Energy distribution at the draft tube outlet.

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Abstract

A thermodynamic measurement of hydraulic efficiency, with special attention to the energy distribution at the outlet cross section of the draft tube, is presented. The measurement was carried out at Bratsberg Power plant in Norway on a Francis turbine with a head of 130 meters. Six temperature probes and six velocity probes were used to measure the energy distribution.. The efficiency was calculated in two different ways, first by using the average, and secondly by using area-weighted velocities and temperatures. These calculations gave two efficiency curves with a difference less than the uncertainty, but more than the repeatability of the measurements. Finally, the hydraulic efficiency was calculated for each measured point, and this gave a good picture of the energy distribution. In addition, a comparison with another measurement, which used the collector method at the outlet of the draft tube, is presented.

Introduction

Thermodynamic measurement of hydraulic efficiency is a well established method to establish turbine efficiency. In Norway, this method is well known and it is widely used for high head turbines. Since 1960 Norwegian engineers have utilized this method in more than 400 measurements. The uncertainty for this kind of measurements has been disputed for a long time. H. Suzuki [4] studied the overall uncertainty of hydraulic efficiency for heads between 50 meters and 500 meters under good measuring condidtions and estimated the total uncertainty. These were presented in 1968 as shown below:

Head [m]	50	100	200	300	500
Error [%]	$\pm$ 1,9	$\pm$ 1,1	$\pm$ 0,8	$\pm$ 0,7	$\pm$ 0,6

In 1982 T.R. Ford [5] presented measurements from Dinorwig Power Plant on pump/ turbines with a head of 519 meters. He obtained an overall uncertainty in the hydraulic efficiency of $\pm$ 0,5%. In 1989, three different teams [7] from Norway measured the same turbine to control the difference of measured efficiency between the teams. The comparison test was carried out on a Francis turbine with a head of 450 meter, and the obtained overall uncertainty in the hydraulic efficiency was estimated to be $\pm$ 0,5%. The difference of the mean measured efficiency between the teams was $\pm$ 0,2%. The measurements presented in this paper are carried out on a Francis turbine with a head of 130 meters and the obtained overall uncertainty in the hydraulic efficiency was estimated to be $\pm$ 0,93% according to IEC 41[1]. Since H. Suzuki presented his estimates the uncertainty has decreased due to more precise instruments which is confirmed by the measurements mentioned above. In this paper, the emphasis will be on the uncertainty of the temperature and velocity measurements at the outlet of the draft tube. In 1972, professor Knut Alming[2] studied the specific energy distribution at the outlet of the draft tube, and concluded that one shall use a measuring point for each $1m^2$ to $4m^2$ of the cross section depending on the conditions. Leif Vinnogg [6] presented a paper describing the sources of errors that can occur when measuring temperature in the draft tube. These were the uneven temperature distribution, foreign water and backflow in the measuring section. Professor Hermod Brekke[3] presented a paper in 1992 on the energy distribution at the outlet of the draft tube. The measurement was carried out on a Francis turbine with a head of 540 meters, and the difference in measured efficiency based on the lowest and highest temperature in the draft tube outlet was 0,8%. The presented measurements has taken the work of K. Alming [2], L.Vinnogg [6] and H Brekke[3] into consideration.

General comments to expected shift in best efficiency point from model to prototype.

If the leakage flows from both upper and lower labyrinths of a Francis turbine are collected in the draft tube, all losses except for bearing loss and minor heat transfere losses will be found at the draft tube outlet. Because of this, it is important to study the distribution of the specific hydraulic energy and the thermodynamic energy TS at the draft tube outlet. According to the IEC code the average meridional kinetic energy = $(Q/A)^2/(2g)$ at the draft tube outlet is not regarded to be part of the turbine loss. However, the kinetic energy integrated across the outlet will normally be larger than the outlet energy expressed by the average kinetic energy and it is important to collect data of the temperature and velocity distribution over the cross section. Work has been presented of temperature measurements at the GPMT meeting in Salzburg and for this meeting a more detailed measurement which is part of the dr.ing. work of Ole G. Dahlhaug will be presented.

Before the presentation of the measuring results a brief discussion of the nature of the major losses in a Francis turbine will be given. The reason for this is the shift of the magnitude of the different losses as function of the turbine output. Because of decreased friction losses from model to prototype there will also be a shift in best Efficiency Point (BEP). Often a weak swirl flow in the draft tube is necessary to avoid separation and increased draft tube losses. These factors are necessary to consider when discussing measuring results and distribution of the energy distribution at the draft tube outlet.

The shift in BEP with increased friction losses.
If we compared an old or a new turbine or a prototype versus model, the friction loss will be different. As an example the change in specific hydraulic energy caused by friction through spiral casing, stay vanes runner and draft tube will increase approximately proportional to the square of the flow for a given turbine. Because of increasing Reynolds number the loss is smaller in the prototype than in a model leading to a shift in best efficiency point towards larger guide vane opening for the prototype. This is illustrated in Figure 1.

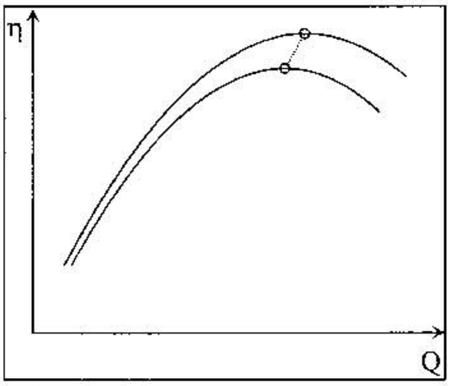

Figure 1.

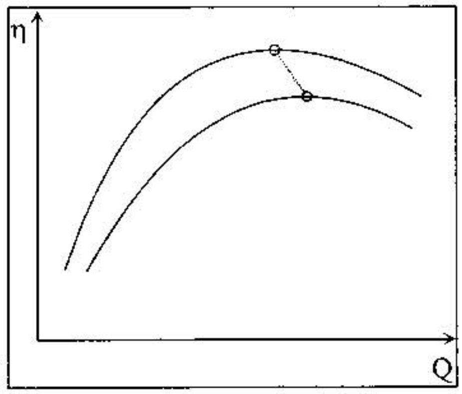

Figure 2.

The disk friction loss, however, shows a trend opposite to the flow friction loss. This is because the disk friction loss is nearly constant and not increasing with the flow. This is illustrated in Figure 2, where the best efficiency is shifted to decreased load on the prototype compared with the model. Other losses are the guide vane end clearance loss which is larger in a prototype than in a model and gives a shift towards decreasing load with increasing loss.

Influence on the energy distribution from the different losses.
In the draft tube the heated water from the losses is normally mixed at operation around best efficiency. However, at off BEP operation a skewed temperature profile has often been observed together with a skewed velocity profile and the swirl flow. According to the theory the fine turbulence from the boundary layers is converted into heat. How the swirl flow component and the large eddy losses at the draft tube outlet are measured as heat is also an interesting discussion. Because of this, measurements of both the temperature profile and the velocity profiles has been carried by Dr.ing. students Ole Dahlhaug and his colleague Roar Vennatrø in a medium head power plant. A detailed description of this work will be presented in the following.

Equipment details

Figure 3 shows the experimental setup used in the measurements of the hydraulic efficiency. There were six resistance thermometers and six electromagnetic velocimeters mounted in the outlet of the draft tube. One temperature and one pressure probe was mounted in front of the turbine. Pressure was measured with a differential pressure probe at the inlet of the spiral casing. The outlet water level at the draft tube outlet was measured manually. The setup for resistance thermometers and electromagnetic velocimeters are shown in Figure 4.

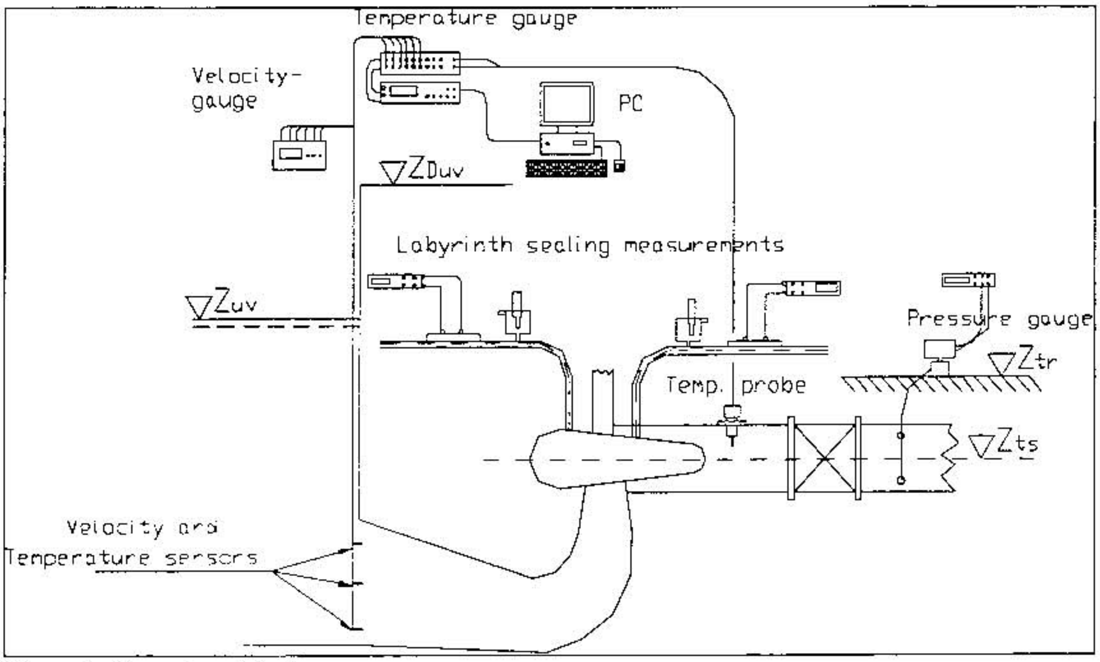

Figure 3. Experimental setup.

The temperature at the inlet was measured between the spherical valve and the spiral casing. The temperature probe was of the type PT-100. At the outlet six temperature probes of the same type was fixed to the frame as showed in Figure 4. The probes were isolated from the frame in order to avoid temperature influence from the frame, and it was used a bridge of the type Automatic System Laboratories, F-17A with an uncertainty of $\pm$ 1mK.

Velocity measurements at the outlet were carried out with six electromagnetic velocity meters of the type SENCA RMX. Measurements of the mean velocities were done over the same sample time as for the temperature measurements. The sample time for each measurement was 60 seconds for both the velocity and the temperature measurement with a measured point each second.

The inlet pressure was measured at the wall in the penstock for the calculation of hydraulic energy, and from the measuring vessel for the calculation of mechanical energy. There was used a differential pressure probe of the type Fuji, and to determine the absolute pressure a barometer of the type System Paulin was used.

The flow trough the labyrinth seals was measured with an ultra sonic flow meter of the type Fuji. The temperature in the flow was measured with an resistance thermometer of the type QUAT 100, and the setup of this equipment is shown in Figure 3.

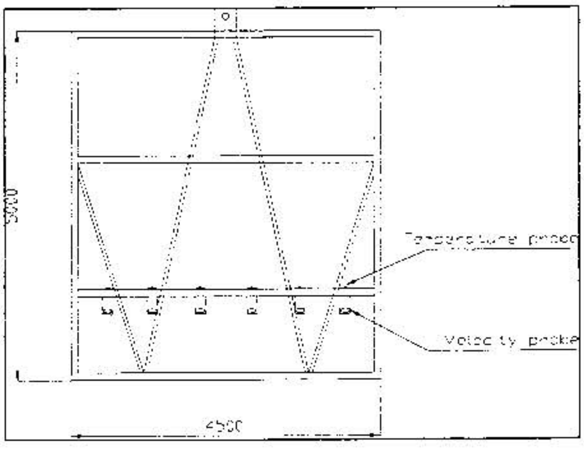

Figure 4. Frame used at the outlet of the draft tube.

Equations to describe the hydraulic efficiency.

The equations for the hydraulic efficiency are described in the International Standard, IEC 41[1], and are also shown in equations 1 to 6. In addition to the IEC equations, the modified equations for the calculation of the weighted mean efficiency from 30 points at the outlet cross section of the draft tube is shown in equations 7 and 8. The equation for hydraulic energy was not modified due to its definition in IEC 41. The hydraulic efficiency is denoted η_h and is the ratio of mechanical energy, E_m, to hydraulic energy E_h. The equation for mechanical energy is traditionally given as

$$E_m = \bar{a}\cdot\left[p_{T1} - p_{T2}\right] + \bar{C}_p\cdot\left[T_1 - T_2\right] + \frac{1}{2}\cdot\left[V_{T1}^2 - V_{T2}^2\right] + g\cdot\left[Z_{T1} - Z_{T2}\right] + \delta E_m \qquad [1]$$

where the correction term is estimated as:

$$\delta E_m = -\frac{Q_{sp}}{Q_1}\left\{\bar{a}\cdot\left[p_{sp} - p_{T2}\right] + \bar{C}_p\cdot\left[T_{sp} - T_2\right] + \frac{V_{sp}^2 - V_{T2}^2}{2} + g\cdot\left[Z_{sp} - Z_{T2}\right]\right\} \qquad [2]$$

The pressure terms are given as:

$$p_{1-t} = p_{tr} - p_{atm} + \frac{g}{\bar{v}}\cdot\left[Z_{tr} - Z_{T1}\right] \qquad [3]$$

$$p_{2-1} = \frac{g}{\bar{v}}\cdot\left[Z_{uv} - Z_{T2}\right] = \frac{g}{\bar{v}}\cdot\left[Z_{Duv} - h_{uv} - Z_{T2}\right] \qquad [4]$$

The hydraulic energy yields:

$$E_h = \bar{v}\cdot\left[p_1 - p_2\right] + \frac{1}{2}\cdot\left[V_1^2 - V_2^2\right] + g\cdot\left[Z_1 - Z_2\right] \qquad [5]$$

where the pressure term p_1 is given as:

$$p_1 = p_{tr} - p_{atm} + \frac{g}{\bar{v}}\cdot\left[Z_{tr} - Z_{ts}\right] \qquad [6]$$

The equation for weighted average mechanical energy E_m, due to the velocity and temperature measurements at the outlet cross section of the draft tube in 30 different possitions, is given as:

$$E_m = \bar{a}\left[p_{T1} - p_{T2}\right] + \overline{C_p}\left[T_1 - \sum_{j=1}^{30}\left(T_{2,j}\cdot\frac{V_{T2,j}\cdot A_j}{Q}\right)\right] + \frac{1}{2}\left[V_{T1}^2 - \sum_{j=1}^{30}\left(V_{T2,j}^2\cdot\frac{A_j}{A_2}\right)\right] + \\ \bar{g}\left[Z_{T1} - \sum_{j=1}^{30}\left(Z_{T2,j}\cdot\frac{A_j}{A_2}\right)\right] + \delta E_m \qquad [7]$$

The modified correction term yields:

$$\delta E_m = -\frac{Q_{sp}}{Q_1}\left[\begin{array}{l}\bar{a}\left(p_{sp} - p_{T2}\right) + \overline{C_p}\left\{T_{sp} - \sum_{j=1}^{30}\left(T_{2,j}\cdot\frac{V_{T2,j}\cdot A_j}{Q}\right)\right\} + \dfrac{V_{sp}^2 - \sum_{j=1}^{30} V_{T2,j}^2\cdot\frac{A_j}{A_2}}{2} + \\ g\cdot\left(Z_{sp} - \sum_{j=1}^{30}\left(Z_{T2,j}\cdot\frac{A_j}{A_2}\right)\right)\end{array}\right] \qquad [8]$$

Measuring results

Results of the hydraulic efficiency are shown relative to the efficiency measured at BEP from a measurement done in 1979, and the mean values of the hydraulic efficiency are based on equations 7 and 8. The frame (Figure 4) was traversed in five vertical positions, which gave 30 measured points at the outlet cross section. Based on the temperature and velocity at each measured point, the hydraulic efficiency was calculated. In this paper, measurements at three different turbine loads will be presented. The turbine loads are at 68%, 83% and at 95%, and the position on the efficiency curve are shown in Figure 5. For each of the turbine loads, velocity distribution, temperature distribution, and efficiency deviation will be presented.

The BEP occurs at a smaler turbine load compared to the measurement performed in 1979. This can be caused by an increased friction loss as mentioned on page 2.

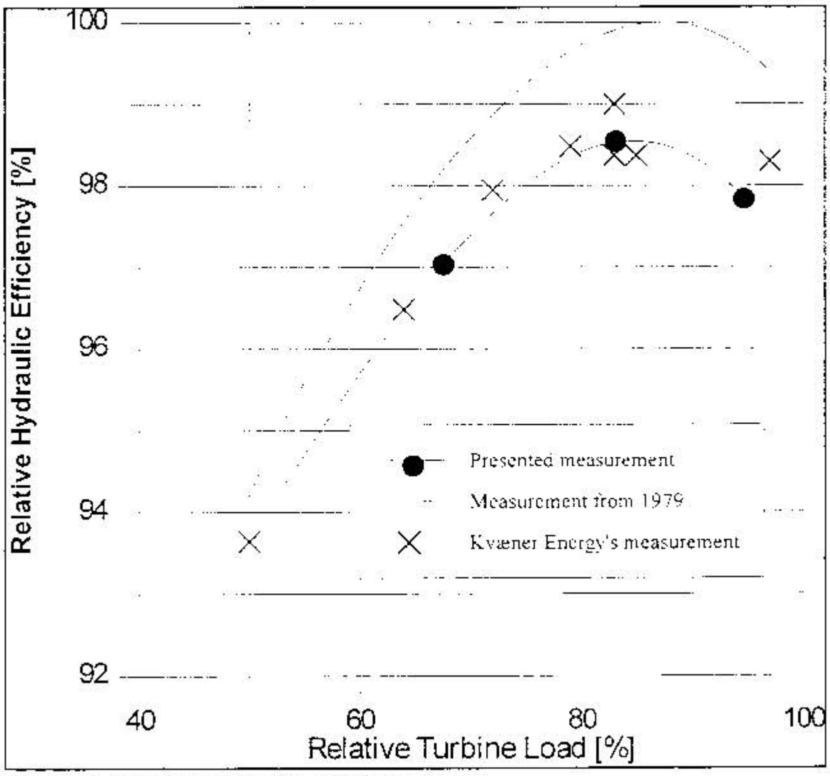

Figure 5. Relative hydraulic efficiency curve

The deviation from the mean hydraulic efficiency are shown as a difference between the calculated hydraulic efficiency using Equation 7 and the calculated hydraulic efficiency from each measured point in Figure 8, 11 and 14. Kværner Energy performed a hydraulic efficiency measurement in December 1995, which are shown in Figure 5. The measured points at turbine load of 83% is comparable with one of the presented data in this paper. Kværner Energy used a collector to gather water from different locations in a horizontal plane. The collector was mounted on a frame that was traversed in different vertical locations at the outlet cross section. Their calculated hydraulic efficiency at the turbine load of 83% shows the same trend as the presented data in Figure 11, and their measured differential temperature shows the same trend as in Figure 10.

Measurements at the outlet cross section for turbine load at 68% are presented in Figure 6, 7 and 8. The velocity distribution is shown in Figure 6, and it shows a higher velocity at the right side towards the bottom. The unsymmetric distribution was probably caused by the swirl flow initiated by the turbine running below from the best efficiency point and the draft tube bend. The temperature distribution in Figure 7 shows the difference in temperature between the inlet and the outlet measured temperature. The highest difference temperature is found at the bottom of the draft tube, and the lowest difference temperature at the top. This gives a higher deviation from the mean hydraulic efficiency at the bottom of the draft tube. The maximum deviation from mean hydraulic efficiency was 0,6 %, and was found at the left hand side.

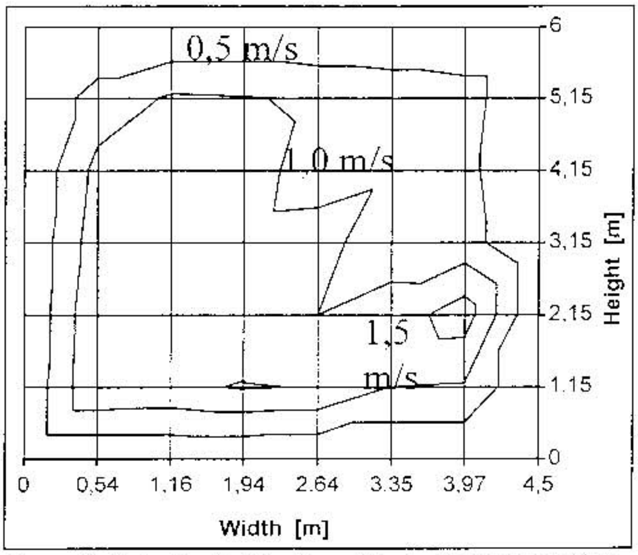

Figure 6. Velocity distribution at the outlet cross section for P=68%

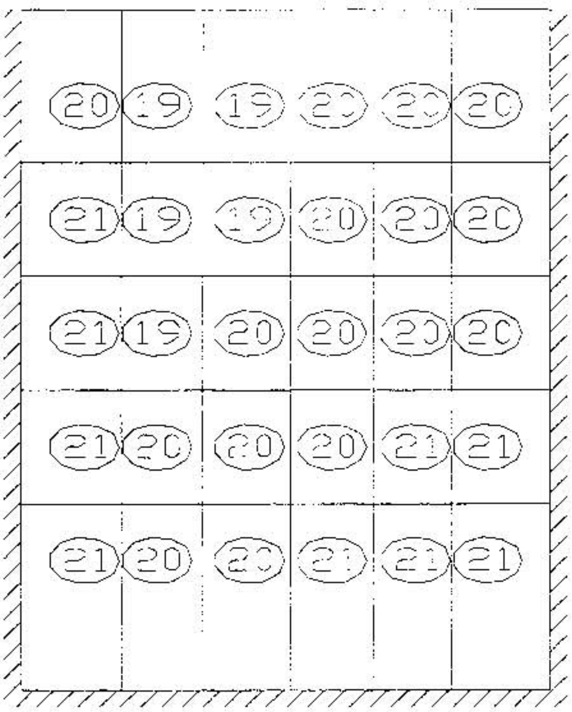

Figure 7. Measured difference temperature (mK) distribution at the outlet cross section for P=68%

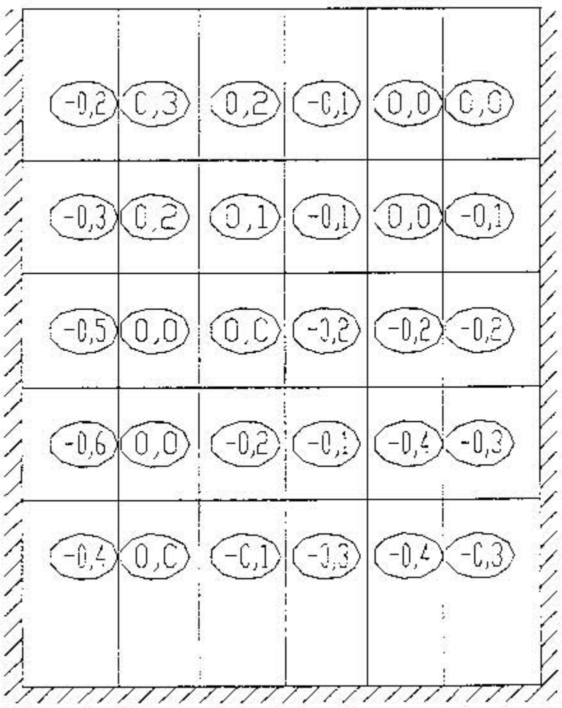

Figure 8. Calculated deviation from mean hydraulic efficiency (%) at the outlet cross section for P=68%

Measurements at the outlet cross section for turbine load at 83% are presented in Figure 9, 10 and 11. This load is close to the best efficiency point, and no or low swirl it is expected in the draft tube. The temperature distribution, Figure 10 shows the highest difference temperature at the top of the draft tube, and the lowest difference temperature at the bottom. For the measurement carried out by Kværner Energy this trend repeated itself. The maximum deviation from mean hydraulic efficiency was 0.5 %.

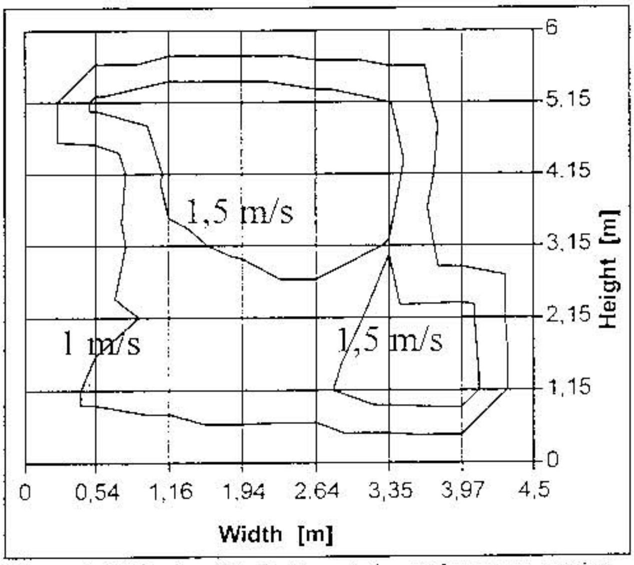

Figure 9. Velocity distribution at the outlet cross-section for P=83%

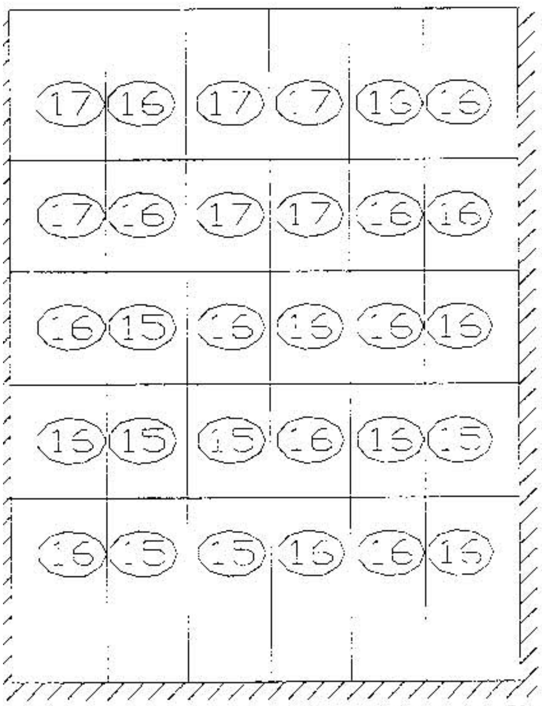

Figure 10. Measured difference temperature (mK) distribution at the outlet cross-section for P=83%

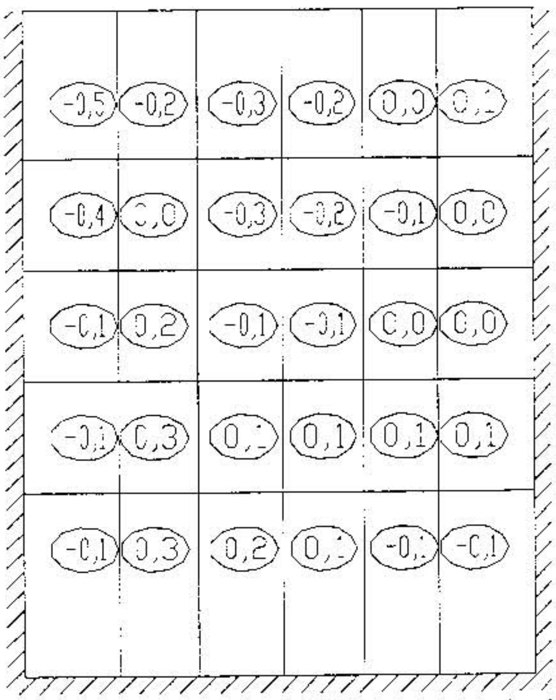

Figure 11 Calculated deviation from mean hydraulic efficiency (%) at the outlet cross section for P=83%

Measurements at the outlet cross section for turbine load at 95% are presented in Figure 12, 13 and 14. This is overload for the turbine, and a swirl is expected in the draft tube. The velocity distribution is shown in Figure 12, and it shows a higher velocity to the right side.. The temperature distribution in Figure 13 shows the difference in temperature between the inlet and the outlet measured temperature. It shows that the highest difference temperature is at the bottom of the draft tube, and that the lowest difference temperature at the top. The maximum deviation from the mean hydraulic efficiency was 0.5 %, and it was found at the left side of the outlet cross section.

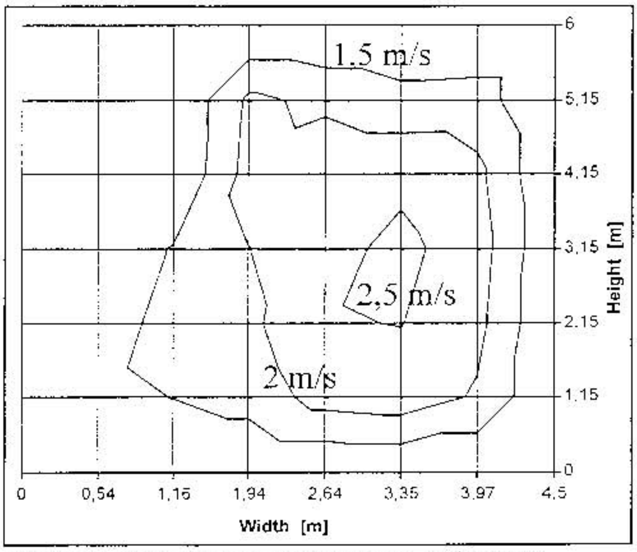

Figure 12. Velocity distribution at the outlet cross-section for P=95%

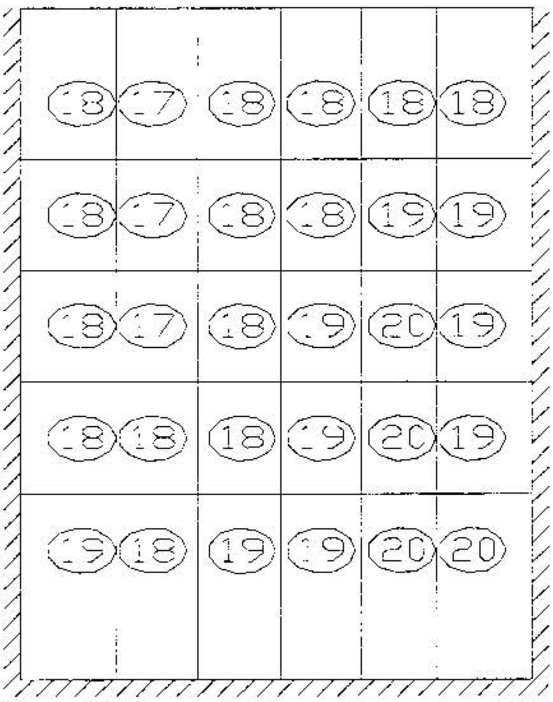

Figure 13. Measured difference temperature (mK) distribution at the outlet cross-section for P=95%

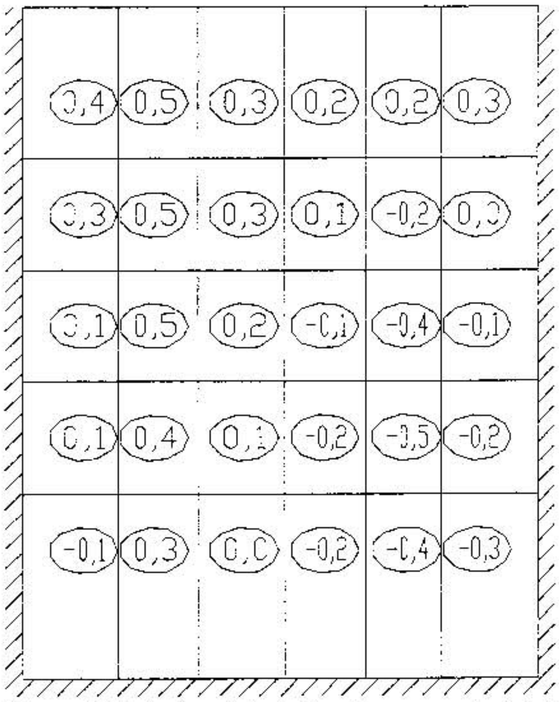

Figure 14 Calculated deviation from mean hydraulic efficiency (%) at the outlet cross section for P=95%

Conclusion

The measurements of the velocity and the temperature distribution over the outlet area of the draft tube show how important a mapping of these quantities are to the calculation of the hydraulic efficiency. The calculation of the hydraulic efficiency on the basis of average velocities and temperatures versus area weighted velocities and temperatures showed no significant difference in this case. The calculation of the hydraulic efficiency on the basis of one measuring point at outlet gave a maximum deviation from the mean hydraulic efficiency of 0,6 %, which confirms the uncertainly specified in IEC 41 [1]. The comparison between the presented data, and Kværner Energy's measurement shows that the collector method used at the outlet cross section measures a temperature well within their specified uncertainties. However, when using the collector method, it is important to map the temperature and velocity in several locations at the outlet cross section.

Acknowledgments

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Nomenclature

a	Isothermal factor	[m^3/kg]	p_2	Outlet pressure	[N/m^2]
A_j	Partial area	[m^2]	Q_1	Flow at the inlet	[m^3/s]
A_2	Outlet area	[m^2]	Q_{sp}	Flow in the labyrinth sealing	[m^3/s]
C_p	Specific heat	[J/kg°C]	ρ	Specific density	[kg/m^3]
δE_μ	Correction term	[J/kg]	T_1	Inlet temperature	[°C]
E_h	Hydraulic energy	[J/kg]	T_2	Outlet temperature	[°C]
E_μ	Mechanical energy	[J/kg]	T_{sp}	Temperature in labyrinth sealing	[°C]
η_h	Hydraulic efficiency	[-]	V_1	Inlet velocity	[m/s]
g	Acceleration of gravity	[m/s^2]	V_2	Outlet velocity	[m/s]
h_{uv}	See Figure 3	[m]	V_{T1}	Velocity at inlet temperature probe	[m/s]
H_n	Measured head of the turbine	[m]	V_{T2}	Velocity at outlet temperature probe	[m/s]
$*H_n$	Recalculated head	[m]	Z_{Duv}	See Figure 3	[m]
v	Specific volume	[m^3/kg]	Z_{tr}	See Figure 3	[m]
p_{atm}	Atmospheric pressure	[N/m^2]	Z_{ts}	See Figure 3	[m]
p_{sp}	Pressure in the labyrinth sealing tube	[N/m^2]	Z_{T1}	Geodetic height for the inlet thermometer	[m]
p_{tr}	Inlet static pressure (used to find E_m)	[N/m^2]	Z_{T2}	Geodetic height for the outlet thermometer	[m]
p_{T1}	Inlet pressure at temperature probe	[N/m^2]	Z_1	Geodetic height for the inlet measurements	[m]
p_{T2}	Outlet pressure at temperature probe	[N/m^2]	Z_2	Geodetic height for the outlet measurements	[m]
p_1	Inlet pressure	[N/m^2]	Z_{sp}	Geodetic height for the labyrinth sealing measurements	[m]