Pressure pulsations and hydraulic efficiency at Smeland power plant

V S Ulvan*, J O Kverno and O G Dahlhaug
Department of Energy and Process Engineering Norwegian University of Technology and Science, Trondheim, Norway
* Corresponding author (vegarsul@student.ntnu.no)

Abstract. Smeland power plant in Norway is experiencing pressure pulsations in their Francis turbine when running above best efficiency point. By measuring both the pressure pulsations and runner efficiency, the cause and effect of the pulsations are to be investigated thoroughly, which is this work’s main purpose. To find the Francis runners efficiency the thermodynamic method has been used, which builds on the principle that all of the hydraulic losses turns into heat in the flow itself. By measuring the change of temperature before and after the turbine one can, with little other data, calculate the hydraulic efficiency. To identify the pressure pulsations, pressure transducers were placed on the inlet to the spiral casing, draft tube, and upper labyrinth. While doing measurements, air-injection through the runner was tested on full load, which nearly eradicated the pressure pulsations. This might be due to an increase of volume in a pulsating full load vortex that changed its eigenfrequency, and therefore stopped resonating.

1. Introduction

Table 1. Smeland power plant characteristics [4]

| Characteristic      | Data            |
|---------------------|-----------------|
| Runner              | Low head Francis|
| Head                | 95 m            |
| Installed power     | 24 MW           |
| BEP                 | 20 MW           |
| Annual production   | 119 GWh         |
| Owner               | Agder Energi    |
| Built               | 1985            |
| River system        | Mandalsvassdraget |

A hydro power plant in Vest-Agder County, Smeland power plant, is experiencing pressure pulsations in their turbine. The pulsations they have registered were of a low frequency variety, 2.8 Hz, which indicated that the the problem may be in the draft tube. The turbine is equipped
with a check valve which lets air into the draft tube at sub-atmospheric pressures, however this proved to be blocked. The owners of the power plant, Agder Energi, deemed the pressure pulsation matter great enough to seek help from the Waterpower Laboratory at NTNU in order to find out what is causing this phenomenon. By taking measurements of both the hydraulic efficiency and the pressure pulsations one might be able to discern and find the cause of the pressure pulsations.

2. Hydraulic efficiency and the thermodynamic method

In a water turbine, energy is converted from hydraulic energy to mechanical energy which in turn drives an electric generator. However, not all of the energy is converted into mechanical energy, i.e. there are losses through the system. The turbine efficiency says how much of the available hydraulic energy that is turned into mechanic energy. The thermodynamic method builds on the principle that all of the losses in the flow going through the turbine turns into heat in the flow itself. Then the hydraulic losses in a water turbine can be accounted for by the change of temperature in the water running through it, or more accurately the change in enthalpy. By mainly measuring the temperature and pressure at the inlet and outlet, the turbine hydraulic efficiency can be found. The thermodynamic method is a method for measuring efficiency which does not require a direct measurement of the volumetric flow, which can be a very difficult parameter to measure in existing power plants. [1]

The general equation for calculating the hydraulic efficiency is found by dividing the mechanic power by the hydraulic power [3]. The subscripts denotes place of measurement in figure 1.

$$\eta_h = \frac{P_m}{P_h} = \frac{(\rho Q g)E_m}{(\rho Q g)E_h} = \frac{E_m}{E_h}$$ (1)

$$\eta_h = \frac{E_m}{E_h} = \frac{\sigma(p_{1-1} - p_{2-1}) + g(z_{1-1} - z_{2-1}) + \frac{1}{2}(c_{1-1}^2 - c_{2-1}^2) + c_p(T_{1-1} - T_{2-1})}{\frac{1}{\rho}(p_{1} - p_{2}) + g(z_{1} - z_{2}) + \frac{1}{2}(c_{1}^2 - c_{2}^2)}$$ (2)

Figure 1. Different places of measurement
3. Pressure pulsations and full load vortex

One characteristic of hydraulic turbines and pipe flow is pressure pulsations, as the system is dynamic and the flow is often unstable to some extent. These instabilities do however tend to reach a point of equilibrium with the dampening effects of friction as the oscillation amplitude is increased. When a Francis turbine with a fixed rotational frequency operates outside of its design point, the flow leaving the Francis runner will have a rotating component. The direction and magnitude of the rotating velocity component will depend on whether the turbine is operating at part or full load, and how far off the design point it is, respectively. As a swirling flow moves through a cylinder, the bulk of the fluid transport will be along the walls, while a more stagnant region is found at the centre. If the swirl is severe enough, this stagnant flow might stop or move upstream, and a vortex breakdown occurs. [2]

When a Francis runner is operating at full load, i.e. above BEP, a symmetrical vortex appears. Due to the rotation of the flow, the bulk mass flow will occur along the walls of the draft tube, which severely increases the downwards velocity of the water. This vortex can pulsate if the frequency of this pulsation resonates with an exciting frequency from the system. This is when resonance can occur. The exact frequency of this vortex pulsation is difficult to pinpoint but it will change depending on the volume and pressure, as both are parameters that dictates the natural frequency of a gas bubble suspended in a liquid. [2]

![Figure 2. Full load vortex](image)

Figure 2. Full load vortex [5]

We can simplify the full load vortex to be a bubble. To find the natural frequency of such a bubble, the problem was simplified and regarded as an analogy to a mass spring system

\[
\omega = \sqrt{\frac{k}{m}} \rightarrow f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \tag{3}
\]

where \( k \) is the spring constant and \( m \) is the mass attached to the spring. \( k \) is defined to be

\[
k = \frac{dF}{dx} \tag{4}
\]

and could, when dealing with changing volumes and pressure, be defined as

\[
k = \frac{dp}{dV} \tag{5}
\]

\[
\Rightarrow dp = -\frac{\gamma p_0}{V_0} dV \tag{6}
\]
where $\gamma$ is the ratio of specific heats for the gas, $p_0$ is the pressure, and $V_0$ is the volume of the gas. Equation 6 stems from the derivative of the equation of state. Professor Nielsen at the Waterpower Laboratory suggested that this, combined with the momentum equation could be used to find the natural frequency of the vortex. He derived the following expression

$$f_e = \frac{1}{2\pi} \sqrt{\frac{\gamma p_0}{V_0 I}}$$  \hspace{1cm} (7)$$

where $f_e$ is the eigenfrequency of the vortex filament and $I$ is an inertial factor related to the mass and inertia of the surrounding water. It has the unit $kg/m^4$, which Nielsen suggested could be something like $\rho/l_c$, where $\rho$ is the density of the water, and $l_c$ is some length scale related to the filament. The natural frequencies for various vortex volumes were calculated, with $l_c$ being set to both the circumference, diameter and radius of the vortex cross section, and compared with the frequencies observed at Smeland power plant. Out of all of these, the diameter seems to make most sense in terms of what type of length scale that would affect the flow.

4. Measurement set-up

In order to calculate the efficiency seven sensors were used, five temperature sensors and two pressure transducers. Using a probe on the inlet bleed valve, one temperature and pressure sensor were used to find inlet temperature with corresponding pressure. One temperature sensor measured the leakage water, and the last three measured the temperature in the outlet of the draft tube. The last pressure sensor was placed directly on the inlet pipe in order to calculate the hydraulic specific energy. To find the pressure in the outlet, water column calculations were utilized using the measured height from upper draft tube floor to the water surface.

Pressure pulsation measurement were done with five pressure sensors; one upstream and downstream the main inlet valve, two on the draft tube cone 180° apart, and lastly, one on the upper labyrinth.

![Figure 3. Measurement of temperature and pressure](image)

5. Measurement procedure

Thirteen measurements were made with ten different points of operation. BEP was measured twice due to unstable temperature in the first measurement, 23 MW was done thrice to check repeatability and to test air injection.
Table 2. Sensors

| Name                                | Type     | Systematic accuracy |
|-------------------------------------|----------|---------------------|
| Oceanographic SeaBird 38           | Temperature | 0.0001°C            |
| GE Druck UNIK-5000 3bar a          | Pressure | 0.01%               |
| GE Druck UNIK-5000 5bar a          | Pressure | 0.01%               |
| GE Druck UNIK-5000 15bar a         | Pressure | 0.01%               |
| GE Druck UNIK-5000 50bar a         | Pressure | 0.04%               |

Table 3. Points of operation

| Measurement # | \(P_{\text{gen}}\) [MW] |
|---------------|--------------------------|
| 1             | 19.6                     |
| 2             | 21                       |
| 3             | 22.2                     |
| 4             | 23                       |
| 5             | 24                       |
| 6             | 6.2                      |
| 7             | 9.8                      |
| 8             | 13.25                    |
| 9             | 15.0                     |
| 10            | 16.9                     |
| 11            | 19.6                     |
| 12            | 23                       |
| 13 (w/air injec.) | 23                  |

6. Results

The pressure pulsations started right after BEP and were present toward maximum effect. The pulsations varied in amplitude and frequency, where both peaked around 23 MW. The results presented here will mainly concern the pulsations at its worst, namely at 23 MW. In the figures, \(H\) is the measured head and \(H_0\) is net head.

As seen in figure 4, the pulsations have a peak to peak value of 18% of the head and are dominating the measurements. The pulsations were of a low frequency, about 2.8 Hz.

In discussion with the staff at Smeland power plant it was revealed that the runner had a check valve designed to let air through its center, and it was decided to test air injection through the valve. Upon inspection it was found that the the valve was blocked, not allowing the draft tube to "breath". The valve was removed and a standard compressor for tools was connected and turned on with a pressure of 8-10 bar. This nearly eradicated the pulsations as can be seen in figure 5.
As the compressor ran out of air, its pressure dropped to about 2 bar. Still, the air injection nearly halved the initial pulsations as can be seen in figure 6. It was under these circumstances an additional point for hydraulic efficiency were measured. The efficiency curve can be seen in figure 7, and it is the circle that represents the additional point.

Figure 4. Pressure in draft tube, measurement no. 12

Figure 5. Air injection through the runner, measurement no. 13
Figure 6. Compressor runs out of air, measurement no. 13

Figure 7. Hydraulic efficiency

An injection of water was also tried, however, as can be seen in figure 8, this had no perceivable effect.
Figure 8. Water injection, measurement no. 13

Figure 9. Staff at Smeland checking valve

Figure 10. Compressor connection
Figure 11. Vertical cross-section of turbine showing check valve and air pathway

7. Discussion
The pressure pulsations at Smeland power plant occur at full load, and the frequency is seemingly dependent on the load. This implies that there is a full load vortex happening in the draft tube, which has its own eigenfrequency. If this eigenfrequency were to match any other pressure pulsations happening in the power plant, resonance would present itself. This other frequency could be a number of many things; pressure wave propagation in the water, von Karman vortices, natural frequency in construction elements, pole passing frequency, RSI, the list goes on.

When injecting air through the runner vertically, with no rotational element, mass in form of gas is essentially added. This would change the volume of the bubble in the draft tube cone, which in turn changes the bubbles eigenfrequency, as the eigenfrequency is a function of volume. Moving the eigenfrequency means leaving the range in which resonance would occur, as all other frequencies would stay the same.

Looking at figure 4, it looks like the pressure pulsations collapse reaching maximum value. If there really is a full load vortex present, and it collapses, then added air could function as a dampener. This dampening effect might be another reason why air injection worked against the pressure pulsations.
When it comes to the hydraulic efficiency it would seem like the pressure pulsations had no effect at all, as the efficiency is quite close to the measurements done in 1985 by the turbine supplier. One could argue that the pressure pulsations were present from day 1, and therefore the pulsations were included during the very first efficiency measurement.

The drop in efficiency during air injection is difficult to say anything certain about. When the uncertainty analysis is done the three points at 23 MW will surely overlap, and one could not conclude that air injection reduced the hydraulic efficiency. However, the uncertainty is highly determined by IECs strict standards [3], and the fact that the repeatability of the measurement without air injection gave nearly exact results several hours between is cause for concern. On the other hand the temperature of the air can influence the temperature measurement at the outlet. A more thorough look at the data is needed.

8. Further work
There is a lot of measurement data that still need to be addressed and analysed. An uncertainty analysis on the efficiency measurements are essential, and a harder look at the efficiency with air injection is needed. A spectral analysis of the pressure pulsations must be done, and will give a more clear view of all the present frequencies and their peak to peak values. Furthermore, the theory of a full load vortex pulsating with another pulsation needs to be fully investigated.

If possible a real-life simulation of a full load vortex with air injection will be done in the Waterpower Labaraorty NTNU. This might give more insight into the effects of air injection, as the draft tube cone is see-through.

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