Francis turbine with tandem runners: a proof of concept

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Abstract. Hydraulic turbines operating at off-design regimes inherently have a large residual swirl at runner outlet. This swirling flow is further ingested by the draft tube and its deceleration in the discharge cone results in self-induced instabilities (e.g. precessing vortex rope, low frequency pressure pulsations, power swing) that may hinder the turbine operation. Moreover, for low head Francis turbines the draft tube losses sharply increase as the operating point departs from the best efficiency one, practically shaping the efficiency hill chart. We arrive at the conclusion that an effective approach to flatten the hill chart, while mitigating the instabilities in the draft tube cone is to adaptively adjust the swirling flow exiting the runner for variable operating regimes. As a result, a novel concept of tandem runners is introduced, whereas the classical Francis runner is conceptually split into a high-pressure runner with a constant speed and a low-pressure runner with variable speed. We explore this concept using realistic Francis turbine tandem cascades, with constant and variable transport speed, respectively.

1. Introduction
For medium-high specific speed Francis turbines the efficiency hill-chart shape is practically given by the sharp increase of the hydraulic losses in the draft tube [1] as the operating point departs from the best efficiency one. Moreover, the pressure fluctuations also increase sharply especially at partial discharge [2], with synchronous, asynchronous and random components, hindering the smooth turbine operation. These phenomena are essentially the result of the flow ingested by the draft tube, with a swirl component too large with respect to the discharge velocity as intuitively shown by a simple analysis of the velocity triangles at the trailing edge of the runner blades. The simple model for hydraulic turbine operation within a wide operating range developed and validated in [3] and [4] clearly shows the inherent increase of the flux of moment of momentum downstream the Francis runner away from the best efficiency regime. Moreover, the flux of moment of momentum remains practically unchanged in the bladeless region downstream the runner while the flux of axial momentum decreases in the discharge cone of the draft tube. As a result the ratio of two fluxes that defines the swirl number increases as the swirling flow is decelerated in the discharge cone [5], triggering the self induced instabilities such as the well known precessing helical vortex [6].

Recent theoretical investigations of the decelerated swirling flow instabilities [7] led to the determination of a purely theoretical body force distribution in the discharge cone that practically removes the vortex rope by minimizing the dominant unstable eigenmode of the time averaged mean flow. Although there are no indications on how such a body force might be practically achieved, it is obvious from its spatial distribution that the water jet injection from the runner crown along the turbine axis [8] can be seen as a technologically feasible implementation. The water jet injection concept, first introduced and numerically proved in [9] was experimentally shown to effectively mitigate the swirling flow instability in the discharge cone [10].
Other approaches for stabilizing the decelerated swirling flow in the discharge cone at off-design operating regimes aim at hindering the rotating component of the flow by using fins attached to the cone wall or by providing shallow grooves mounted parallel to the pressure gradient on the diffuser wall [11]. These so-called “J-grooves” were further optimized in [12] for suppressing the draft tube surge in Francis turbines.

All the approaches described above address the swirling flow exiting a conventional Francis runner. A different approach is proposed in [13] for mitigating the loss of turbine efficiency and strong pulsations in the draft tube in all turbine operating regimes other than optimum.

According to figure 1, the Exit Stay Apparatus proposed in [13] practically introduce a stator immediately downstream the Francis runner to reduce the swirl velocity component entering the draft tube. A possible design approach for the stay blades downstream the runner is also presented in [13].

In this paper we introduce and examine a different approach for maintaining the swirling flow ingested by the draft tube close to the optimal configuration (i.e. with minimum draft tube losses and maximum pressure recovery) while the turbine operating point spans a wide range of discharge values. More precisely, instead of using stationary blades downstream the Francis runner, we propose a variable speed runner in tandem with the main constant speed runner. Our concept is functionally different from various counter-rotating tandem-runner axial machines such as the bulb turbine [14], the counter-rotating micro-turbine [15], or the counter-rotating pump-turbine [16]. Using an additional axial runner in tandem with the main radial-axial runner has also been proposed in [17] for a radial-axial pump-turbine (the RAPT concept), but in this case both runners are installed on the same shaft and rotate with the same speed. Section 2 presents the concept of the tandem runner turbine, and Section 3 examines the basic hydrodynamics of its operation. The last section summarizes the main conclusions.

2. The Francis turbine with tandem runners
The classical Francis turbine design includes in the bladed region, figure 1, the stationary stay vanes and guide vanes (shown in blue) and a radial-axial runner with fixed pitch blades (shown in magenta). The swirling flow exiting the runner, usually optimized for a peak overall efficiency with corresponding minimum draft tube losses, dramatically departs from the optimum configuration at off-design operating points with abrupt increase in draft tube hydraulic losses and severe flow instabilities.
In order to keep the swirling ingested by the draft tube optimal no matter the operating regime (or at least in a large operating range) we introduce a variable speed runner, colored in red in figure 1, with the speed correlated with the operating point. As a result, the turbine will have:

a) a high pressure runner (HPR), with constant speed, similar to the regular/classical Francis turbine runner; the HPR generates the whole turbine power at design operating point, and most of the power at part load;

b) a low pressure runner (LPR), with variable speed, that generates a fraction of total turbine power at part load; the LPR is designed to work at runaway speed at the turbine design operating point, where it practically generates no power at all (vanishing torque).

The variable speed of the low pressure runner adds the required flexibility in the Francis turbine regulation in order to avoid the draft tube performance deterioration at off-design. Moreover, as further shown in the paper, LPR generates only a small fraction from the turbine nominal power, and as a result its associated electrical generator could employ a rather large range for speed.

A good source of inspiration for designing the LPR blades are the very high specific speed Francis or propeller runners. An innovative design, named “Mixer”, is developed in [18] for refurbishing very old low head Francis turbines. Although the “Mixer” runner has no band, in our case a runner band/rim must be provided in order to accommodate the variable speed generator.

From technological point of view the low-pressure runner should be fitted with a rim generator attached to its own band. At part load the rim generator insure the regenerative braking of the LPR, by converting the inherent excess or the kinetic swirl energy into electricity instead of feeding it to the draft tube where it is mostly dissipated. This technology has been developed for STRAFLO (STRAight-FLOW) turbines [19] for low-head conditions, and in contrast to bulb turbines most of the generator components are kept out of the water with a rim-lip-seal [20]. For vertical axis machines the rim generator can also provide magnetic sustentation for the LPR.

3. A simple proof of concept
In order to illustrate the operation of the tandem runner turbine, we will examine the basic problem of two hydrofoil cascades working in tandem, with different tangential speeds. We consider a realistic cascade as obtained by intersecting the Francis runner with an axisymmetric streamsurface in the crown neighborhood, as shown in [21, Ch.7], and further projecting it into the conformal plane.

![Figure 2. Flow kinematics for tandem cascades at the rated operating point.](image1)

![Figure 3. Flow kinematics for tandem cascades at half discharge from rated point.](image2)
The analysis domain, figures 2 and 3, corresponds to a periodic strip with three hydrofoils for the upstream cascade (corresponding to the HPR) and two hydrofoils for the downstream cascade (originating from the LPR). The two subdomains move with different transport speeds $U_1$ and $U_2$, respectively, being coupled at the interface using the sliding mesh technique. The pitch angle for the downstream symmetric foils is chosen such that they are aligned with the relative flow exiting the upstream cascade at the rated operating point, figure 2.

Figure 2 shows the flow kinematics, i.e. the streamlines for the relative flow and the velocity triangles at the rated operating point. The upstream flow, in reality provided by the turbine distributor, has an absolute velocity with axial component $V_{1\text{ axial}} = 3.0 \text{ m/s}$ and tangential component $V_{1\text{ tang}} = 0.156 \text{ m/s}$ and $V_{1\text{ tang}} = 0.132 \text{ m/s}$. The transport velocity for the upstream cascade is $U_1 = 4.0 \text{ m/s}$ resulting in an optimum angle of attack for the relative velocity $W_1$. The absolute flow exiting the upstream cascade, $V_2$, is practically axial, with negligible tangential component. The pitch angle for the downstream cascade corresponds to the angle of the relative velocity, $W_{2s}$. However, when examining the flow into the downstream cascade we noticed that the transport velocity should be slightly adjusted to $U_2 = 4.1 \text{ m/s}$ to correspond to the runaway speed, with vanishing tangential force. This is why the relative velocity entering the downstream cascade, $W_{2s}$, has a slightly different angle with respect to $W_{2s}$. At this operating point there is no relative flow deflection in the downstream cascade, thus $V_3 \approx V_2$, or to be more precise $V_{2\text{ tang}} = 0.156 \text{ m/s}$ and $V_{3\text{ tang}} = 0.132 \text{ m/s}$.

The partial discharge regime shown in figure 3 corresponds to half discharge and the same head. As a result, the inlet velocity components are $V_{1\text{ axial}} = 1.5 \text{ m/s}$ and $V_{1\text{ tang}} = 4.0 \text{ m/s}$, resulting in the same velocity magnitude as in the previous case, as it should be when operating the turbine at constant head [3]. As a result, the relative velocity $W_1$ becomes axial, with a positive angle of attack for the upstream cascade. The transport velocity for the upstream cascade is kept $U_1 = 4.0 \text{ m/s}$, since the high pressure runner works at the synchronous generator speed. In this case, the absolute flow exiting the upstream cascade, $V_2$, has a large tangential component $V_{2\text{ tang}} = 2.07 \text{ m/s}$. It is this high level of swirl, compared to the discharge velocity, that is ingested by the draft tube further increasing the hydraulic losses the triggering the flow self-induced instabilities. The recuperative braking of the downstream cascade down to $U_2 = 2.6 \text{ m/s}$ practically recovers this swirl excess leaving $V_{3\text{ tang}} = 0.6 \text{ m/s}$ at the draft tube inlet. The relative streamlines in figure 3 clearly show the loading of the downstream cascade. Note that at the interface the relative streamlines have an angular point corresponding to the change in the transport velocity from $U_1 = 4.0 \text{ m/s}$ to $U_2 = 2.6 \text{ m/s}$.

The above analysis illustrates the benefits of the recuperative braking of the downstream runner, from a kinematical perspective. From the dynamic point of view, we show in figures 4 and 5 the variation of the total pressure (mechanical energy per unit volume) through the two tandem cascades. According to figure 4, at the rated operating point the whole energy conversion takes place in the upstream cascade, i.e. the high pressure runner with synchronous speed. The low pressure runner practically does not extract energy from the fluid since it is rotating at its runaway speed, practically equal by design to the synchronous speed.

At partial (50%) discharge, however, both runners contribute to the energy conversion, while keeping the same turbine head. The total pressure decrease in the downstream cascade accounts for the recuperative braking that kicks in at part load. The values of the total pressure, as shown in figures 4 and 5, simply illustrate the fact that both operating points have the same total pressure drop, or
constant turbine head, but the actual value of the head is irrelevant for the present discussion. We recall at this point the main difference with respect to the Stay Vane Apparatus proposed in [13], where the downstream cascade is fixed, thus acting as a stator in an attempt to convert the excess or swirl into static pressure. However, the recuperative runner proposed in this paper contributes to the full hydraulic energy conversion in the turbine, while keeping the optimum swirl at draft tube inlet.

Figure 4. Total pressure at rated operating point. Energy conversion only in the upstream cascade.  

Figure 5. Total pressure at half the discharge and the same head as the rated point. Both cascade extract energy from the fluid.

Table 1. Mechanical power extracted by the runners, percentage from the rated power.

|                       | Upstream cascade | Downstream cascade |
|-----------------------|------------------|--------------------|
| High Pressure Runner (HPR) | 99.3%            | 0.7%               |
| Low Pressure Runner (LPR)   |                  |                    |
| Partial discharge operating point, total 50% rated power. | 33.7%            | 16.3%              |

Table 1 summarizes the mechanical power extracted from the fluid by each runner, for the two operating points examined above. At the rated operating point, practically all energy is extracted by the high pressure runner at synchronous speed, while the low pressure runner operates at the runaway speed. At 50% discharge and the same head, 2/3 from the total mechanical power is delivered by the high pressure runner, and further transferred to the main synchronous generator, and 1/3 is recovered by the low pressure runner and transferred to the variable speed rim generator.

4. Conclusions

The paper introduces a concept for double regulated Francis turbine, by adding to the classical design a low pressure runner, with variable speed, immediately downstream the high pressure runner (typical Francis runner) with constant speed. At the rated operating point the LPR is designed to work at
runaway speed, equal to the high pressure runner synchronous speed, without altering the swirl ingested by the draft tube. However, at partial discharge the rim electrical generator of the LPR provides regenerative braking, thereby converting the excess of swirl kinetic energy into electricity. Since the power provided by the variable speed rim generator is only a rather small fraction, no more 10%...15%, from the main synchronous generator rated power, its frequency converter could provide braking speed as low as 60% from the synchronous speed at part load regimes.

The main advantage of employing a regenerative braking low-pressure runner is that the swirling flow ingested by the draft tube could achieve a near-optimum configuration within a wide range of operating regimes. As a result, for high specific speed Francis or propeller turbines the draft tube losses are kept to a minimum thereby flattening the efficiency hill chart. In addition, the flow instabilities in the draft tube are altogether avoided by providing a stable swirl configuration at the draft tube inlet within the whole turbine operating range. The resulting efficient and smooth operation of the hydraulic turbine insures the robust flexibility required by the integration of the new and highly fluctuating renewable energy sources.

Appendix
The details of the numerical computation are summarized in this appendix in order to keep the flow of the article. Numerical simulations are performed using the FLUENT 16.2 expert code, with a basic $k-\varepsilon$ turbulence model.

In order to define the averaged quantities on sections $S_1$ – inlet, $S_2$ – interface, and $S_3$ – outlet one starts with the conservation of the relative total pressure on streamlines of the relative flow

$$\vec{W} \cdot \nabla \left( \rho \bar{p} - \rho \vec{U} \cdot \vec{V} \right) = 0,$$

where $\rho \bar{p} = \rho V^2/2$. Integrating over the upstream or downstream domains, and using the Gauss theorem, one obtains

$$\int_S \left( \rho \bar{p} - \rho \vec{U} \cdot \vec{V} \right) \vec{W} \cdot \vec{n} dS = 0,$$

where $\vec{n}$ is the outer normal pointing outwards, $\vec{W} \cdot \vec{n}$ is the discharge component of the velocity, and the scalar product $\vec{U} \cdot \vec{V}$ is usually written as $UV_{\text{tang}}$.

It is obvious from equation (2) that since $\vec{W} \cdot \vec{n}$ vanishes on the blade surface, and the surface integrals over the periodic boundaries cancel each other, we are left only with the integrals over the sections mentioned above. The corresponding integrals correspond to the fluxes of the quantities in the brackets, thus one must define their flux weighted averages

$$\bar{p}_{\text{tot}} Q = \int_{S_1} \rho \bar{p} \vec{W} \cdot \vec{n} dS, \quad \rho Q \left( UV_{\text{tang}} \right) = \int_S \rho \vec{U} \cdot \vec{V} \vec{W} \cdot \vec{n} dS,$$

where $Q \equiv \int_S \vec{W} \cdot \vec{n} dS$. (3)

From the fundamental equation of turbo-machines, the mechanical power on the moving blades is

$$P_m \equiv U \sum F_{\text{tang}} = \rho Q \Delta \left( UV_{\text{tang}} \right),$$

where, as an example, $\Delta \left( UV_{\text{tang}} \right) = U_{1\text{tang}} - U_{2\text{tang}}$ for the upstream cascade. The tangential force $F_{\text{tang}}$ is reported from the FLUENT postprocessing tools for integral quantities. Examples for the numerical values obtained with either $U \sum F_{\text{tang}}$ or $\rho Q \Delta \left( UV_{\text{tang}} \right)$ are shown in table 2.
Table 2. Mechanical power extracted by the runners, from equation (4).

| Rated operating point, total 100% rated power | Upstream cascade | Downstream cascade |
|---------------------------------------------|------------------|-------------------|
| $P_{m1} = 5961.7(5976.4)W$                  | $P_{m2} = 44.4(51.7)W$ |
| Partial discharge operating point, total 50% rated power | $P_{m1} = 2021.4(2027.5)W$ | $P_{m2} = 978.4(1003.7)W$ |

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