Dynamic Behaviour of Reversible Pump-Turbines in Turbine Mode of Operation

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ABSTRACT

A reversible pump turbine is a machine that can operate in three modes of operation i.e. in pumping mode, in turbine mode and in phase compensating mode (idle speed). Reversible pump turbines (RPTs) have an increasing importance for regulation purposes for obtaining power balance in electric power systems. Especially in grids dominated by thermal energy, reversible pump turbines improve the overall governor stability. Increased use of new renewables (wind, wave and tidal power plants) will utterly demand better regulation ability of the more traditional power suppliers, enhancing the need for reversible pump turbines.

It is, however, a necessity that both machine and system is designed for maximum flexibility in changing modes of operation from pump to turbine and vice versa. A reversible pump turbine is known for having very steep speed-flow characteristics which might give stability problems.

During the design process, focus on the system dynamics in such power plants is essential. Simulation of the dynamic behaviour of the system implies a good representation of the turbine by its characteristics. It is necessary to be able to represent the characteristics before any measurements of the actual machine are available.

This paper presents an possible explanation of the reason for these steep characteristics. Based on this hypothesis, a representation of the turbine for system dynamic simulation purposes is suggested.

Symbol | Quantity | Unit
--- | --- | ---
D1 | Inlet diameter, turbine mode | m
D2 | Outlet diameter, turbine mode | m
B | Runner height at inlet in turbine mode | m
H | Head | m
Hth | Theoretical pump head | m
β1 | Inlet angle, turbine mode | -
ω | Angular speed | 1/s
v | Relative velocity | m/s
s | Diameter relation | -
u1 | Peripheral velocity at inlet, turbine mode | m/s
u2 | Peripheral velocity at outlet, turbine mode | m/s
g | Gravitational constant | m/s²
κ | Guide vane opening degree | -
Q | Discharge | m³/s
cu | Absolute velocity in peripheral diameter | m/s
c | Absolute velocity | m/s
Qc | Discharge with zero incident losses | m³/s
Hp | Head for pump | m
Hn | Head at nominal speed | m

Introduction

The special geometry of the reversible pump turbine gives an effect of pumping in operation outside the best performance point in turbine mode. Hence, from a global point of view, the turbine will have an increased throttle effect as the rotational speed increases. As the speed increases, the flow decreases more than that of a Francis turbine with the same specific speed. This is due to the difference in geometry. A reversible pump turbine is a compromise between a turbine and a pump, where the priority is on the performance in pumping mode of operation. The steep characteristics might cause severe stability problems, especially at idle speed. Stability in idle speed is a necessity for phasing in the generator to the electric grid. Also severe water hammer pressure is known to be caused by these steep characteristics. As the turbine increase the speed because of a load rejection, the wicket gates go towards closure, giving retardation pressure in front of the turbine.

In all reaction turbines, the speed of rotation will have an influence on the flow. In general, a low specific speed Francis turbine will have steeper characteristics than a high specific speed Francis. This is mainly due to the acting centripetal forces. Dependent of the ratio between inlet and outlet diameter, the pumping effect will be different in different machines. For a low specific speed Francis, the inlet diameter is much larger than the outlet diameter, hence the centripetal forces works against the driving pressure. This results in a throttling effect, which decreases the flow when the speed of rotation increases. If the outlet diameter is the larger one, the effect will be opposite, the flow increases when the speed of rotation increase. In Figure 1 and Figure 2, this is illustrated.
By developing Euler’s turbine equation, eq. (1), one can see that the speed of rotation affects the pressure so that when the inlet diameter is larger than the outlet diameter, the turbine runner will give increased throttling effect as the speed of rotation increases.

According to the Euler turbine equation, the pressure difference between inlet and outlet (marked 1 and 2 in Figure 3), is given by:

\[ g(H_1 - H_2) = u_1 c_{1w} - u_2 c_{2w} \]  

Equation (1)

Applying the cosine rule on the inlet velocity triangle, Figure 4, gives:

\[ v_1^2 = u_1^2 + c_1^2 - 2u_1 c_1 \cos \alpha_1 = u_2^2 + c_1^2 - 2u_1 c_{1w} \]  
\[ u_1 c_{1w} = \frac{1}{2} c_1^2 - \frac{1}{2} v_1^2 + \frac{1}{2} u_1^2 \]  

Equation (2)

And accordingly for the outlet triangle, Figure 4:

\[ u_2 c_{2w} = \frac{1}{2} v_2^2 - \frac{1}{2} v_1^2 + \frac{1}{2} u_2^2 \]  

Equation (3)

Implemented in eq. (1) and substituting \( u \) with \( \omega r \) gives:

\[ g(H_1 - H_2) = \frac{1}{2} (c_1^2 - c_2^2) - \frac{1}{2} (v_1^2 - v_2^2) + s \omega^2 \]  

Equation (4)

Where \( s \) is the diameter relation defined as:

\[ s = \frac{1}{8} D_2^2 (1 - \frac{D_1^2}{D_2^2}) \]  

Equation (5)

Introducing the opening degree of the turbine, \( \kappa \), defined by:

\[ \kappa = \frac{Q}{\sqrt{2gH}} \]  

Equation (6)

where \( n \) denotes the design point of the turbine. Solving the equation with respect to the head \( H \) gives:

\[ H = H_n \left( \frac{Q}{\kappa Q_n} \right)^2 \]  

Equation (7)

which in the design point is the same head as \( H_1-H_2 \), as expressed in eq.(4), hence:

\[ gH_n \left( \frac{Q}{\kappa Q_n} \right)^2 = \frac{1}{2} (c_{1n}^2 - c_{2n}^2) - \frac{1}{2} (v_{1n}^2 - v_{2n}^2) + s \omega_n^2 \]  

or

\[ \left( \frac{1}{2} (c_{1w}^2 - c_{2w}^2) - \frac{1}{2} (v_{1w}^2 - v_{2w}^2) \right) = gH_n \left( \frac{Q}{\kappa Q_n} \right)^2 - s \omega_n^2 \]  

Equation (8)

Implemented in eq. (1) gives:

\[ gH = gH_n \left( \frac{Q}{\kappa Q_n} \right)^2 + s (\omega^2 - \omega_n^2) \]  

Equation (9)
When $s$ is positive, i.e. $D_1>D_2$, the flow will decrease and if $s$ is negative, i.e. $D_1<D_2$, the flow will increase as the speed of rotation increases, see Figure 5.

![Figure 5 - Illustration of the tendency of the speed-flow characteristics.](image)

Figure 5 - Illustration of the tendency of the speed-flow characteristics.

This is all according to the Euler turbine equation. Simulations of the turbine characteristics and efficiency of a high head Francis turbine with this model are shown in Figure 6 and Figure 7. It can be observed that the flow characteristics end up in one point for the different guide vane opening. As there is no loss involved, the peak efficiency will be 1 for best operational point, $\kappa=1$.

For high head Francis turbines, and especially for RPTs, the speed-flow characteristics are in fact much steeper. Even for low head RPTs, where the inlet diameter is equal or even smaller than the outlet diameter, the speed-flow characteristics is decreasing. This indicates that it is not only the diameter ratio that decides the characteristics.

The flow dependency of the speed of rotation is often referred to as “pumping effect” because as the speed increases, the turbine will act like a pump, throttling the flow through the turbine. There are reasons to model the turbine in accordance with the Euler turbine equation, adding a term based on the pump equation in order to include this “pumping effect”.

### Pump characteristics

A centrifugal pump and a Francis turbine is in fact the same kind of machine, both obeying the Euler equation. Assuming rotation free inlet, the theoretical head of a pump may be expressed by the Euler equation:

$$gH_{th} = u_z(u_2 - \frac{c_{2m}}{\tan \beta_2}) = u_z(u_2 - \frac{u_2 - \frac{Q}{\pi B_2 D_2 \tan \beta_2}}{\tan \beta_2})$$

(10)

At a given speed of rotation, i.e. for a given $u_2$. The QH-characteristics will be ascending if the outlet angle $\beta > 90^\circ$ and descending if $\beta < 90^\circ$, i.e. forward or backward leant runner blades as illustrated in Figure 8. In order to get stable pump characteristics, centrifugal pumps must have backward leant runner blades. This is also the case for a RPT, when running as a pump. However, when the speed of rotation changes direction to turbine mode of operation, the pump effect will be even stronger because of forward leant blades.

In eq. (10), it is assumed no rotation at the pump inlet. For this case, there is rotation at the inlet because of the guide vanes. This rotation term is included in eq. (11) below.

$$gH_{th} = u_z(u_2 - \frac{c_{2m}}{\tan \beta_2}) - u_1 - \frac{Q}{A_2 \tan \beta_2}$$

$$gH_{th} = u_z(u_2 - \frac{Q}{A_2 \tan \beta_2}) - u_1 - \frac{Q}{A_1 \tan \beta_1}$$

(11)

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Pumping effect in a Francis turbine

Because of the pumping effect in a Francis turbine, the turbine will behave in the same way as a centrifugal pump. The water flows through the turbine from upper to lower reservoir and will give rotational power on the turbine shaft according to the Euler turbine equation. At the same time, the pump equation will also apply, but with the direction of the rotation as in turbine mode of operation. Seen from the pump equation, the flow will be negative until the flow really turns, which will happen if one increases the speed of rotation sufficiently. As pump regarded, this will be a pump with forward leant blades, i.e. with increasing characteristics, (see Figure 8). Using notations common for turbine mode, the pump equation will be:

\[
gH_p = u_1 c_{1a} - u_2 c_{2a} = u_1 \left( \frac{c_m}{\tan \beta_1} \right) - u_2 \left( \frac{c_m}{\tan \beta_2} \right)
\]

(12)

Introducing the angular speed of rotation, \( \omega \):

\[
gH_p = \omega^2 \left( \frac{Q}{A_2 \tan \beta_2} \right) - \left( \frac{Q}{A_1 \tan \beta_1} \right) \]

(13)

The first term is identical with the term that one gets when developing the turbine equation, see eq. (9). The last terms enhance the pumping effect dependent on the difference in diameters and the blade angle. Implemented in eq.(9), adding the term \( H_p \), gives:

\[
gH = gH_s \left( \frac{Q}{\kappa Q_s} \right)^2 + s(\omega^2 - \omega_0^2) - (\omega - 1)R_s (Q - Q_s)
\]

(14)

\[
R_s = \frac{D_1}{2} \frac{1}{A_1 \tan \beta_1} - \frac{D_2}{2} \frac{1}{A_2 \tan \beta_2}
\]

(15)

The same Francis turbine as in Figure 6 in is now simulated with the pumping effect included. The turbine characteristics are now steeper, as shown in Figure 9.

Design principles for reversible pump-turbines

Bigger difference between inlet and outlet diameter gives a higher pumping effect and thereby steeper characteristic is to be anticipated. The difference of diameters does not explain that the characteristics become significant steeper. Also low head RPTs, where the diameters are nearly equal, have very steep characteristics. The explanation must be on the blade angle i.e. the outlet blade angle seen from the pumping point of view.

In RPTs, the blade angle is back leant in pumping mode of operation in order to have stable pump characteristics. Compared with a turbine with given head, flow and speed of rotation, the outlet diameter of a pump must be increased to be able to pump the same flow back to the upper reservoir. See Figure 11.
In turbine mode, the pumping part of the runner is somewhat unnecessary. In order not to disturb the turbine behaviour, it is reasonable to design the addition in diameter with neutral. Figure 12 shows simulated RPT characteristics based on this model. The speed-flow characteristics are much steeper than for a Francis turbine, see Figure 9.

![Figure 12 - Simulated speed-flow characteristics of RPT](image)

**Simulated and measured characteristics**

The new model shows reasonable good agreement with measured characteristics. The simulated characteristics are plotted against measured data from the corresponding model turbine in Figure 10.

![Figure 13 - Comparison for different opening degrees](image)

**Conclusion**

The model seems to have good overall agreement with measured characteristics, as Figure 13 indicates. The model is a quadratic algebraic equation and it is then impossible to have an s-shaped curve as in the measured characteristics. There are reasons to believe that there is a hysteresis effect in the measurements for the s-shaped region and the s-shape is subject to discussion among turbine experts. As the flow approaches zero, the deviation between the model and the measured characteristic curve is highest. At least, the similarity is good enough for a priori simulations, i.e. before the turbine design is complete and no model measured characteristics are available.

**References**

Dörfler, P.K., el al “Stable operation achieved on single-stage reversible pump turbine showing instability at no-load operation”. Proceeding of the 19th Symposium on Hydraulic Machinery and Systems (Singapore, 1998)