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The Effects of Runner Surface Roughness on the Performance of a Francis Turbine*

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The effects of runner surface roughness on the performance of a Francis turbine have been studied experimentally.

The performance data were obtained for seven various type roughnesses from 0.007 to 1.0 mm in protuberant height.

Preliminarily, the frictional resistance of enclosed rotating disks with respective roughnesses was measured for laminar and turbulent flows, in order to relate these different roughnesses to hydraulic frictional resistance.

Through these disk-friction tests, the roughness of these surfaces could be expressed by the equivalent Nikuradse sand-grain diameter.

The following conclusions may be drawn from this experimental study.

(1) The turbine efficiency decreases in proportion to the frictional resistance of its equivalent sand roughness.

(2) With an increasing roughness, the maximum efficiency point shifts to the lower speed and the lower discharge.

(3) Concerning the admissible roughness, the Schlichting's formula is in good agreement with this experiment.

Nomenclature

(1) Disk-friction test

- a : radius, mm
 b : thickness of disk, mm
 t : disk radial-tip clearance, mm
 s : axial clearance between disk and end wall, mm
 ω : angular velocity, rad/sec
 ρ : mass density, kg sec³/m²
 C_{mt} : tip-friction-torque coefficient
 C_m : torque coefficient due to axial gap
 M : frictional torque (moment) = $(C_m + C_{mt}) \rho \omega^2 a^5$, kg m on two faces of disk
 ν : kinematic viscosity, m²/sec
 R_e : disk Reynolds number = $\omega a^2 / \nu$
 k : effective height of roughness protuberance, mm
 k_s : mean grit particle diam. or equivalent sand roughness, mm
 k_{adm} : admissible roughness, mm

(2) Performance test of model Francis turbine

- η : turbine efficiency, %

- H : effective head, m
 N : rotating speed, rpm
 Q : discharge, m³/t
 P : output, kW
 n : unit speed = $N/\sqrt{H}$, rpm/m^{1/2}
 q : unit discharge = $Q/\sqrt{H}$, (m³/sec)/m^{1/2}
 p : unit output = $P/H^{3/2}$, kW/m^{3/2}
 a_g : gate opening, mm
 n_s : specific speed, m kW
 l : length of runner passage, mm
 m : hydraulic mean depth of runner passage, m
 A : sectional area of runner passage, m²
 g : acceleration due to gravity, m/sec²

1. Introduction

Concerning the study of roughness effects on the fluid turbo-machines, the studies on a pump have been presented in Reference (1), but there have been no studies on a turbine.

To investigate this question, the performance tests (the efficiency tests) of a model Francis turbine were carried out for seven various type roughnesses of the runner surface.

These surfaces included a smooth surface and six rough surfaces, which were two surfaces roughened by the grinder, three sand surfaces and a

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wavy surface with the rubber plates.

As the performance change of a turbine due to runner surface roughness depends mainly upon the difference of its frictional resistance, it is necessary to measure the hydraulic frictional resistance of each surface quantitatively in advance.

For this purpose, the frictional resistance of enclosed rotating disks with the same roughness as the turbine runner's was measured for laminar and turbulent flows.

Through these disk-friction tests, the grinder finished surface roughnesses could be expressed by the equivalent Nikuradse sand-grain diameter (the equivalent sand roughness).

First, the disk-friction tests are discussed, next, the performance change of Francis turbine due to the runner surface roughness is described, and the relation between the two is connected with the equivalent sand roughness.

Concerning the admissible roughness, it was confirmed that the Schlichting's formula⁽⁴⁾ was in good agreement with these experiments.

2. Roughness effects on frictional resistance of enclosed rotating disks

2.1 Test equipments and procedures The general layout of the test stand is indicated in Fig. 1. The disk was driven by a 20 kW DC motor with belts. Shaft speeds were measured by a digital counter and the speed range was set at 250~2 500 rpm. Torques were measured by a torque meter containing a strain-gage bridge. A static calibration was performed over the full torque range.

As the bearing friction was included in the gross torque measurements, it was necessary to deduct the friction. Therefore, the bearing friction was measured without a disk in the air for all test runs.

Table 1 Surface characteristics

No.	Roughness patterns	Roughness k^* or k_s^{**} mm	Remark	Symbols
1	Runner	File finish	$k=0.005\sim0.008$	Hydraulically smooth
	Disk	Machine finish	$k=0.003\sim0.005$	Hydraulically smooth
2	Grinder finish	$k=0.035\sim0.045$	$k_s=0.008$ 2 mm	×
3	Grinder finish	$k=0.090\sim0.110$	$k_s=0.018$ 1 mm	△
4	Sand cementing	$k_s=0.245$	Sieve mesh size=48~65	+
5	Sand cementing	$k_s=0.505$	Sieve mesh size=28~35	⊙
6	Sand cementing	$k_s=1.020$	Sieve mesh size=14~20	□
7	Wavy surface	$k=1.0$	Reference in Fig.3	●

* k : effective height of protrusions.

** k_s : mean grit diam. or equivalent sand roughness.

Fig. 1 Schematic assembly drawing of disk-friction test stand

Fig. 2 Test disks

directly proportional to the distance from the center, here the wavy shape at the periphery of the disk was adopted nearly the same as the wavy shape on the runner vane.

The protuberances of the surface roughened by the grinder were measured by means of a roughness indicator of a needle contact type. On the other hand, the sand surfaces being too rough to be measured by the indicator, the negative patterns were made of the softened modeling compound by pressing them on the sand surfaces. By cutting those patterns, the sand roughness protuberances were measured by means of observation of those cutting sections through a magnifying glass.

2.2 Test results and considerations of disk-friction tests According to the combination of the roughnesses on the rotor and the stator, many cases can be considered. In this study, as described in Paper⁽¹⁰⁾ by NECE and Daily, experimental torque data were obtained for the case of equal roughnesses on both the rotating disk (rotor) and the stationary, circular end wall of the casing (stator), for the seven roughnesses at a constant axial-clearance and a constant radial-tip clearance as shown in Fig. 1.

The cylindrical surface of the chamber was smooth in all runs.

The data on the frictional resistance are presented in the form of diagram of moment coefficient ($C_m + C_{mt}$) versus disk Reynolds number Re .

Fig. 5 is a summary plot of the experimental data. Many studies and experiments for the smooth plane disk enclosed within a right-cylindrical chamber have been reported in Reference (5) to (8).

Recently, Daily and Nece⁽⁹⁾ verified the exis-

tence of four basic flow regimes in the axial gap between the disk and the casing wall as shown in Fig. 4, and frictional torque coefficients for these regimes are given as follows;

Regime I

$$C_m = \pi \left/ \left(\frac{s}{a} \right) R_e \right. \dots\dots\dots (1)$$

Laminar flow, merged boundary layers on disk and end wall. Namely, $s < \delta + \xi$.

Regime II

$$C_m = 1.85 \left(\frac{s}{a} \right)^{1/10} / R_e^{1/2} \dots\dots\dots (2)$$

Laminar flow, separate boundary layers on disk and wall. Namely, $s > \delta + \xi$.

Regime III

$$C_m = 0.04 \left/ \left(\frac{s}{a} \right)^{1/8} R_e^{1/4} \right. \dots\dots\dots (3)$$

Turbulent flow, merged boundary layers. Namely, $s < \delta + \xi$.

Regime IV

$$C_m = 0.051 \left(\frac{s}{a} \right)^{1/10} / R_e^{1/5} \dots\dots\dots (4)$$

Turbulent flow, separate boundary layers. Namely, $s > \delta + \xi$.

The tip-friction-torque coefficient C_{mt} was given in Reference⁽⁹⁾, and computed assuming two-dimensional laminar or turbulent flow in the small radial gap and with the laminar wall shear for a linear velocity distribution and with the turbulent wall shear corresponding to the smooth wall resistance of a duct at the equivalent Reynolds number. Considering the case of this experiment, the magnitude of C_{mt} is only 3 per cent of C_m , then the effect of radial-tip-clearance is considered to be negligible. Therefore, the frictional torque M could

Table 2 Roughness dimensions

No	1	2	3	4	5	6	7*
k mm	0.004	0.040	0.100	0.245	0.505	1.02	1.00
L mm	0.13~0.15	0.49~0.65	0.78~0.90	0.47~0.60	1.10~1.30	1.80~2.3	49.0
L/k	33~38	12~16	8~9	2~2.5	2.2~2.6	1.8~2.3	49

* The No.7 data are calculated at the periphery of the disk.

Fig. 6 Roughness dimensions

Fig. 7 Dimensionless correlation of torque data, $1/\sqrt{C_m}$ versus a/k_s , fully rough zone

be given approximately as $M = (C_m + C_{mt})\rho\omega^2 a^5 \div C_m \rho\omega^2 a^5$.

In this experiment, since the axial gap ($s = 20$ mm, $s/a = 0.04$) was sufficiently large in comparison with the boundary layer thickness, theoretical torque equations to be used were Eqs. (2) and (4) for laminar and turbulent flows respectively.

Using a log-log plot of C_m versus R_e , above equations gave straight lines as shown in Fig. 5.

Agreement of the No. 1 data with theoretical equations was very good except the transitional zone. However with further increases in R_e , the data deviate from the theoretical lines and the deviation point at $R_e \div 3.7 \times 10^6$ could be considered one of the onset of rough surface behavior, at which the effects of roughness are first visible, for the No. 1 roughness ($s = 0.003 \sim 0.005$ mm).

Considering the case of a rough disk with rough end walls, the behaviours of relationship of C_m versus R_e were the same as the results of a rough pipe or a rough plate measurement as shown in Fig. 5, i.e. at the highest Reynolds numbers, regarding the No. 4 to No. 6 sand surfaces, the C_m values became independent of R_e , and were seen to depend upon only the relative roughness k_s/a . But for the wavy surface, even in the range of the highest Reynolds numbers, the C_m values depended upon both R_e and k/a . Concerning the grinder finished surfaces No. 2 and No. 3, the form of C_m versus R_e relationship was intermediate between that for the wavy surface and that for the sand surface.

As well known, the behavior of surface roughness depends upon the ratio of the pitch of roughness L and the protuberance height k , as indicated in Fig. 6.

A trace for each roughness was analyzed; the results are listed in Table 2.

In the case of the sand roughnesses, the ratio L/k was about two, and when the L/k increases to about fifteen or so, the roughness behaviour becomes that of the wavy surface.

Considering the case of a fully rough turbulent flow for the No. 4 to the No. 6 in which C_m became independent of R_e , a plot of $1/\sqrt{C_m}$ versus $\log(a/k_s)$ gave a straight line as shown in Fig. 7. Then, after Nece⁽¹⁰⁾, the expression for the full rough

zone became.

$$1/\sqrt{C_m} = 5.13 \log(a/k_s) - 1.85 \quad \dots\dots\dots (5)$$

2.3 Equivalent sand roughness of surface roughened by grinder The types of roughness except the sand surface be expressed in terms of the equivalent Nikuradse sand-grain diameter for the fully rough zone, i.e. the equivalent sand roughness k_s is defined as that value which gives the actual coefficient of resistance when inserted into Eq. (5).

For the No. 2 and the No. 3 surfaces roughened by the grinder, the results using the C_m values at $R_e = 5 \times 10^6$ are indicated in Table 3. From these results, the equivalent sand roughness of the surface roughened by the grinder seems to be one fifth of its protuberance height.

The fully rough regime being nonexistent for the No. 7 wavy surface, the correlation is unreasonable. If the k_s value is calculated for reference by substituting the C_m value at $R_e = 5 \times 10^6$ into Eq. (5), the equivalent sand roughness will be much smaller than the height of the wave as indicated in Table 3.

3. Roughness effects on performance of Francis turbine

3.1 Tests equipments and procedures The distributor section of the model Francis turbine is shown in Fig. 8. The surface roughnesses of the model Francis runner were the same as those of disks except the No. 1 runner. The form of the runner vanes was too complicated for machine work, therefore the No. 1 runner was filed smooth by hands, so its protrusions ($k = 0.005 \sim 0.008$ mm) are somewhat larger than those of the No. 1 disk ($k = 0.003 \sim 0.005$ mm).

Each roughened runner is shown in Fig. 9. The layout of the test equipments indicated in Fig. 10.

The tests were run with 8 mm to 40 mm gate opening and for each gate opening the measurements were made over the range of unit speed from about 60 rpm to 160 rpm.

Fig. 8 Distributor section of model Francis turbine

Table 3 Equivalent sand roughness

No	C_m ($R_e = 5 \times 10^6$)	k mm	k_s mm	k/k_s
2	0.002 6	0.035~0.045	0.008 2	4.3~5.5
3	0.003 15	0.090~0.110	0.081 1	4.9~6.1
7	0.003 3	1.0	0.022 0*	45.4

* The k_s for the No. 7 was calculated by substituting the C_m value at $R_e = 5 \times 10^6$ into Eq. (5).

Water was supplied to the model turbine through a Venturi meter from the upper reservoir and discharged into the lower reservoir through a draft tube. The test was run under almost constant water level of the upper and the lower reservoirs. The static heads on the model turbine were about 4 m during the test. The brake force was measured by a 38 kW mechanical dynamometer. The speed measuring device was a digital counter. The static head was indicated on a scale on the lower float

No. 3 (Grinder finished surface)

No. 5 (Sand surface)

No. 6 (Sand surface)

No. 7 (Wavy surface)

Fig. 9 Test runners

as shown in Fig. 10. The Venturi meter was used to measure the discharge. The coefficient of this Venturi meter was determined by checking against a 1.0 m rectangular weir with end contraction.

3.2 Performance of the smooth surface runner (No.1) The relative velocity in the runner passages is maximum at the runner exit. Whereas the test static head was about 4 m, that maximum velocity V_0 was about 5.8 meter per second. From the Schlichting's formula⁽⁴⁾, the admissible roughness was given as $k_{adm} = 100\nu/V_0 = 0.022$ mm. Therefore, the No.1 runner could be said to be hydraulically smooth for the 4 m test head.

The performance data at the best efficiency point of the No.1 runner are listed in Table 4. where, the test data of the model turbine were all converted to unit values, which represent the performance obtained when operating under one meter head.

Table 4 Characteristics of the smooth surface runner at the best efficiency point

Maximum efficiency	η_1 opt	%	88.1
Speed	n_1 opt	rpm	118.5
Discharge	q_1 opt	l/sec	108.0
Output	p_1 opt	kW	0.932
Gate opening	a_g	mm	29.8
Specific speed	n_s	m kW	114.5

Table 5 Dimensions of the runner passage and order of finish

No.	Runner exit opening B mm	Opening ratio (on the basis of No.1)	Mean area of runner passage A mm ²	Hydraulic mean depth of passage m mm	Order of finish
1	21.41	1.000	3 568	13.04	3
2	21.29	0.994	3 548	12.99	2
3	20.99	0.980	3 504	12.89	1
4	21.61	1.010	3 591	13.08	4
5	21.84	1.020	3 628	13.17	5
6	21.53	1.006	3 583	13.07	6
7	21.38	0.999	3 560	13.01	7

Fig. 10 Schematic assembly drawing of model turbine test stand

3.3 Correction of model turbine performance for the small difference in dimensions of runner passage caused by the finishing error The model runner was cast with gum metal and was similar to the prototype one, so it had three-dimensional vanes as shown in Fig. 9.

The tests proceeded one by one with the roughness of the runner passage on the same runner changed in the order of the No. 3, the No. 2..., as indicated in Table 5.

In the case of the sand glued runner, the dimensions of the passage were kept much the same as those of the No. 1 runner by scraping off

the vanes as much as the thickness of sand and cement.

But in the case of the grinder finished runner, with changing the roughness in the order of the No. 3, and the No. 2, and the No. 1, the thickness of the runner vane decreased little by little and the area of passage increased.

And also for the sand roughness, it was very difficult to make the passages exactly of the same dimension as the No. 1 runner's.

Preliminarily, to investigate the dimension effects of runner passage, the performance tests (the efficiency tests) of the model turbine were carried out for three kinds of passage dimension at similar passage and same surface roughness.

As an example, Fig. 11 represents the dimension effects of runner passage. Solid and dashed lines corresponded to the runner exit opening 21 mm and 23.7 mm respectively. The top curves were the envelope of the efficiency curves for each guide vane opening, and the bottom curves expressed the q - n diagram and iso-efficiency curves.

From these tests, the influence of the small difference of the passage dimension can be summarized as follows: with an increasing dimension, the maximum efficiency point shifted to the higher speed and the larger discharge, and the maximum efficiency increased. In these tests, the magnitude of the efficiency rise was 1.8 per cent.

In the running zone of the larger speed and the smaller discharge, the efficiency decreased with an increasing size of passage, but in the other running zone the efficiency increased.

So, only the roughness effects could be drawn from the tests data by means of the correction for the difference of the size of the runner passage.

3.4 Roughness effects on turbine performance

Whereas the frictional loss caused by roughness is proportional to the square of the velocity, a practical difficulty will occur at the normal discharge and the overdischarge running.

As an example, the speed characteristics for the roughness effects at around the normal gate opening and the over gate opening are shown in Fig. 12 and Fig. 13 respectively.

In these diagrams, the correction for the dimension effects was not adopted, therefore, the magnitudes of the efficiency and the discharge in the No. 3 runner indicated somewhat small values and those in the No. 5 runner somewhat large values with reference to Table 5.

It is natural that the efficiency decreases with an increasing roughness, and besides, the maximum efficiency point shifts to the lower

Fig. 11 Passage dimension effects on turbine performance

Fig. 12 Roughness effects on speed characteristics (around normal gate opening)

speed and the lower discharge. And also, at the constant gate opening and the constant roughness, the magnitude of a drop in the efficiency increases with an increasing speed. Therefore the run-away speed decreases with an increasing roughness.

The preceding behaviours can be explained qualitatively as follows.

If the geometrical dimensions of a turbine are given, the efficiency η may be written for the given R_s as

$$\eta = f(a_g, n, q, k) \quad (6)$$

Without considering the losses of leakage, seal friction and disk friction, the relationship among η , n , and q for the given gate opening are written by the following equations from the momentum theory⁽²⁾,

$$\eta = -a_1 n^2 + 2b_1 n q \quad (7)$$

$$a n^2 - 2b n q + c q^2 - 1 = 0 \quad (8)$$

Where a_1 , b_1 , a , and b are factors with relation to one or several from among the geometrical dimensions, the gate opening the shock-loss at entrance to the runner, and the draft tube loss due to a whirl. But c is related with the frictional loss of the passage and the preceding elements.

When the roughness on the runner passage is changed on condition of a constant gate opening, only the factor c is varied and other factors are constant.

If Δc and Δq denote the change of roughness and discharge respectively, from Eq. (8)

$$\begin{aligned} \Delta q/q &= -q \Delta c / 2c \left(q - \frac{b}{c} n \right) \\ &= -q \Delta c / 2c q_L \quad (9) \end{aligned}$$

Fig. 13 Roughness effects on speed characteristics (over gate opening)

When a specific speed n_s is situated about 114.5 m kW, such as in the test turbine, the curve of q against n in Eq. (8) forms an ellipse. The q_L in Eq. (9) is the distance between the q - n curve and the straight line as shown Fig. 14.

As the q_L can be considered to be approximately equal to q for the range of the turbine running, the relative change of the discharge becomes $\Delta q/q = -\Delta c/2c$.

The preceding equation denotes that the relative change of the discharge is independent of the speed.

Since the increase of the roughness is equivalent to the decrease of the discharge, from Eq. (7) the efficiency change $\Delta \eta$ is given by

$$\Delta \eta = 2b n \Delta q \quad (10)$$

As seen in the above equation, the efficiency change is proportional to the product of the speed and the discharge change. Therefore, with an increasing roughness, the maximum efficiency point shifts to the lower speed and the lower discharge.

In order to observe the roughness effects for the whole running zone, the efficiency curves of $\eta = 70\%$ and $\eta = 82\%$ on the q - n diagram are shown in Fig. 15.

According to this diagram, for the zone of high speed and small discharge, the order of decrement in efficiency was not evident, then it may be considered that the roughness has some stabilizing effects for the complicated flows in those zone.

Namely, roughness does not reduce the efficiency in a certain running zone.

Considering that the rotating speed of the actual turbine is constant, the discharge characteristics curves at the optimum speed of the No. 1 runner ($n_{1\text{opt}} = 118.5 \text{ rpm}$) is shown in Fig. 16.

Consequently, especially in the No. 7 runner with the wavy surface, the performance was much the same as that of the grinder roughened runner, although the height of the protuberance was nearly equal to that of the No. 6 sand roughness and ten per cent of runner vane thickness.

This shows that the results of the No. 7 run-

Fig. 14 Illustration of turbine performance

ner corresponded to the results of the No.7 disk friction test in turbulent flow.

Namely, concerning the manufacturing error of runner vane profile of a Francis turbine, there is no need of excessive accuracy in view of hydraulic efficiency.

4. Admissible roughness

4.1 Admissible roughness in Francis turbine runner In order to compare only the roughness effects on the performance, it is necessary that all hydraulic factors except the roughness be kept identical, i.e. the data should be analyzed under the same conditions for the inlet and outlet velocity diagrams of a runner. But that will be impossible, because if the outlet velocity diagram is given, (i.e. n and q are given), the inlet velocity diagram is varied with the roughness as shown in Fig. 16, and if the inlet velocity diagram is given, (i.e. n , q , and a_0 are given), the same thing happens.

In these tests, when the outlet velocity diagram was given, the maximum difference of the gate opening between the No.1 and the No.6 runners at the largest discharge was three degrees, namely, the discharge of the No.6 runner could not be equal to that of the No.1 runner, unless the gate opening of the No.6 was larger by three degrees than the No.1.

The changes of the inlet shock-loss and hydraulic mean depth were considered to be negligibly small as compared with the roughness effects for those small changes of the gate opening. Therefore, only the roughness effects could be drawn by comparing the data under certain values of q and n .

Fig. 15 Roughness effects in iso-efficiency curves

In Fig. 17 are shown the relationships between the efficiency and the roughness which were obtained from Fig. 16 for $n=118.5$ rpm and the four discharges q of 90, 100, 110, and 117 l/sec.

The dashed lines in Fig. 17 represent the admissible roughness k_{adm} which were obtained by using the Schlichting's formula $k_{adm}=100\nu/V_0$ and the outlet velocity for the static test head of 4 m.

In Fig. 17, there are three groups of plots, namely, the grinder finished surface, the sand glued surface and the wavy surface, then with a decreasing magnitude of the roughness in each group, the efficiency values increase and coincide with that of the smooth surface at k_{adm} .

4.2 Admissible roughness in enclosed rotating disk

In the case of the enclosed rotating disk, Nece et al.⁽¹⁰⁾ had given experimentally the relation of the roughness to the initial onset of turbulence, at which the effects of roughness are first visible, after the same process based on Nikuradse

Fig. 16 Roughness effects on discharge characteristics

Fig. 17 Efficiency versus surface roughness, and admissible roughness

pipe measurements.

But by giving only the relative roughness, it is impossible to obtain directly, through the above process, the numerical value of the Reynolds number at which the roughness effects occur.

From the practical point of view, it is shown here that the simple formula of k_{adm} by Schlichting can be also applied to the enclosed rotating disks.

The relative velocity is maximum at the periphery of the disk, and in this place, the radial flow need not be considered, because the disk is enclosed by the right-cylindrical chamber. Therefore, the flow pattern becomes as shown in Fig. 18, and the maximum relative velocity leads to the following equations.

$$\left. \begin{array}{l} \text{On the rotor } V = a\omega(1-\beta), \\ \text{On the stator } V = a\beta\omega \end{array} \right\} \dots\dots\dots(11)$$

In the case of the same roughness on rotor and stator, it can be assumed that $\beta=0.5$, then the roughness effects occur at the same time both on rotor and stator.

If R_e is defined as Reynolds number, at which the roughness effects are first visible, the following expression is obtained from Eq. (11), $R_e = aV/\nu$; and $k=100V/\nu$

$$R_e = 200 \frac{k}{a} \dots\dots\dots(12)$$

Where k must be regarded as the effective height of the protuberance. So the sand roughness, $k=0.67 k_s$ was assumed in consideration of the thickness of the bonding cement.

For comparison, the R_e values for each roughness are indicated in Fig. 5 with the broken lines.

This means that R_e approximately gives the transitional point from the smooth regime to the rough one.

5. Equivalent sand roughness and turbine efficiency

If the runner passages become rougher than the admissible roughness, the turbine efficiency, of course, decreases. In order to obtain quantitatively the relationship between the frictional loss and the efficiency, the frictional loss h_f in the runner passage was calculated by using the equivalent sand

Fig. 18 Flow patterns at the periphery of the disk

roughness for each runner.

The passage was treated as a duct and the frictional coefficient λ at the equivalent Reynolds number was determined from a Moody diagram⁽³⁾, namely,

$$\left. \begin{array}{l} h_f/H = \lambda l q^2 / 8m g A^2, \\ R_e = 4m q \sqrt{H} / \nu A \end{array} \right\} \dots\dots\dots(13)$$

As indicated in Fig. 19, the data points of η versus the ratio of h_f/H fall in a straight line for the various roughness patterns. Consequently the turbine efficiency decreases in proportion to the frictional loss of its equivalent sand roughness, and the amount of efficiency decrease is between 4 times (in small discharge) and 5.6 times (in large discharge) the frictional loss as a duct.

For the No. 7 wavy surface runner, the frictional loss of the passage was not calculated, because the correspondence between the disk-friction and the pipe-friction was not distinct quantitatively. But on account of the same trend in the disk-friction tests and turbine performance tests, the efficiency decrease of the wavy surface runner also may be considered to be proportional to the frictional resistance of its equivalent sand roughness.

6. Conclusions

The following conclusions may be drawn from this experimental study of the roughness effects on the turbine performance and the frictional resistance of the enclosed rotating disks.

(1) The surface of an actual Francis turbine runner is usually finished by the grinder, and the equivalent sand roughness of the finished surface is one fifth the effective protuberance height.

(2) The turbine efficiency decreases in proportion to the frictional resistance of its equivalent sand roughness for the near-normal and larger discharge.

Fig. 19 Efficiency versus frictional loss calculated by equivalent sand roughness on runner passage

(3) With an increasing roughness, the maximum efficiency point shifts to the lower speed and the lower discharge.

(4) Concerning the admissible roughness, the Schlichting's formula $k_{adm} = 100\nu/V$ is in good agreement with this experiment.

(5) When a Francis turbine has more gentle protrusions, such as wavy surface on the runner vanes, the efficiency is affected only by the increase of the skin friction due to the wavy protrusions.

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Experimental Studies on the Air Curtain*

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The first air curtain apparatus in Japan was installed several years ago. Since then, it has been used in places extensively with many practical experiences on its utility and performance, and a further wide range of application is promised.

However, little has been studied concerning the operating principle and the performance of the air curtain, particularly, the effects of the outside wind and the outlet opening on the floor are fully taken into account.

In this respect, comprehensive tests and analysis have been made so as to find how the air curtain is deflected and how much outdoor air blows into the room under various conditions of geometrical configuration of air curtain and outside wind velocity.

As a result, general instructions for designing the air curtain have been made available, which are reported herein.

1. Introduction

Since the first air-curtain apparatus was produced for experimental purpose in Switzerland about ten years ago⁽¹⁾, the apparatus was also constructed and used practically in our country, presenting many practical informations about its utility and performance⁽¹⁵⁾. Basic studies of the flow pattern and the performance of it, however, have hardly been made as yet. The main reasons may be that the direct application of a curved jet to a mechanical equipment has not been so developed as to make

it an important practical problem and that it is very difficult to obtain the exact analytical solution to be compared with experiments, especially for the general curved jet affected not only by finite width of the slot but also by side pressure and outlet opening as in an air curtain apparatus.

Now, one of the most important problems in designing the air curtain is how to estimate its resistance to the cross wind. If the criteria of jet velocity, spouting angle, depth of the air curtain etc. required to intercept the cross wind can be obtained, the room will be easily protected from the outside wind. Moreover, the room condition with an unvarying air curtain apparatus also can be held constant only by knowing the entering mass flow due to cross wind.

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