Experimental investigation to determine equivalent sand grain for very rough hydraulic surfaces

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Abstract. Considering the actual roughness shape, distribution and orientation found on hydraulics surfaces, obtaining a sand grain equivalent for loss evaluation remains a challenge. To investigate this problem, a variety of surfaces ranging from hydraulically smooth to a very rough surface representing a heavily rusted penstock were tested on an experimental cylinder rotating in water. The larger-scale roughness was replicated through rapid prototyping. The surface effect on the flow for an increasing Reynolds number was measured and the resulting equivalent sand grain was validated against experimental intake head loss measurements.

1. Introduction and some theory

Despite representing an important proportion of the overall losses in many turbines, roughness has received little attention lately. It is generally acknowledged that improving surface finish through mechanical means (cleaning, smoothing and painting) can be beneficial for both power output and efficiency. However, because of the associated costs, the potential return on investment must be thoroughly evaluated before moving forward with such work. This is not an easy task. To gain more insight into the subject, Hydro-Quebec decided to investigate the problem by combining fluid, experimental and material expertise. This paper presents some of the findings of that work.

In the literature, roughness is generally approached through an equivalent sand grain k_s description as suggested by Schlichting [13]. To qualify the roughness regime, sand grain size can be expressed in a dimensionless way as in (1) with the kinematic viscosity (v) and the friction velocity u* (2).

\[ k_s^+ = \frac{k_s u^*}{v} \quad (1) \]
\[ u^* = \sqrt{\frac{\tau_w}{\rho}} \quad (2) \]

Figure 1 explains the generally acknowledge theory of roughness. In the turbulent regime, the surface is considered smooth when \( k_s^+ < 5 \) and becomes fully rough whenever \( k_s^+ > 70 \). In the latter regime, the friction coefficient has the interesting characteristic of taking a finite value independent of the Reynolds number. A transition exists between the smooth and the fully rough regimes, sometime exhibiting inflectional behavior like the one observed by Nikuradse [12], but more generally showing asymptotic behavior as observed by Colebrook [4]. Theodorsen [14] later explained this discrepancy by using variable size roughness elements and demonstrated that there was departure from the smooth curve as soon as the largest asperities tipped outside the viscous sublayer. The inflectional behavior is thus associated with very uniform or manufactured roughness elements. While some studies, like the one of Flack [8], recently revisited the limits of the transition region to lower values, the theory mostly hold true to this day.
To evaluate the friction-related losses in a pipe or hydraulic channel (3), one can use the Darcy friction coefficient ($f$), which can be obtained with the implicit Colebrook and White equation (4) or its graphical version, the Moody diagram [10]. However, both require a priori knowledge of the equivalent sand grain roughness, $k_s$.

$$ h_f = f \frac{L U^2}{D \sqrt{g}} $$

$$ \frac{1}{\sqrt{f}} = -2 \log_{10} \left( \frac{k_s / D}{3.7} + \frac{2.51}{\text{Re} \sqrt{f}} \right) $$

It is sometimes possible to obtain information about roughness for long penstocks, as global intake losses are measured during efficiency tests. However, the quantity that is measured includes a range of energy dissipating phenomena such as the trash rack, intake transition, elbow, surge chamber, restriction and manifold. To isolate the roughness effect, it is important to quantify those local effects. For older powerhouses, we have seen an increase in intake losses within 10-20 years after commissioning due to surface degradation. Properties of water, concrete and metal but also of coating are thought to be the principal parameters determining degradation speed and the final characteristics of the surface. Figure 2 is an example of what an old penstock can look like.

Figure 2: Close up view of penstock “roughness” after 50+ years of service.

Roughness assessment is even more complex for other parts of the turbine, namely the distributor, runner and draft tube, as there is no easy way to isolate the surface losses from the other quantities measured during efficiency tests. Looking at Figure 3, is it obvious that there are major surface-related losses in this turbine, but it is not clear exactly what the $k_s$ of the surfaces is, or even what the roughness
is in this situation. There are features, such as the large dents on the stay vanes that should probably be considered as geometry.

Figure 3: Spiral casing and distributor “roughness” after 50+ years of service.

One way to obtain the impact of roughness is to perform efficiency tests before and after surface smoothing. Such tests are costly, however, and often have an uncertainty margin of about the same order of magnitude as the estimated roughness impact. Another possibility is to rely on the literature. Yet the values tend to be available mostly for new surfaces; data for older, worn out surfaces are very sparse and it is thus difficult to use them with confidence. Another way to obtain $k_s$ is to use an expression involving surface characterization. A final option is to investigate roughness through an experimental setup, which was the approach taken for this article.

1.1. Conceptual setup

Our setup is presented schematically in Figure 4. Its goal is to obtain friction losses on the surface of a rotating cylinder. It is similar to the setups of Theodorsen in 1944 [14] and Dierich [6]. If we neglect secondary losses (mostly those generated at the tips), the power required to rotate the setup is linked to the wall shear of the cylinder (5). Once the wall shear is known, it is possible to determine the Fanning friction coefficient, designated as $C_f$ (6), with the circumferential velocity (7). While the Darcy friction factor is mostly used for pipes, the Fanning coefficient is more commonly used with flat plates or external flow. Its value equals 1/4 of the Darcy friction factor (8)[11]. Putting the friction coefficient back into the Colebrook equation (4), we can finally estimate an equivalent sand grain size. We can make sure we are in the fully rough regime by verifying that an asymptotic value is reached or by looking back at the $k_s^+$. While the wall shear at a given speed is not a characteristic of the surface, the equivalent sand grain is, so it is possible to use this value to estimate the hydraulic losses.

\[
P = \frac{\tau_w \pi D^2 L \omega}{2} \quad (5) \quad \tau_w = \frac{C_f \rho U^2}{2} \quad (6) \quad U = \omega R \quad (7) \quad C_f = \frac{f}{4} \quad (8)
\]

The specific goals of the experiment were:
- To validate the setup for smooth and mildly rough surfaces
- To measure the geometric characteristics of the surfaces
- To identify equivalent sand grain for very rough surfaces
- To compare the values obtained with the available data
2. Surfaces characteristics

Many researchers, including ourselves [16], have tried to directly relate measurable surface characteristics with sand grain equivalent. As mentioned by Bons [2], however, most results remain case-specific. International standard IEC 62097 [3] deals with model-to-prototype performance scale-up and suggests what might be the most popular relation between average roughness $R_a$ and $k_s$ (9). One of the problems with such a formulation is that, despite some recommendations applicable to new surfaces [1], there is no clear indication of how to measure $R_a$ and which cutoff length should be used to separate surface geometry, waviness and roughness for real surfaces with large-scale features.

$$k_s = 5R_a \quad (9)$$

To validate our setup, we created a smooth aluminum cylinder 304.8 mm long with a 152.4 mm radius. It was polished to a mirror finish and installed on a testbench developed for robotized grinding research. After a first set of tests, its surface was increasingly roughened (three times) by the use of abrasive grit. To assess some of the surface characteristics, average roughness ($R_a$), maximal surface amplitude ($R_z$) and quadratic roughness ($R_q$) were measured with a Mitutoyo SJ-201 portable roughness tester. Equations for calculating roughness are shown below (10-12). Measurements were taken both axially and circumferentially at 20 points (4 on the circumference at 5 axial positions), for a total of 40 measurements per cylinder. No cutoff filter was applied to the measurements. Results are shown in Table 1, which clearly shows the difference between axial and circumferential roughness for the larger abrasive grit size, highlighting the difficulty of obtaining adequate roughness measurements, even for simple geometries.

$$R_a = \frac{\sum_{i=1}^{n} z_i}{n} \quad (10) \quad R_z = z_{max} - z_{min} \quad (11) \quad R_q = \sqrt{\frac{\sum_{i=1}^{n} z_i^2}{n}} \quad (12)$$

In 2009, a field campaign was started to assess the degradation state of our turbine hydraulic surfaces. Pictures and samples of various components (mostly penstocks, spiral casings, guide vanes and runners)
were taken in a total of nine different powerhouses. To characterize the surfaces, replicas were taken with polysiloxane from 3M and brought back to the lab for further analysis. Figure 5 shows a replica taken on a penstock for later testing on our setup.

![Figure 5: Sample taken from a penstock.](image)

Once in the lab, a 75 mm x 75 mm patch at the center of the imprints was measured with a CHR 150 optical profilometer from Micro Photonics to perform additional analysis of the surface characteristics. Measurements were taken every 100 µm to provide a high definition 3D surface (Figure 6).

![Figure 6: Penstock sample surface measurement.](image)

The plan was to use this surface to create a rotating cylinder with rapid prototyping. As it was not possible to have a large enough surface from the sample to cover the desired circumference and height for our cylinder, the surface patch was mirrored axially and circumferentially to create small sections like the one showed in Figure 7. Four sections were stacked to reach the intended height. Later, a second rough cylinder was created by scaling the roughness height by a factor of 0.25.

![Figure 7: Rough cylinder section build from prototyping based on the penstock surface.](image)

|     | $R_a$ [mm] | $R_z$ [mm] | $R_q$ [mm] |
|-----|------------|------------|------------|
| Penstock | 1.80       | 10.03      | 2.06       |
3. Experimental setup and measurement

3.1. Experimental setup

The rotating cylinder was attached to a submersible permanent magnet synchronous electric motor with maximum power of 3.5 kW capable of reaching 6,000 RPM and a maximum torque of 6 Nm. The rotating speed could be controlled with an accuracy of about 1%. To limit tip losses and allow the setup to be placed at the bottom of the water tank, rigid plates were fixed on each side of the cylinder, leaving a gap of only 1.2 mm. The experimental setup is shown in Figure 8.

The water tank was a 1,040 x 2,080 mm rectangle with a height of 1,070 mm. During the tests, the water level in the tank was kept at 910 mm, which provided a water volume of about 2 m³. A filtration system was installed on the tank to make sure the water remained clean during the tests. The first tests generated waves at the surface, which increased in amplitude with power consumption and introduced noticeable oscillations in the power output. A rigid floating cover was therefore used to stabilize the free surface of the tank for the remainder of the tests.

![Experimental setup with rough cylinder.](image)

As power measurement is at the heart of the experiment, two different methodologies were used:
- measuring the electrical input power of the motor with a Yokogawa WT300 wattmeter and subtracting all motor-related losses (copper, iron, mechanical), and
- measuring the current, rotation speed and temperature of the K089200-5Y electric motor, estimating the power using the motor torque constant and subtracting iron and mechanical losses.

Both methods are valid at high speed but, due to internal loss assumptions, become less reliable at low speed, when there is a higher proportion of internal losses/total power. The average of both methods was used for the final results. To obtain stable measurements, the power output was averaged over a period of 400 to 900 s. The following theoretical losses (13-15) from Volkov [15] were used to estimate the tip losses associated with the upper and lower surfaces of the cylinder:

\[
C_M = \frac{M}{\frac{1}{2} \rho \omega^2 R^5} \quad (13) \quad \text{Re}_{\text{Disk}} = \frac{\omega R^2}{V} \quad (14) \quad C_M = 0.040 G^{-1/6} \text{Re}_{\text{Disk}}^{-1/4} \quad (15)
\]

Where \( G = \text{(distance between rotating disk and a fixed wall)/R} \).
3.2. Results

The setup with the smooth surface was first validated. Experiments by Theodorsen [14] and Dierich [6] on a rotating cylinder were used as a reference. Both made the assumption that the wall shear obtained from their setup was valid because the thickness of the boundary layer was small compared with the radius of the cylinder. From Figure 9a, we can see that the results of all the experiments superimpose. The friction factor from Colebrook [4] and Nitchawitz [3] obtained from pipe flow are added to the plot to confirm that the effect of the Reynolds number (Re) on the wall friction is the same for a pipe flow or a rotating cylinder. As mentioned previously, the better fit of our experiment at higher Re was expected based on the assessment of internal losses.

![Figure 9: C_f for increasing Re a) smooth surface b) grinded surface.](image)

Looking at the three roughnesses obtained by abrasive grinding (Figure 9b), we can see that the TrizacA300 departs slightly from the smooth curve at higher Re without reaching a plateau. For the other two grinded cylinders (flexcut Gr36 and Cup Gr 24), there is clear separation from the smooth curve. The Gr 36 shows almost a constant friction value while the Gr 24 shows an increase in $C_f$ at first, which might be attributable either to measurement errors at lower Re or to Nikuradse-type inflectional behavior in the transition regime. While it is not clear whether the fully rough regime was reached, the setup behaved properly, allowing us to move forward with the realistic surface from the penstock.

We can see in Figure 10 that the friction values obtained for the penstock printed roughness are higher by an order of magnitude when compared with the others. This is not surprising given the size of the roughness elements. Because of the increased flow resistance and motor power limit, the setup could not reach a high rotating velocity.
Figure 10: $C_f$ for printed roughness.

Table 2 presents the results of the final measured power for each cylinder, along with an analysis to estimate the friction coefficient, equivalent sand grain and roughness regime. It confirms that the smooth cylinder was actually smooth and that the Gr 24 and 36 cylinder did not reach the fully rough regime. With $k_s +$ over 70, however, the two realistic printed surfaces are definitely in the fully rough regime and can thus provide an estimate of the equivalent sand grain of the surface roughness.

| Cylinder          | D [m] | Max RPM | $R_a$ [μm] | Re max | $U^*$ [m/s] | $C_f$ [N/m²] | $K_s$ [μm] | $K_s/R_a$ | $K_s+$ |
|-------------------|-------|---------|------------|--------|-------------|--------------|-----------|----------|--------|
| smooth            | 0.16576 | 2089   | 0.027      | 2.743E+06 | 0.577       | 0.00248      | 0.17       | 6.24     | 0.09   |
| Trizact A300      | 0.16574 | 2051   | 1.15       | 2.701E+06 | 0.644       | 0.00262      | 2.49       | 2.16     | 1.46   |
| Flexcut Gr36      | 0.16565 | 1950   | 2.68       | 2.524E+06 | 0.652       | 0.00297      | 10.3       | 3.84     | 6.10   |
| Cup 6° Gr24       | 0.16555 | 1893   | 5.35       | 2.779E+06 | 0.652       | 0.00330      | 22.3       | 4.18     | 13.25  |
| penstock z/4      | 0.16510 | 1607   | 454        | 2.094E+06 | 0.673       | 0.00469      | 131        | 0.29     | 80.20  |
| penstock          | 0.16510 | 747    | 1815       | 9.690E+05 | 0.707       | 0.02392      | 14768      | 8.14     | 9490   |

Table 2: Roughness regime for the different surfaces.

It is interesting to look at the $k_s/R_a$ relationship between the measured average roughness and the estimated equivalent sand grain. We can see that while the IEC estimate (9) seems adequate for most surfaces, it is off by an order of magnitude for the penstock z/4 surface. The plausible explanation for this is that the reduction in height of the asperities made them fall into the waviness regime. The flow is thus able to move around them without generating the typical micro-wakes associated with roughness.

4. Analysis

It is not easy to validate whether the values obtained previously are accurate, especially the estimated $k_s$ for the penstock roughness. Table 3 provides an estimation of the sand grain for the various speeds that were tested. We can see that all tests provided a $k_s$ of between 13.63 and 14.77 mm, for an average of 14.0 mm.

| Re               | 639828 | 708031 | 769927 | 838648 | 905685 | 968955 |
|------------------|--------|--------|--------|--------|--------|--------|
| Ks (mm)          | 13.63  | 13.55  | 14.42  | 13.61  | 14.15  | 14.77  |

Table 3: Estimated penstock roughness for the various speeds.

A first validation of the values obtained can be done based on the literature, as shown in Table 4. With the $k_s$ obtained, Darcy friction coefficients can be estimated for a 5.03 m (16.5 ft) pipe. Manning
coefficients $n$ from the civil sphere are also included. They can be converted into a Darcy coefficient using equation (16). Also included are some geometry-based formulas, such as those of IEC and Flak. Overall, we can see that the results generated by our setup are plausible.

$$f = 12.7 \frac{n^2 g}{D^{1/3}} \quad (16)$$

| Description                                      | $k_s$ [mm] | Manning ($n$) | $f$  |
|--------------------------------------------------|------------|---------------|------|
| With iron bacterial corrosion, heavily rusted [9]| 3.0 - 4.0  | n/a           | 0.0186 |
| Heavily corroded, with incrustation from 3 to 25 mm [9] | 6.0 - 6.5 | n/a           | 0.0209 |
| Riveted steel [10]                                | 0.9 - 9    | n/a           | 0.0227 |
| Steel, old – tuberculated [5]                     | n/a        | 0.013 - 0.020 | 0.0123 - 0.0291 |
| Steel – riveted [5]                               | n/a        | n/a           | 0.0144*(216/16.5) |
| CEI 62097 (5*R_o) [3]                             | 9.0        | n/a           | 0.0227 |
| Flack (4.43*R_e(1+Sk)^1/3)) [7]                   | 12.25      | n/a           | 0.0247 |
| Penstock (the present setup)                      | 13.55 - 14.77 | 0.0188 - 0.019 | 0.0254 - 0.0260 |

**Table 4: Comparison of the penstock roughness value with values from the literature.**

Another way to validate our results is to compare the intake head losses that would be obtained using $k_s$ from our setup with what was measured during efficiency testing. The complete intake head loss evaluation is presented in Table 5. A $k_s$ of 1.5 mm was assumed for the concrete part based on experimental values obtained from an 11 km intake channel (we are fairly certain of this value), which includes a combination of local ($K_L$) and length-related losses ($fL/D$). For this powerhouse, head losses of 2.77 m and 2.88 m were measured on two different but similar units, for a flowrate of 177 m/s.

| Description                                      | Coefficient ($K_L+fL/D$) | Speed [m/s] | Losses [m] |
|--------------------------------------------------|--------------------------|-------------|------------|
| Intake                                           | 0.1                      | 6.06        | 0.187      |
| Inclined concrete channel, 20’ dia, length 377’  | 0.0144*(377/20)          | 6.06        | 0.509      |
| Elbow 40 deg, 20’ dia, r/d = 4.5                  | 0.05+0.0144*(62/20)      | 6.06        | 0.177      |
| Horizontal concrete channel, 20’ dia, length 280’ | 0.0144*(280/20)          | 6.06        | 0.378      |
| Transition from 20’ to 16.5’ dia, length 30’      | 0.05+0.0149*(30/18.25)   | 7.48        | 0.201      |
| Shielded penstock, 16.5’ dia, length 216’         | 0.0256*(216/16.5)        | 8.91        | 1.356      |
| **Total**                                        |                          | **2.81 m**  |            |

**Table 5: Estimation of head losses.**

We can see that the estimated losses match the measurements quite well. About half the total losses (1.356 m) are linked to the rusted surface of the metal penstock. Another way to look at these results is to compare the losses for this part with those that would be obtained from a perfectly smooth surface ($f = 0.0067$ at this Re). We then see that that losses of 0.98 m can be attributed to the roughness itself, thus representing the potential improvement achievable for a smoothing project.

5. **Conclusion**

Our rotating cylinder setup demonstrated that it could be used to evaluate equivalent sand grain roughness from a realistic surface. This opens the way to a better understanding of surface-related losses for refurbishment projects.

Results of wall shear stress tests for the very rough surface of a rusted penstock showed that it was equivalent to a roughness formed by closely packed 14 mm sand grain. This value is consistent both with what is seen in the literature and what was measured during efficiency tests.

We have yet to determine what the next step should be to better understand roughness and take its effects into account in Hydro-Quebec’s practices. Possible options are:
• Continue to evaluate the shape factor formulas stemming from the geometric characteristics of the surface (i.e., IEC 62097, shape factor, Flack, Yuan, etc.), with a focus on roughness orientation in the flow and the cutoff filter that should be used for measurement
• Use CFD to obtain friction coefficients from our database samples
• Perform additional experiments to obtain more information on typical hydraulic roughness

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