IAHR AIRH

XVII IAHR SYMPOSIUM
Beijing, China 1994

IRCHMB

ACCURATE DETERMINATION OF VOLUMETRIC AND MECHANICAL EFFICIENCIES AND LEAKAGE BEHAVIOR OF FRANCIS TURBINE AND FRANCIS PUMP - TURBINE

JUNICHI KUROKAWA Professor, Yokohama National University, Japan
TAKAYA KITAHORA Asistant, Yokohama National University, Japan

ABSTRACT

In order to determine the volumetric and the mechanical efficiencies of Francis turbine and Francis pump-turbine, the flow in the front and the back spaces of a runner equipped with several thrust balancing devices, such as balancing pipes and balancing holes, is analyzed theoretically. The flow characteristics in sixteen types of model Francis turbine and twelve types of model Francis pump-turbine of different specific speed are calculated, and mechanical and volumetric efficiencies are expressed as a function of specific speed. The influence of several dimensions of thrust balancing devices on those efficiencies are also determined together with the behaviour of axial thrust.

RESUME

L'écoulement dans les espaces avant et arrière d'une roue équipée de dispositifs d'équilibrage d'effort, tels des tuyaux et des orifices d'équilibrage, est analysé théoriquement, de manière à déterminer les efficacités volumétrique et mécanique d'une turbine de modèle Francis, ainsi que d'une turbo-pompe de modèle Francis. Les caractéristiques de l'écoulement sont calculées pour seize types de turbines Francis et douze types de turbo-pompes Francis, avec différentes vitesses spécifiques, et les efficacités volumétrique et mécanique sont exprimées comme fonctions de la vitesse spécifique. L'influence sur ces efficacités de la dimension de différents dispositifs d'équilibrage est également analysée, ainsi que le comportement de l'effort axial.

1. Introduction

The performance of water turbines have largely improved recently and the overall efficiency today is over 94% in Francis and Kaplan turbines. It is then required to make accurate determination of a leakage loss and a disk friction loss and to minimize them. These losses are caused by the gap flow at the back of a runner, and cannot be measured individually, because of the difficulty in separating these losses from the measured data which include many kinds of losses.

Hence, the designers are obliged to predict these losses by a theoretical method. However, the gap flow is very complicated, because many types of thrust balancing device and leakage sealing system are adopted in an actual water turbine. As the thrust balancing devices and leakage sealing systems adopted and also the method of predicting those losses are different depending on the manufacturing companies, the data of these losses so far reported have a large scatter when plotted against a specific speed n_{SQ}[1]. It is then difficult to evaluate these losses as a function of the specific speed for the standardization of the performance evaluation, such as the scale-up formula of the JSME Standard-008 "Performance Conversion Method for Hydraulic Turbines and Pumps(1989) ". In order to evaluate these losses accurately, it is necessary to develope an accurate prediction method which is applicable to any types of water turbines, and to apply the method to all the data so far published.

One of the present authors has long studied about the leakage and disk friction in radial flow turbomachinery[(2)-(6)] and revealed that these losses vary largely depending on the flow pattern in the back spaces of a runner. The present study is aimed to establish an accurate method of predicting leakage loss and disk friction loss applicable to any types of Francis turbine and Francis pump-turbine, and to determine the volumetric and the mechanical efficiencies over the wide range of the specific speed.

2. Nomenclature

$C_q = (\Delta Q/2\pi\, r_{1C}^3\omega)(r_{1C}^2\omega/\nu)^{0.2}$; leakage parameter (ΔQ ; leakage flow-rate)
$C_M = M/(\rho r_{1C}^5\omega^3/2)$; friction torque coefficient (M ; friction torque)
$C_T = T/(\rho\pi r_{1C}^4\omega^2)$; non-dimensional axial force on runner back surface (T ; axial force)
d ; diameter of D.T. balancing pipe, C-B balancing pipe or balancing hole[mm]
D ; reference diameter of Francis turbine or Francis pump-turbine[mm]
n_{SQ} ; specific speed expressed in [m,m^3/s,rpm]
r_1 ; radius of runner periphery [mm]
$Re = D^2\omega/\nu$; reference Reynolds number
δ ; seal gap width[mm]
$\Delta C_T = C_{TC} - C_{TB}$; coefficient of downward axial thrust caused by gap flow
Δp ; pressure drop from runner inlet to outlet [Pa]
η_v ; volumetric efficiency
η_m ; mechanical efficiency
ν, ρ ; kinetic viscosity and density ($\gamma=\rho g$) [m^2/s],[kg/m^3]
ω ; angular velocity of runner[1/s]

Subscripts
B, C = at the band side and the crown side, respectively
i, o = at the inner radii and the outer radii, respectively

3. Outline of Present Theory and Calculation Procedure

3.1 Outline of the theory

The leakage and the disk friction of a water turbine is caused by the gap flow at the back of a runner. Generally, gap flow is of two types. One is the axial gap flow between a rotating disk and a stator such as that at the back of a runner, and the other is the annular gap flow such as that at an annular seal. In the former, centrifugal force is dominant, while in the latter, wall shearing stress is dominant.

One of the present authors has proposed a gap flow analysis for both hydraulically smooth and rough wall cases[2][3] and also for both wide and narrow gap cases[4]. These theories are based on the equations of continuity, momentum and angular momentum and the results were confirmed to give accurate estimation of the flow characteristics by comparison with the actual measurements[6]. It was revealed that the pressure and the velocity distributions of the gap flow are largely influenced by the flow-rate and pre-whirl velocity of the leakage entering into the gap.

Gap flow analysis then requires accurate determination of the boundary values of the flow velocity and the pressure at the entrance and the exit of all the gaps. In applying the theories to actual machines, the determination of boundary values is especially important and the method is presented in the reference[6].

Among many boundary values of gap flow in an actual machine, the pressure and velocity at the entrance and the exit of a runner/impeller and at the exit of balancing devices, such as balancing holes and balancing pipes, have a dominant influence on the leakage characteristics. Here those boundary values are predicted by a quasi-three-dimensional analysis of a runner flow.

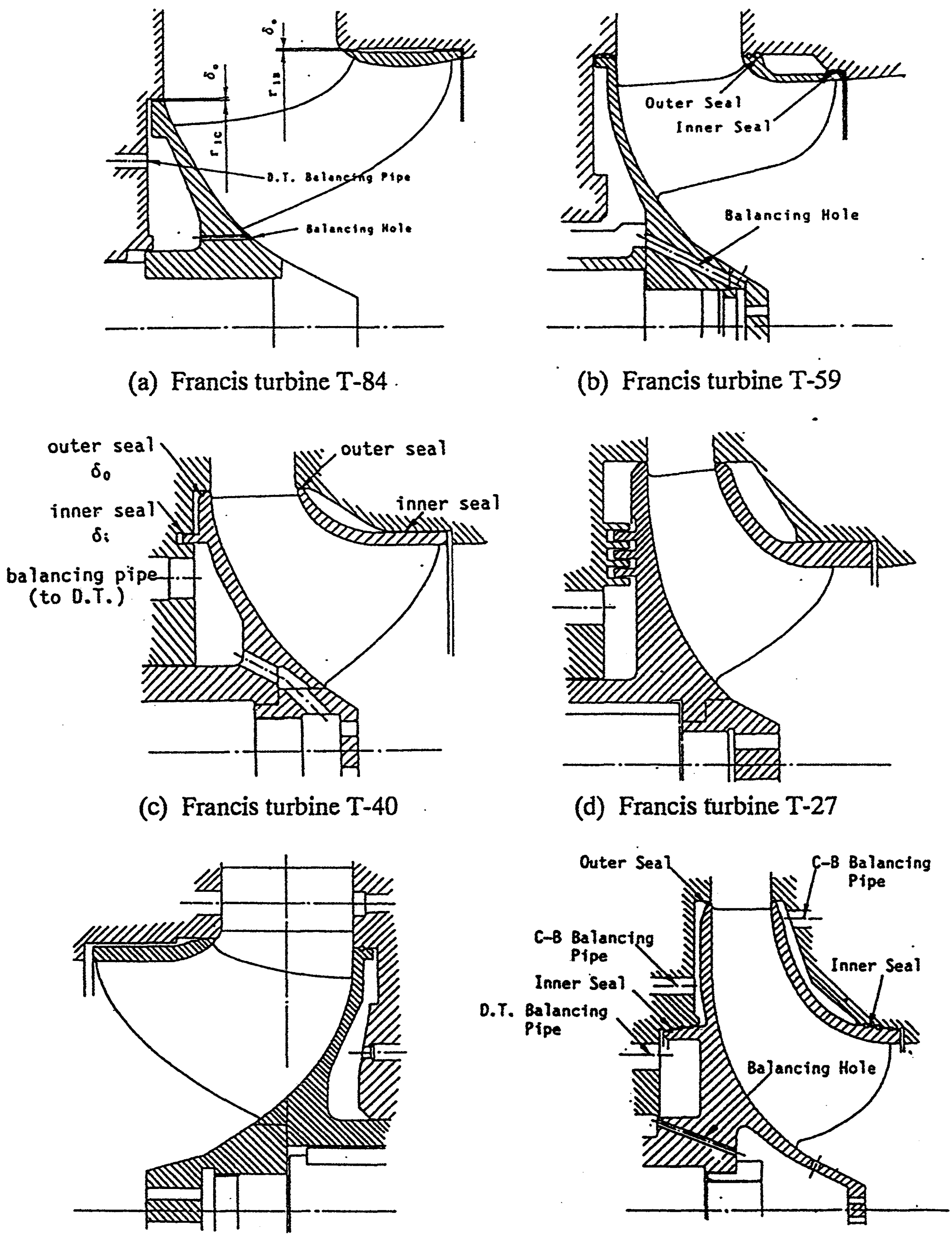
3.2 Calculation Procedure

Assuming the leakage flow-rates in the crown and the band side gaps, the flow characteristics in both side gaps are determined as a function of radius by solving the equations of continuity, momentum, and angular momentum, which also yield the boundary values of the pressure at the entrance of all balancing devices and seal gaps. Using the boundary values of the pressure, the second approximations of the leakages through all the balancing devices are determined. A trial-and-error method is thus needed to determine the higher order approximation of leakage.

As many kinds of thrust balancing devices are adopted in an actual machine, such as balancing holes, balancing pipes connecting the crown side gap to the band side gap and the balancing pipes from the crown side gap to the draft tube, the calculation is to be preceded by determining the leakage of the balancing device at the inner radii, and is then to be performed step-by-step to determine the one at the outer radii.

4. Description of Francis Turbines and Pump-Turbines Calculated

The calculation were performed in sixteen kinds of model Francis turbines and twelve kinds of model Francis pump-turbines of different manufacturing companies and of different specific speed(n_{SQ}) covering almost all of the n_{SQ} range of a Francis turbine and a pump-turbine. The main dimensions, the measured performance data and the test Reynolds numbers of these model turbines and pump-turbines are reported in the JSME-S008[1]. The configurations



(a) Francis turbine T-84 (b) Francis turbine T-59

(c) Francis turbine T-40 (d) Francis turbine T-27

(e) Francis pump-turbine PT-76 (f) Francis pump-turbine PT-30

Fig. 1 Runner configurations of model Francis turbines and pump-turbines calculated.

of some of the runners are illustrated in Figs. 1(a)~(f), in which the symbols "T" and "PT" are the abbreviations of turbine and pump-turbine respectively, and the number following the hyphen shows the value of the specific speed expressed in [m, m^3/s,rpm].

Generally, axial thrust and leakage are serious problems in low specific speed runners and double annular seals, the inner and the outer seals,are usually adopted together with several thrust-balancing devices, such as balancing pipes connecting the crown side to the band side (expressed as "C-B balancing pipe"), balancing pipes connecting the crown side to the draft tube ("D.T. balancing pipe") and balancing holes. However, in high specific speed runners leakage and axial thrust are relatively small and thrust balancing device and leakage sealing system are relatively simple as shown in Figs. 1(a), (b) and (c).

In high-head pump-turbines thrust balancing is especially important because of high pressure, and many kinds of balancing device are adopted as shown in Fig.1(f). At the start of pumping mode, air is supplied into a runner to minimize start-up torque, and it is then necessary to exhaust air rapidly at the depression period (blow down). Balancing holes of small diameter are then equipped not only for thrust balance but also for depression.

Generally, a model configuration is made strictly similar to that of a prototype except for an annular seal gap and a surface roughness because of machining tolerance. In the present model turbines and model pump-turbines calculated, some of them do not have balancing devices though the corresponding prototypes have. Some of them have much larger seal gap ratio, because they are made only for cavitation testing.

5. Calculated Results and Discussions

5.1 Volumetric and mechanical efficiencies as a function of specific speed

The theory was applied to the above-described turbines and pump-turbines. The pressure drop from the inlet to the outlet of a runner obtained by a quasi-three-dimensional analysis are plotted against n_{SQ} in Fig. 2, in which Δp_C and Δp_B are the pressure drops at the crown and the band sides, respectively. The non-dimensional pressure drop decreases rapidly with n_{SQ} in the range $n_{SQ} < 50$ and little in $n_{SQ} > 50$, and is larger in pump-turbines than in turbines. The empirical formulae shown in Fig. 2 are obtained for ΔP_C and ΔP_B.

Using the above-obtained boundary pressure, the leakage in the band and the crown side gaps are predicted for 16 Francis turbines and 12 Francis pump-turbines of different n_{SQ} and different manufacturing companies.

Plotting the non-dimensional leakage against n_{SQ}, the scatter of the results is considerably large as shown in Fig. 2. This is because the seal gaps, the thrust balancing devices adopted and the test Reynolds num-

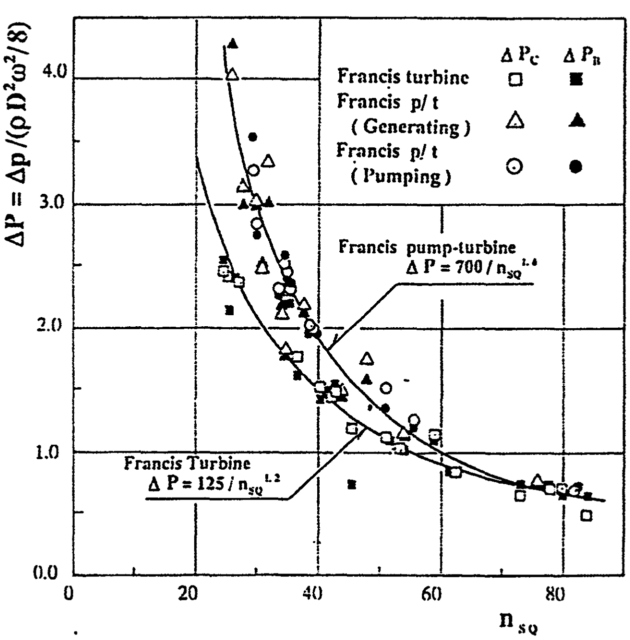


Fig. 2 Pressure drop of runner vs. n_{SQ}

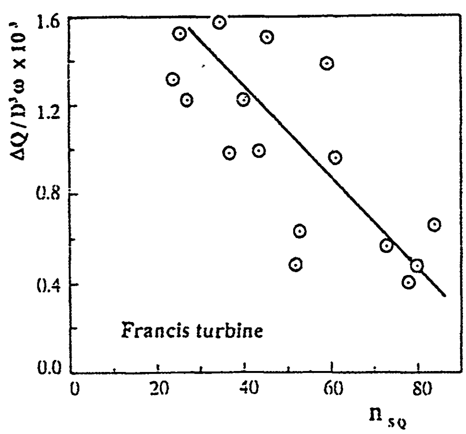


Fig. 3 Non-dimensional leakage vs. n_{SQ}

bers are much different depending on each manufacturing company. It seems difficult to determine the dependence of leakage and disk friction upon n_{SQ}.

The calculated volumetric efficiency η_v and mechanical efficiency η_m are plotted in Figs. 4 and 5 as a function of the specific speed n_{SQ}. The scatter of the data becomes much smaller and might be acceptable if compared with Fig. 3. The three data enclosed with a circle in Fig.4 are neglected, because their seal gaps are relatively large as mentioned in Section 4.

It is recognized from Fig. 4 that the volumetric efficiency η_v of Francis turbine amounts to about 99.5 % in high n_{SQ} runners, and around 99% in low n_{SQ} runners. It is also recognized that η_v of Francis pump-turbine takes almost the same value as that of Francis turbine, though the pressure drop of a runner is much larger in pump-turbine as shown in Fig. 2. This suggests that the annular seal gaps in pump-turbine are controlled more carefully than those in turbines.

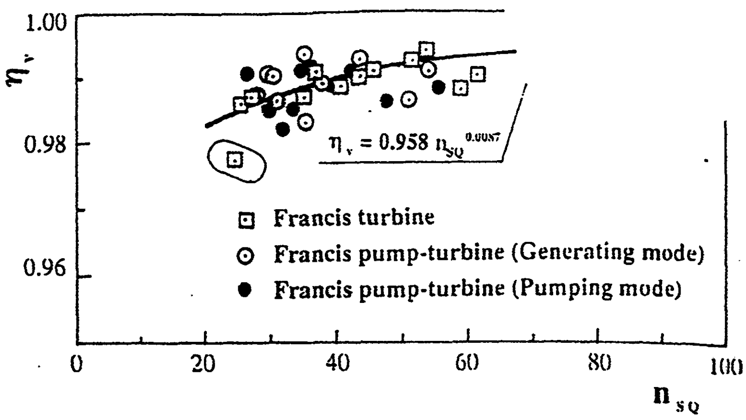


Fig. 4 Volumetric efficiency η_v vs. n_{SQ}

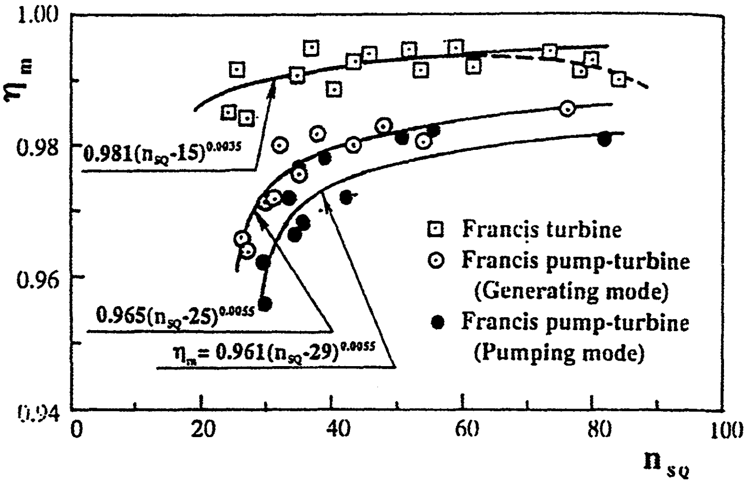


Fig. 5 Mechanical efficiency η_m vs. n_{SQ}

Figure 5 reveals the clear tendency that the mechanical efficiency η_m of a Francis turbine is the highest and takes the value of more than 99% in the wide n_{SQ} range, and that it decreases both in a very high and a very low n_{SQ} range. It is also revealed that η_m of a pump-turbine is about 1% lower than that of a turbine and that η_m value in pumping mode is lower than that in generating mode. In both operating modes of a pump-turbine, η_m-value decreases rapidly with a decrease in n_{SQ} in the range of $n_{SQ}<40$. The mechanical efficiency in the zero leakage case becomes about 0.25% lower than the normal gap case[7].

Generally, a runner outer diameter of a pump-turbine is much larger than that of a turbine under the same specification, which results in lower mechanical efficiency in a pump-turbine than in a turbine, as described above.

As shown in Figs. 4 and 5, the fitting curves are obtained as a function of n_{SQ} as;

$$
\begin{aligned}
\eta_v &= 0.958\, n_{SQ}^{0.0087} && \text{for Francis turbine and pump-turbine} \\
\eta_m &= 0.981\,(n_{SQ}-15)^{0.0035} && \text{for Francis turbine} \\
&\quad 0.965\,(n_{SQ}-25)^{0.0055} && \text{for Francis pump-turbine(G-mode)} \\
&\quad 0.961\,(n_{SQ}-29)^{0.0055} && \text{for Francis pump-turbine(P-mode)}
\end{aligned} \qquad (1)
$$

5.2 Leakage Behavior and Disk Friction and Axial Thrust Performances

The difference in types of leakage sealing and thrust-balancing devices causes large difference in leakage flow-rate, disk friction torque and axial thrust. The influences of main dimensions of these devices upon leakage, disk friction and axial thrust are examined in this section. To begin with, the non-dimensional parameters representing the leakage flow-rate(ΔQ), the disk friction torque(M) and the axial force(T) are defined as follows;

$$C_q = (\Delta Q/2\pi r_{1C}^3\omega)(\omega r_{1C}^2/\nu)^{0.2},\quad C_M = M/(\rho r_{1C}^5\omega^3/2),\quad \Delta C_T = (T_C - T_B)/(\rho\pi r_{1C}^4\omega^2) \qquad (2)$$

Here C_q is the leakage parameter deduced from the equation of motion of the gap flow.

(a) Influence of seal gap δ ; The seal gap δ has a dominant influence on the leakage. Taking as the simplest case a high n_{SQ} runner T-73 which has a single annular seal at the runner periphery and D.T.balancing pipes, the behavior of leakage parameter C_q , disk friction coefficient C_M and axial thrust coefficient ΔC_T is illustrated in Fig. 6, when the seal gap δ_o is varied. It is seen that the leakage increases rapidly and almost linearly with an increase in δ_o, and that the leakage is much larger in the band side (C_{qB}) than in the crown side(C_{qC}).

However, the disk friction decreases with an increase in δ_o and the band side disk friction C_{MB} amounts to more than five times the crown side one C_{Mc}. In the actual gap range shown in the figure, the disk friction torque changes a little in spite of a large change in the leakage. It is also seen that axial thrust increases rapidly with an increase of δ_o in the large gap range.

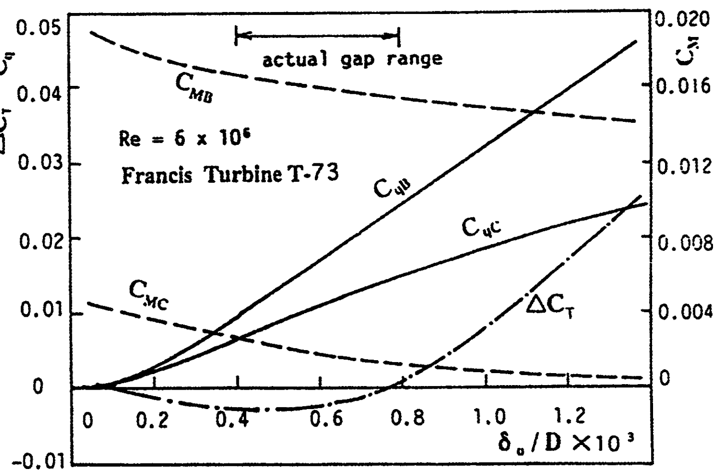


Fig. 6 Influence of seal gap (single seal case)

In the case of double annular seals at the outer and the inner radii, the leakage characteristics become more complicated, and the variation of C_q in the T-40 runner is shown in Fig.7 for the variation of the outer seal gap δ_0 by a solid line and the inner seal gap δ_i by a chain-dotted line. It is recognized that the leakage increases rapidly with an increase in δ_0 but soon become saturated, while it increases almost linearly with an increase in δ_i without saturation. These two lines intersect

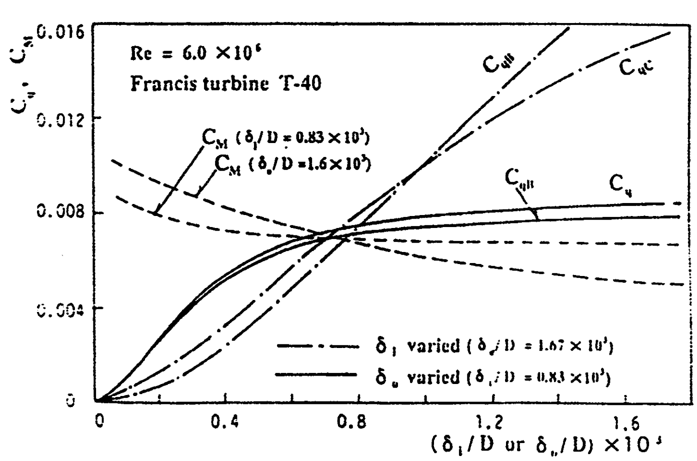


Fig. 7 Influence of seal gap (double seal case)

around the actual gap range.

It is then concluded that the reduction of an inner annular seal gap is much more effective to reduce the leakage than that of an outer seal gap, and that the outer annular gap need not be necessarily small. On the other hand, there also exists the possibility that the leakage increases remarkably due to wear or aging of an inner seal metal.

(b) Influence of balancing hole and D.T. balancing pipes ;

Taking as an example two Francis turbines, T-59 shown in Fig. 1(b) and T-27 in Fig. 1(d), the influence of balancing hole and D.T. balancing pipes are revealed. The former has four balancing holes and the latter two D.T. balancing pipes as an axial thrust balancing device. The behavior of leakage, disk friction and axial force in the crown side gap is plotted in Figs. 8 and 9 against the variation of the total sectional area ratio nd^2/D^2 of balancing holes and D.T.balancing pipes, respectively.

In both cases, it is clearly recognized that the leakage increases rapidly with an increase in nd^2/D^2 but soon becomes saturated in the range of $nd^2/D^2 > 0.003$ in the balancing hole case and $nd^2/D^2 > 0.005$ in the D.T. balancing pipe case.

Further increase in nd^2/D^2 of balancing holes or D.T. balancing pipes causes no change in C_q, C_M and C_T of the crown side. Figures 8 and 9 also reveals that the friction torque and axial force in the crown side changes a little in spite of large change in the leakage.

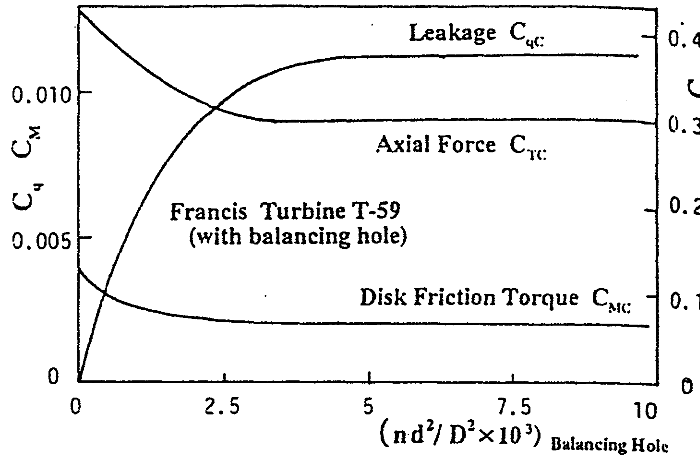


Fig. 8 Influence of balancing hole on Cq, C_M and C_T of crown side gap

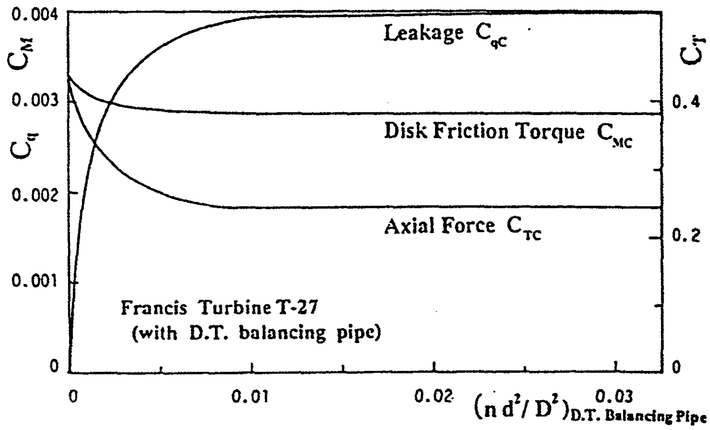


Fig. 9 Influence of D.T. balancing pipe on Cq, C_M and C_T of crown side gap

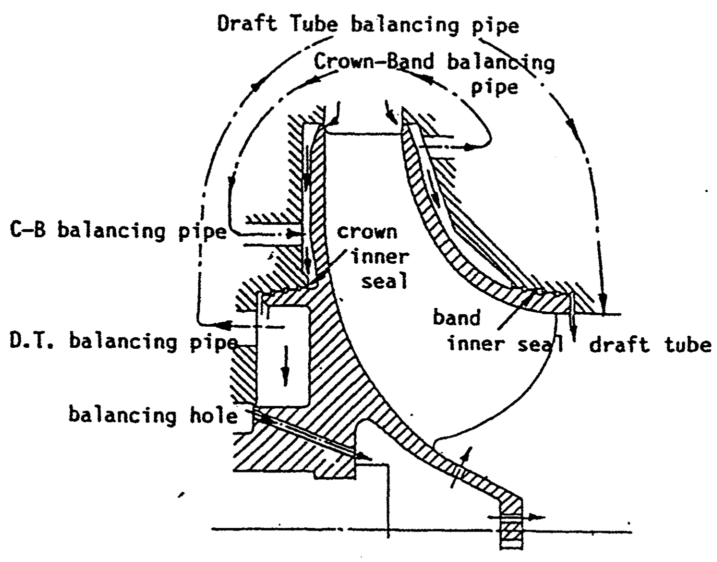


Fig. 10 Several leakage passes in PT-30 runner

(c) Combined Effects of B-C Balancing Pipe, D.T. Balancing Pipe and Balancing Hole ;

In low specific speed pump-turbines many kinds of thrust balancing devices are adopted such as C-B balancing pipes, D.T balancing pipes and balancing holes. Here the influences of these devices on the leakage behavior is examined taking as an example the PT-30 runner shown in Fig.1(f), of which leakage pass is illustrated in Fig. 10.

The C-B balancing pipes combine the flow field at the crown side with that at the band side, and the problem here is how large interaction between two flow fields is caused.

Figure 11 shows the behavior of every leakage, when all of the inner annular seal gaps δ_{ic} in the crown side are changed. The notations "crown inlet" and "band inlet" indicate the leakages at the runner outer periphery entering into the crown and the band side gaps, respectively.

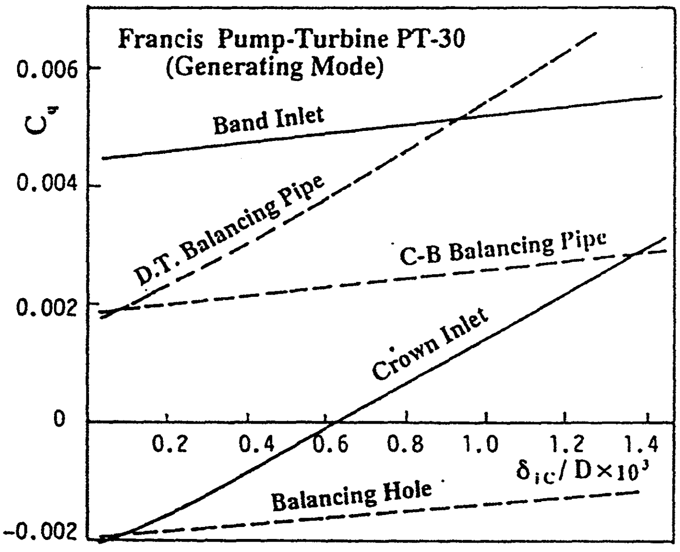


Fig. 11 Leakage behavior for the variation of inner seal gap δ_{ic} in PT-30 runner.

It is seen from Fig.11 that the increase in annular gap δ_{ic} causes large increase in the crown inlet leakage, most of which flows through the D.T. balancing pipes to the draft tube, and that the leakages both in the C-B balancing pipes and at the band inlet changes a little.

Figure 11 also reveals the following remarkable and interesting behaviors of leakage;

(1) The balancing hole leakage is negative, which means that the leakage comes from the runner channel into the crown gap, which is against expectation and reverse to the that shown in Fig.10. This is because the pressure in the gap inner region is very low, and because the balancing hole works as a pump and gives pumping head to the balancing hole leakage. This fact suggests that the balancing holes combined with D.T.balancing pipes is more effective in thrust balancing than without the ones.

(2) In the range of small annular seal gap δ_{ic}, the crown inlet leakage takes the negative value, which means that the leakage leaves from the crown side gap to the main flow through the runner outer annular seal, which is also against expectation. Figure 11 reveals that this leakage comes from the runner band inlet through C-B balancing pipes, and that this leakage flows outward in the crown side gap. This leakage makes the gap pressure higher near the runner periphery than the runner inlet pressure.

(3) Even if the crown inner seal gap is varied largely, the flow characteristics both in the band side gap and in the inner radii of the crown side gap change a little.

It is then concluded that the change in inner seal gap hardly causes interaction between the crown and the band side gaps, and that the balancing hole leakage flows in the reverse direction to an usual case and gives no contribution to the volumetric efficiency, as it takes the low head fluid from the runner outlet region near the runner cone.

In order to reveal the combined influences caused by balancing pipes, the variation

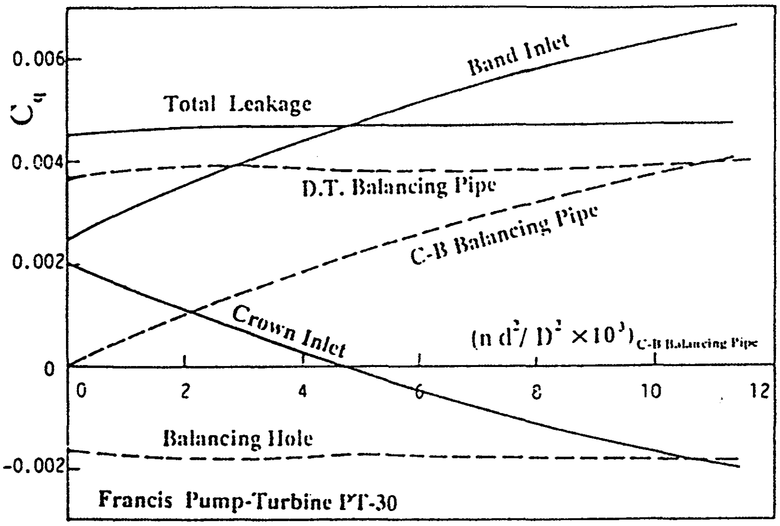


Fig. 12 Leakage behavior for varying total sectional area of C-B balancing pipe

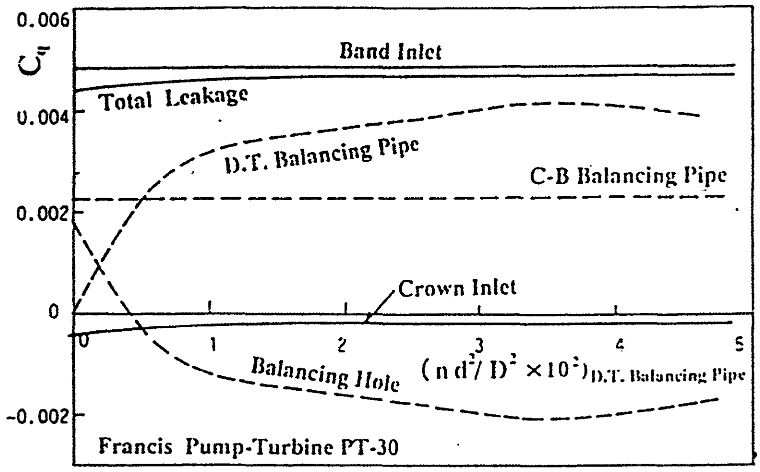


Fig.13 Leakage behavior for varying total sectional area of D.T. balancing pipe

of leakages is illustrated in Figs. 12 and 13 against the variation of the total sectional area ratio nd^2/D^2 of C-B balancing pipes and D.T. balancing pipes, respectively. From Fig.12 the interaction between the crown gap flow and the band gap flow can be examined.

As shown in Fig.12, the increase in nd^2/D^2 of the C-B balancing pipes causes large increase in the band inlet leakage, which flows into the crown gap through C-B balancing pipes and further flows into the D.T. balancing pipes together with the leakage coming from the balancing holes. It is interesting that the leakages in the D.T. balancing pipes and the balancing holes change little. This is because the decrease in the crown inlet leakage is supplied by the increase in the band inlet leakage which comes through the C-B balancing pipes, and thus the total leakage is seen to be kept constant. As a whole, the change in nd^2/D^2 of the C-B balancing pipes makes no change in the total leakage.

Compared with the above-described leakage behavior due to the C-B balancing pipes, the influence of the D.T. balancing pipes shows much different behavior, as shown in Fig.13. The total leakage is also seen to be kept constant as the case of the C-B balancing pipes even if the total sectional area is largely varied. With an increase in nd^2/D^2, the leakage through the D.T. balancing pipes increases rapidly in the range of $nd^2/D^2<0.01$.

It seems strange that the other leakages, the band inlet leakage and the crown inlet leakage, vary little over the whole range of nd^2/D^2. The balancing hole leakage becomes positive in the small nd^2/D^2 range, and flows from the crown side gap to the runner channel. It is then concluded that an increase in nd^2/D^2 of the D.T. balancing pipes causes a rapid decrease in the balancing hole leakage in the small nd^2/D^2 range, and that a further increase of D.T.balancing pipe leakage changes the balancing hole leakage to the reverse direction. The D.T. balancing pipe leakage is thus supplied by the balancing hole leakage.

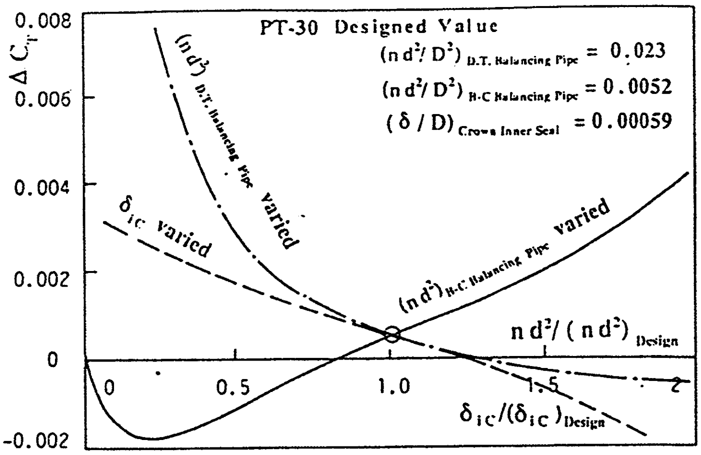


Fig. 14 Axial thrust behavior in PT-30 turbine

Lastly, the axial thrust behavior is illustrated in Fig.14 against the variations of the crown inner seal gap δ_{iC} and the total sectional area $\pi nd^2/4$ of the balancing pipes. The abscissa is normalized by the corresponding designed values shown in the figure.

Figure 14 reveals the following interesting behaviors of the downward axial thrust;

(1) In the range of large total sectional area nd^2 of the D.T. balancing pipes, the upward (negative) axial thrust works. With a decrease in nd^2 of the D.T. balancing pipes, axial thrust changes its direction, and in the small nd^2 range the downward axial thrust increases rapidly. This reveals that the throttling of D.T. balancing pipes causes rapid increase in downward axial thrust.

(2) With a decrease in the total sectional area nd^2 of the B-C balancing pipes, the downward axial thrust decreases almost linearly except for a very small nd^2 range. The throttling of the C-B balancing pipes thus causes an increase of upward axial thrust.

(3) With a decrease in the crown inner seal gap δ_{iC}, the downward axial thrust increases almost linearly. This suggests that a wear or aging of a seal metal causes an increase in downward axial thrust.

The above-described results also suggests that it is possible to control axial thrust of Francis pump-turbine by throttling both the D.T. balancing pipes and C-B balancing pipes.

6. Conclusions

A gap flow analysis is applied to sixteen types of model Francis turbines and twelve types of model Francis pump-turbines of different n_{SQ}, and the volumetric and mechanical efficiencies are determined theoretically. They are expressed by a simple formulae as a function of the specific speed, and the behavior of leakage, disk friction and axial thrust are revealed theoretically. The results are summarized as follows:

(1) Volumetric efficienciy η_v of model Francis turbine and pump-turbine is around 99% and increases with n_{SQ}. Mechanical efficiency η_m is around 99% in model Francis turbine and 1% lower in pump-turbine. In Francis pump-turbine, η_m is higher in generating mode than in pumping mode, and decreases with a decrease in n_{SQ} and rapidly in the range of $n_{SQ}<40$. In

high n_{SQ} runner the band side disk friction amounts to more than five times the crown side.
(2) In the single annular seal case, leakage is much larger in the band side than in the crown side and increases almost linearly with an increase in the seal gap. In the double annular seal case, leakage is nearly same in both gaps and increases largely and almost linearly with an increase in the inner seal gap, but is soon become saturated with an increase in outer seal gap. The decrease of an inner seal gap is much more effective for leakage reduction than that of an outer seal gap, and the outer annular gap need not be necessarily small.
(3)Leakage increases rapidly with an increase in total sectional area of D.T.balancing pipes or balancing holes in the range of $nd^2/D^2<0.04$, and little in the range of $nd^2/D^2>0.05$.
(4) When balancing holes are used together with D.T. balancing pipes of not too small diameter, the balancing pipe leakage flows from the runner channel into the crown gap against our expectation, which does not contribute to the volumetric efficiency..
(5) When D.T.balancing pipes and C-B balancing pipes are used together with balancing holes, the total leakage hardly changes even when the total sectional area of these pipes or holes are changed largely. With an increase in the sectional area of D.T. balancing pipes, only the balancing hole leakage increases and brings low energy liquid from the runner cone region into the D.T. balancing pipes. With an increase in the sectional area of C-B balancing pipes, the band inlet leakage increases in the same rate as the decrease in the crown inlet leakage, and flows through C-B balancing pipes. With an increase in crown inner seal gap, only the crown inlet leakage increases linearly and flows through the D.T. balancing holes. When the total sectional area of D.T. balancing pipes or C-B balancing pipes is small, or when the crown inner seal gap is small, the crown inlet leakage becomes negative.
(6) It is possible to control axial thrust by throttling D.T. balancing pipes and C-B balancing pipes, as the former causes a rapid increase in downward thrust and the latter upward one.

Acknowledgments

The present study was performed as a part of the activity of the W.G. " Performance Conversion of Hydroturbine" under Turbomachinery Society of Japan. The present authors acknowledges the considerable assistance of the W.G. members in presenting valuable data and discussions.

References

(1) JSME Standard-008, "Performance Conversion Method for Hydraulic Turbines and Pumps", (1989).
(2) Kurokawa, J. and Toyokura, T., "Axial Thrust, Disk Friction Torque and Leakage Loss of Radial Flow Turbomachinery", Proceedings of Pumps and Turbines Conference (Glasgow), Vol.1(1976).
(3) Kurokawa, J., Toyokura, T., Shinjo, M and Matsuo, K., "Roughness Effects on the Flow along an Enclosed Rotating Disk", Bullettin of JSME, Vol. 21, No. 162(1978), p. 1725.
(4) Kurokawa, J. and Sakuma, M., " Flow in a Narrow Gap along an Enclosed Rotating Disk with Through-Flow", JSME International Journal, Vol. 31, No. 2(1988), p. 243.
(5) Kurokawa, J., "Simple Formulae for Volumetric and Mechanical Efficiencies of Hydraulic Machinery", Proceedings of 3rd Japan-China Joint Conference on Fluid Machinery, Vol. 2(1990), p. 101.
(6) Kurokawa, J., Kamijo, K. and Shimura, T., "Axial Thrust Behavior of Rocket Engine", AIAA Jr., Journal of Propulsion and Power, Vol.10, No. 2(1994), p. 244.