Effect of surface roughness on hydraulic loss in spiral case

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ABSTRACT
The increment of frictional deficiency by roughening the inner wall of spiral case is extracted from the optimum specific hydraulic energy deficiencies of a model Francis turbine that was measured by changing surface roughness of the spiral case. Assuming the free vortex flow in the spiral case, the frictional deficiency is predicted for the smooth and roughened spiral cases. By comparing the predicted deficiency with the measured increment of deficiency due to the roughness increase, the admissible roughness for the spiral case and the roughness conversion coefficient from the arithmetic mean roughness to the equivalent sand roughness are investigated.

Introduction
The surface roughness of the components in hydraulic turbines may affect the friction loss and the hydraulic performances, if the roughness exceeds the admissible roughness under the given Reynolds number. There is a little paper, so far, that systematically investigates the effect of the surface roughness of the various components in hydraulic turbines. To investigate the effect of surface roughness on the hydraulic performance of a Francis turbine, a series of model tests was executed in IMHEF, EPFL under the test heads of 10 and 5 m with changing the roughness of spiral case, stay vanes/guide vanes, runner and draft tube in the order.1
This paper tries to extract the optimum energy efficiency $\eta_{\text{opt}}$ from the measured hydraulic efficiency $\eta_h$ for both the smooth and roughened spiral cases. Then, the effect of the roughness in the spiral case on the optimum energy deficiency $\delta_{\text{opt}}$ ($=1.0 - \eta_{\text{opt}}$) is evaluated carefully. Assuming the free vortex flow in the spiral case, the frictional deficiency is predicted along a representative stream line for the smooth and roughened spiral cases. By comparing the predicted deficiency with the measured increment of deficiency due to the roughness increase in the spiral
case, the admissible roughness for the spiral case and the roughness conversion coefficient from the arithmetic mean roughness to the equivalent sand roughness is investigated.

**Frictional deficiency obtained with model tests**

**Model turbine and surface roughness**

The tested model Francis turbine has the reference radius $R_{r}$ of 0.200 m at runner outlet. The test Reynolds number was changed by executing the performance measurements under the test heads of 10 m and 5 m. With the model turbine having the hydraulically smooth surfaces of 0.3 to 0.5 Ra from the spiral case inlet to the draft tube outlet, the hydraulic performance characteristics of discharge coefficient $\phi$, specific hydraulic energy coefficient of turbine $\psi$, output power coefficient $\Pi$ and hydraulic efficiency $\eta_h$ were measured as a reference data. It is called as the all-smooth test. Then the only spiral case was roughened to 5 Ra. The hydraulic performances with the roughened-case tests were successively measured to evaluate the roughness effect of the spiral case.

**Hydraulic characteristics of tested model**

As an example of a series of the test results with the different surface roughness, Fig. 1 shows the characteristic diagram of the specific hydraulic energy coefficient $\psi$ versus the discharge coefficient $\phi$ for the roughened-case test at the various guide vane angles from 12.5 to 32.5 degrees under the test head of 10 m. Similarly, Fig. 2 and 3 illustrate the diagrams of hydraulic efficiency $\eta_h$ versus $\phi$, and $\eta_h$ versus $\psi$, respectively. The diagrams with other tests show similar tendency. Among the measured hydraulic characteristics, the energy coefficient $\psi$, discharge coefficient $\phi$, output power coefficient $\Pi$ and hydraulic efficiency $\eta_h$ are calculated with the following equations:

\[
\psi = \frac{2E}{(\omega R_{r})^2}
\]

\[
\phi = \frac{Q/nR_{r}^2}{\omega R_{r}}
\]
\[ II = \frac{2P}{\rho\omega^3 R_{ef}^5} \]  
\[ \eta_h = \frac{\omega T}{\rho QE} \]  

![Graph 1: Energy vs discharge characteristics for Roughened-case H=10m](image1)

![Graph 2: Hydraulic efficiency vs discharge characteristics for Roughened-case H=10m](image2)
where $E$ (J/kg) is the specific hydraulic energy of the turbine, $\omega$ (rad/s) the angular speed of rotation, $Q$ (m$^3$/s) the turbine discharge, $P$ (W) the runner output power and $\rho$ (kg/m$^3$) the density of water, respectively.

*Surface roughness and best efficiency point*

The best efficiency operating points $\psi_{bep}$ and $\phi_{bep}$ for the respective tests are
collectively shown in Fig. 4 under the test heads of 10 m and 5 m. The roughness of spiral case does not affect the $\psi_{wr}$ and $\phi_{wr}$ within the measured inaccuracy. The representative best efficiency coefficients $\psi_{wr}$ and $\phi_{wr}$ are selected as 1.25 and 0.307, respectively.

Energy deficiency obtained with model tests

To find the increment of energy deficiency with the surface roughness of the spiral case, the specific hydraulic energy deficiency $\delta_{t}$ has to be analyzed from the above measured characteristic diagrams. Since the discharge specific speed $n_{o}$ is computed as 74.0 at the representative best efficiency point, the best discharge efficiency $\eta_{Q_{best}}$ and the power efficiency $\eta_{L_{best}}$ can be estimated according to the Ref.(2). The discharge and power efficiencies $\eta_{Q}$ and $\eta_{L}$ at an arbitrary test point can be calculated with the $\eta_{Q_{best}}$ and $\eta_{L_{best}}$ by assuming that the leakage discharge coefficient $\phi_{l}$ and the disc/cylindrical friction power coefficient $\Pi_{f}$ are constant irrespective of operating points as follows:

$$\begin{align*}
(1-\eta_{Q})\phi &= (1-\eta_{Q_{best}})\phi_{best} \quad (5) \\
(1-\eta_{L})\Pi &= (1-\eta_{L_{best}})\Pi_{best} \quad (6)
\end{align*}$$

The energy efficiency $\eta_{e}$ can be computed from the measured hydraulic efficiency $\eta_{Q}$ by using the above $\eta_{Q}$ and $\eta_{L}$ with the following equation;

$$\eta_{e} = \frac{\eta_{L}}{\eta_{Q} \eta_{L}} \quad (7)$$

Finally, the energy deficiency $\delta_{t}$ can be determined at an arbitrary test point as follows;

$$\delta_{t} = 1- \eta_{e} \quad (8)$$

Incidentally, the runner energy coefficient $\psi_{r}$ and the runner discharge coefficient $\phi_{r}$ at an arbitrary operating point can be obtained from the measured $\psi$ and $\phi$ by the following equations;

$$\psi_{r} = \eta_{e} \psi \quad (9)$$
\( \phi_R = \eta_Q \phi \)

**Extraction of optimum energy deficiency**

As an example of the runner characteristic diagrams of \( \eta_R - \phi_R \) thus obtained, Fig. 5 shows the energy efficiency characteristics with the roughened-case under the test head of 10 m. An envelope curve of \( \eta_{Env\phi} - \phi_R \) can be drawn that comes in contact with the various \( \eta_R - \phi_R \) curves for the different guide vane openings. The bold curve from the lower left to the upper right in Fig. 1 corresponds to the envelope curve that is the optimum operating condition with changing discharge in Fig. 5.

By plotting a new curve of energy deficiency (with \( \phi \) basis) \( \delta_{Env\phi} = 1 - \eta_{Env\phi} \) versus the square of runner discharge \( \phi_R^2 \) corresponding to the above envelope curve as shown in Fig. 6, a tangential straight line \( \delta_{Env\phi} \) to the deficiency curve can be drawn through the origin of the diagram. The tangential point determines the optimum discharge \( \phi_{R_0} \) and its deficiency \( \delta_{Env} \).

Similarly, the optimum runner specific hydraulic energy \( \psi_{R_0} \) and its \( \delta_{Env} \) can be determined based on the envelope curve to the \( \eta_R \) versus \( \psi_R \). As imagined from Fig. 3, however, the number of \( \eta_R \) versus \( \psi_R \) curves to be enveloped is smaller especially in the higher \( \psi_R \) region. Therefore, the optimum deficiency \( \delta_{Env} \) based on the \( \psi_R \) seems to be for reference only. The nearly vertical bold line in Fig. 1 corresponds to the optimum operating condition with changing runner energy \( \psi_R \).

![Fig. 5 Energy efficiency vs runner discharge](image-url)
**Effect of spiral case roughness on optimum energy deficiency $\delta_{Eo}$**

The optimum deficiency $\delta_{Eo}$ extracted in Fig. 6 is proportional to the square of discharge only, and does not include the deficiency that is out of proportion to the discharge such as the runner shock loss and swirl loss, etc. It is, therefore, preferable to investigate the variation of frictional deficiency i.e. the effect of roughness. The increment in the optimum deficiency between the $\delta_{Eo-rough}$ obtained with the roughened case and the $\delta_{Eo-smooth}$ obtained with the all smooth components, $\Delta \delta_{Eo} = \delta_{Eo-rough} - \delta_{Eo-smooth}$ can be evaluated as the increase of friction deficiency by roughening the spiral case.
The extracted optimum specific hydraulic energy deficiency and the increment of deficiency due to the roughness increase are summarized in Table 1. The extracted optimum deficiencies amount the values from 5.7 to 6.3% of the test head for the model turbine with all smooth components. Comparing the test head of 10 m and 5 m under the $\phi$ basis, the deficiency under the lower head is higher irrespective of roughness conditions. It is also confirmed that the optimum deficiency of the roughened case is higher than that of the smooth case by 0.1% at the higher head of 10 m under the $\phi$ basis. On the other hand, there is actually no incremental deficiency due to the roughness increase under the lower head of 5 m.

Under the $\psi$ basis, the qualitative tendency of the deficiency is similar, however, the quantity of incremental deficiency of 0.3% is very large. Because of the reason that was mentioned above, the values under the $\phi$ basis will be analyzed mainly.

<table>
<thead>
<tr>
<th>Head (m)</th>
<th>Base</th>
<th>$\delta_n$ of All smooth</th>
<th>$\delta_n$ of Roughened case</th>
<th>Increment of $\delta_n$ by roughness</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>$\phi$</td>
<td>0.0571</td>
<td>0.0582</td>
<td>0.0011</td>
</tr>
<tr>
<td>10</td>
<td>$\psi$</td>
<td>0.0583</td>
<td>0.0617</td>
<td>0.0034</td>
</tr>
<tr>
<td>5</td>
<td>$\phi$</td>
<td>0.0630</td>
<td>0.0635</td>
<td>0.0005</td>
</tr>
<tr>
<td>5</td>
<td>$\psi$</td>
<td>0.0623</td>
<td>0.0653</td>
<td>0.0030</td>
</tr>
</tbody>
</table>

Frictional deficiency obtained numerically

*Velocity distribution in spiral case*

The shape of spiral case used is shown in Fig. 7 having the inlet diameter $D_3$ of 0.474 m. Figure 8 shows the distribution of meridional cross sectional area (nondimensioned with the inlet area $A_{inlet}$) around the circumferential angle $\theta$ (starting at the inlet of nose stay vane). The case has the non-circular cross section at the end of spiral, therefore, the area decreases rapidly at the angle larger than 330 degs.
Assuming the flow of free vortex in the case, the center (representative) stream-line can be obtained by finding the vertical line that divide the discharge in half in the respective cross sections of the spiral case as shown in Fig. 7.

The representative velocity along the center stream-line $V_{\text{center}}$ is obtained by assuming the constant circulation around the spiral in that the product of circumferential velocity component $V_{\theta,\text{center}}$ and the radius from the center of runner shaft is conserved. The radially inward velocity component $V_{r,\text{center}}$ at an arbitrary
angle can be obtained from the relevant angle $\theta_s$ as follows;

$$ V_{r-cone} = V_{o-cone} \tan \theta_s $$

(11)

The velocity along the representative stream-line can be determined with the following equation;

$$ V_{cone} = \sqrt{V_{o-cone}^2 + V_{r-cone}^2} $$

(12)

The distribution of the representative velocity (non-dimensioned with the inlet velocity $V_{inlet}$) around the circumferential angle is shown in Fig. 9. The velocity increases with the angle up to the 1.6 times of the inlet velocity at the spiral end.

![Fig. 9 Velocity distribution along center stream-line](image)

Frictional resistance coefficient and admissible roughness

Defining the cross sectional area $A_s$ and the perimeter $S_s$ of the spiral case as shown in Fig. 10 at an arbitrary circumferential angle, the hydraulic mean depth $m_s$ can be calculated as follows;

$$ m_s = \frac{A_s}{S_s} $$

(13)

The equivalent diameter $D_s$ of the spiral case is determined with the following equation;
Figure 11 shows the distribution of equivalent diameter (non-dimensioned with the inlet diameter $D_s$) versus the circumferential angle that does not decrease linearly. The frictional resistance coefficient for the spiral case $\lambda_s$ is assumed to be applied with Eq. (15) by Colebrook and White;

$$\frac{1}{\sqrt{\lambda_s}} = -2\log\left(\frac{ks/D_s}{3.71} + \frac{2.51}{Re\sqrt{\lambda_s}}\right)$$

(15)
where $k_s$ is the height of grain for equivalent sand roughness, and $Re$ the Reynolds number based on the mean flow velocity and the diameter of the pipe. Since the surface roughness of the model spiral case is measured with the arithmetic mean roughness $Ra$, the roughness conversion coefficient $C_s (= k_s/Ra)$ is necessary. There is a variety of values in the roughness conversion coefficient $C_s$. Assuming $C_s = 4.2$ and using the Reynolds number at the case inlet as a reference Reynolds number, the frictional resistance coefficient $\lambda_s$ is calculated with Eq. (15). Table 2 shows the values of $Ra$, $k_s$, $\lambda_{s,\text{calc}}$ with Eq. (15) and $\lambda_{\text{smooth}}$ obtained with the universal friction formula for the smooth pipe.

### Table 2 Roughness and resistance coefficient

<table>
<thead>
<tr>
<th>Roughness condition</th>
<th>head (m)</th>
<th>$Ra$</th>
<th>$k_s$</th>
<th>$\lambda_{s,\text{calc}}$</th>
<th>$\lambda_{\text{smooth}}$</th>
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</thead>
<tbody>
<tr>
<td>Smooth</td>
<td>10</td>
<td>0.5</td>
<td>2.10</td>
<td>0.01122</td>
<td>0.01110</td>
</tr>
<tr>
<td>Roughened</td>
<td>10</td>
<td>5</td>
<td>21.0</td>
<td>0.01217</td>
<td>0.01113</td>
</tr>
<tr>
<td>Smooth</td>
<td>5</td>
<td>0.5</td>
<td>2.10</td>
<td>0.01187</td>
<td>0.01178</td>
</tr>
<tr>
<td>Roughened</td>
<td>5</td>
<td>5</td>
<td>21.0</td>
<td>0.01267</td>
<td>0.01181</td>
</tr>
</tbody>
</table>

Fig. 12 Variation of frictional resistance coefficient vs circumferential angle
The value of $\lambda_{\text{eler}}$ for the smooth case is close to $\lambda_{\text{smooth}}$. It implies that the surface roughness of 0.5 Ra is nearly admissible roughness. Since the $\lambda_{\text{eler}}$ of roughened case is higher than $\lambda_{\text{smooth}}$, on the other hand, the roughness of 5 Ra exceeds the admissible roughness for the spiral case. The distribution of the frictional resistance coefficient for the segments of the spiral case is shown in Fig. 12. For the smooth case, the region except the spiral end is hydraulic smooth under the test head of 5 m, however, the effect of roughness may slightly appear under the test head of 10 m. On the other hand, the roughened case shows the clear effect of roughness from the start of spiral irrespective of the test heads.

**Frictional deficiency in spiral case**

The frictional loss in a segment of the spiral case $dH_s$ is assumed to be calculated with the Darcy-Weisbach's Eq. (16) as follows:

$$dH_s = \lambda_s \frac{R_{\text{eler}} d\theta V^2}{D_c 2g}$$

where $d\theta$ is the infinitesimal circumferential angle and $g$ the acceleration due to gravity.

The overall frictional loss head $\Delta H_s$ in the spiral case is obtained by integrating $dH_s$ from 0 degree at the front side of the nose stay vane to the rear side.

$$\Delta H_s = \int_0^{\theta_1} dH_s = \int_0^{\theta_1} \lambda_s \frac{R_{\text{eler}} V^2}{D_c 2g} d\theta$$

(17)

The frictional deficiency in the spiral case $\delta_{\text{fr}}$ is calculated against the turbine net head $H$ as follows:

$$\delta_{\text{fr}} = \frac{\Delta H_s}{H}$$

Table 3 shows the predicted frictional deficiency in the spiral case $\delta_{\text{fr}}$. The value of the deficiency $\delta_{\text{fr}}$ amounts 0.64 % (for 10 m head) and 0.75 % (for 5 m head) in the smooth case, and 0.70 % (for 10 m head) and 0.75 % (for 5 m head) in the roughened case. These values correspond to 11 % of the turbine deficiency $\delta_{\text{t}}$, in the smooth case, and 12 to 13 % in the roughened case.
The predicted increment of deficiency by roughening the surface roughness of the spiral case is compared with the test values in Table 4. The good agreement implies that it is pertinent to estimate the roughness conversion coefficient $C_\alpha$ of 4.2, as well as the admissible roughness of 0.5 Ra.

**Table 3** Friction deficiency in spiral case

<table>
<thead>
<tr>
<th>Head(m)</th>
<th>$\delta_\alpha$ of smooth</th>
<th>$\delta_\alpha$ of roughened</th>
<th>Increment of deficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.00639</td>
<td>0.00748</td>
<td>0.00108</td>
</tr>
<tr>
<td>5</td>
<td>0.00700</td>
<td>0.00754</td>
<td>0.00054</td>
</tr>
</tbody>
</table>

**Table 4** Increment of deficiency by roughness

<table>
<thead>
<tr>
<th>Head(m)</th>
<th>$\Delta \delta_\alpha$ of tests</th>
<th>$\Delta \delta_\alpha$ of calculation</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.0011</td>
<td>0.00108</td>
</tr>
<tr>
<td>5</td>
<td>0.0005</td>
<td>0.00054</td>
</tr>
</tbody>
</table>

**Concluding remarks**

The increment of frictional deficiency by roughening the spiral case can be extracted from the optimum specific hydraulic energy deficiencies of a model Francis turbine measured changing surface roughness of the spiral case. On the other hand, the frictional deficiency in the spiral case is predicted under the assumptions of the roughness conversion coefficient $C_\alpha$ of 4.2, as well as the admissible roughness of 0.5 Ra.

(1). The optimum energy deficiency $\delta_{sa}$ extracted from the measured performance diagrams amounts the values of 5.7 % (for 10 m head) and 6.3 % (for 5 m head) of the test head for the model turbine with all smooth components, and 5.8 % (for 10 m head) and 6.4 % (for 5 m head) with the roughened case.
(2). The calculated frictional deficiency $\delta_{fr}$ amounts by 0.64 % (for 10 m head) and 0.75 % (for 5 m head) in the smooth case, and by 0.70 % (for 10 m head) and 0.75 % (for 5 m head) in the roughened case. These values correspond to 11 % of the turbine deficiency $\delta_{ra}$ in the smooth case, and 12 to 13 % in the roughened case.

(3). The predicted increment of frictional deficiency by roughening the spiral case is compared with the measured one. As a result, it is pertinent to assume the roughness conversion coefficient $C_4$ of 4.2 as well as the admissible roughness of 0.5 Ra for the relevant spiral case.

References

2. JSME standard S008 (under the revision).