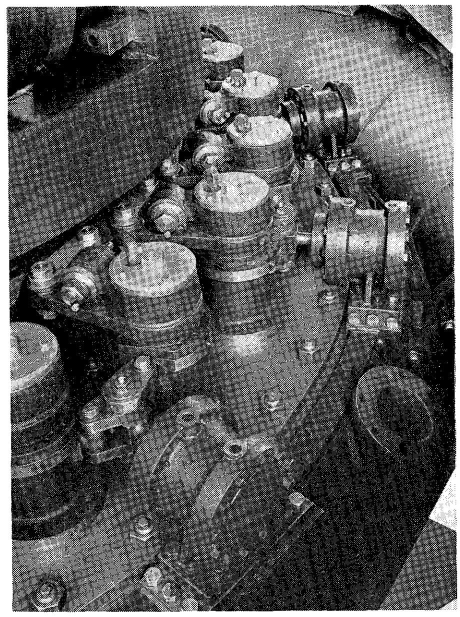


Fig. 20 Guide vane brake device

The casing consists of one steel welded fabrication together with the speed ring, divided into 4 sections of flanged coupling system.

The discharge ring is made of welded steel plate and covered with 18–8 stainless steel for its inside surface against the outer periphery of the runner blade. This inside surface is precision machine-finished.

The clearance between the outer periphery of the runner blade and the discharge ring is preferably as narrow as possible and they must not come into contact with each other even in case of an accident. To insure this clearance, various conditions were considered, these were elongation due to hydraulic thrust and weight, elongation due to temperature rise, elongation due to revolution etc., all of these factors being calculated and then compensated by the factory test and site test, so that the necessary clearance for operation is finally adjusted to a minimum. Moreover, in order to detect abnormal lowering of the runner due to burning down of thrust bearing metals etc., a magnetic induction system of shaft down detector and also a measuring device by dial gauge are provided between the pump turbine shaft and the generator shaft.

2. Inlet Valve

The inlet valve is a butterfly valve having an inlet dia. of 2500 mm, equipped with a butterfly valve having an inlet dia. of 400 mm as a bypass valve. Both main and bypass valves have a counter weight closing system so as to close automatically, should the oil pressure fail. The counter weights are fixed on the lever ends of both sides and the servo-motor is so mounted upon the middle of the lever as to push up the lever from the lower side.

3. Governor and Control Device

In the Deriaz type pump-turbine, as in the ordinary Kaplan type turbine, a control device is naturally necessary in order to maintain the opening of the guide vane and that of the runner blade in a predetermined relation for both turbine operation and pump operation.

However, usually there is a difference in the relative position of the runner blade opening to the guide vane opening between both operation times, turbine and pump, and yet the operation sequence for the guide vane and runner blade must be changed hereby. Accordingly, the control device becomes considerably complicated.

In the present power station, special attention was paid to the above to make the control device as simple and yet reliable as possible. The control device consists of an actuator, 2ry distributing devices for the guide vane and runner blade, opening setting devices for the guide vane and runner blade for pump operation. Above all the opening setting device which is an essential part, adopted a simplified and extremely reliable pure-mechanical system (refer to *Fig. 22*). The actuator is Fuji standard Model EA61, electrical-oil pressure type, the same as used in the ordinary turbine when in turbine operation. However, it is used only as a slow opener of the guide vane for pump operation. For the opening setting device for pump operation, a rotating shaft to be able to deviate along its axis according to the apparent head is provided, and the guide vane opening and runner blade opening are independently controlled by means of each opening setting cam mounted on the rotary shaft.

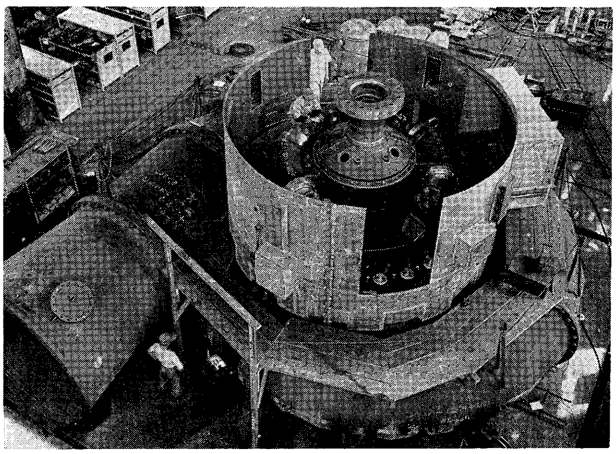


Fig. 21 Shop assembly of pump-turbine

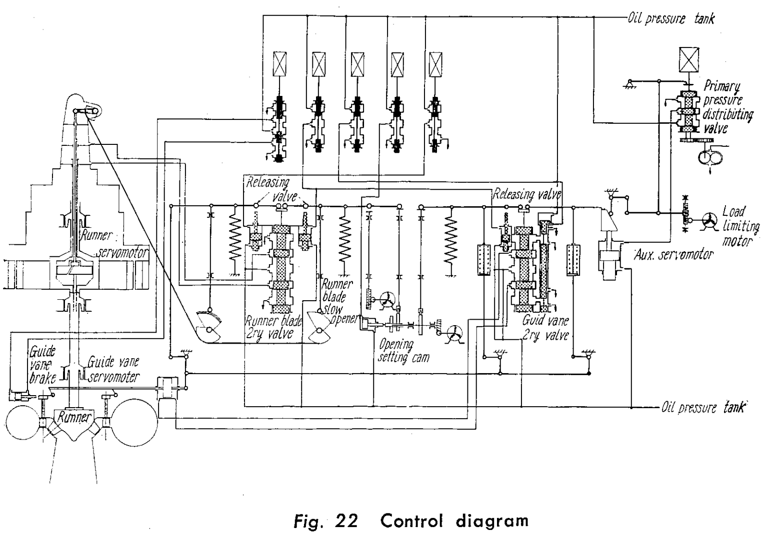


Fig. 22 Control diagram

In addition to the above the opening setting is equipped with the runner blade slow opener, changing over device for the runner blade opening setting cam according to 2 kinds of revolution at the time of pump operation etc.

Each 2ry distributing valve of the guide vane and runner blade is operated by a different mechanism for either turbine operation or pump operation and the mechanism for the unused side is so made as to cut off by the valve of the releasing device.

The control system for turbine operation and pump operation will be as follows.

1) Turbine operation.

During turbine operation, since for the pump operation mechanism the releasing valve releases the interconnection for the 2ry valves for the guide vane and runner blade, these 2ry valves are controlled only by the turbine operation mechanism. Therefore, during turbine operation, the motion of the auxiliary servomotor controlled by the primary distributing valve is transmitted to the 2ry distributing valve of the guide vane through the turbine operation mechanism and the motion of the guide vane servo-motor is restored to 2ry distributing valve of the guide vane by the restoring mechanism and at the same time it is transmitted to the runner blade control mechanism to move the runner blade.

That is, during turbine operation, the guide vane opening governs the runner blade opening in both closing and opening process.

In this case, the runner blade makes an opening corresponding to the head by so-called solid cam provided upon the restoring mechanism of the runner blade servo-motor, thereby achieving a high efficient operation.

2) Pump operation

During pump operation, the pump operation mechanism is so positioned as to be able to control each 2ry distributing valve. Also the turbine operation mechanism for the runner blade is cut off by the other releasing valve.

When starting as pump operation,

it becomes necessary to close the runner blade nearly to the full closure in order to reduce starting torque. For this purpose the runner blade is once closed almost to the full closure by the runner blade slow opener. At that moment the guide vane is maintained on the full closure position by the load limiting device similar to turbine operation.

The runner blade is gradually opened by the slow opener on condition that the generator-motor begins to start and the speed of the pump attains to a certain value, and then stops at an opening settled by the setting device.

The speed (300 rpm or 333 rpm) in pump operation is decided according to the head before starting the pump and either of two kinds of cam shall be selected for use, depending upon the speed.

Upon condition that the runner blade has been opened to a predetermined opening, the guide vane is opened up to a settled value of setting cam by the load limiting device. That is, in the case of pump operation, the load limiting device of actuator acts as a slow opener of the guide vane.

The guide vane, after being opened at a specified opening, is fixed by the guide vane braking device.

When stopping the pump, the guide vane is closed by the load limiting device after the guide vane braking device is released. In this case the runner blade is not closed but kept at the opening during operation in order to obtain a great braking torque.

That is, during pump operation, the guide vane opening is preceded by the runner blade opening in opening process, but the runner blade opening is preceded by the guide vane opening in closing process.

V. CONCLUSION

We have introduced and explained an outline of the Deriaz type pump-turbine for the Kuromatagawa No. 2 Power Station. Since this pump turbine is a quite up-to-date kind of machine, if its characteristics and construction should be greatly improved in the future, we believe that the disadvantage in its cost compared with the Francis type can easily be overcome.

The writers will be pleased if this article can be of any help to those who are working in the planning of new hydro-electric power plants.

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