CORRECTIVE TERMS OF THERMODYNAMIC EFFICIENCY MEASUREMENTS

André Abgottspon
Etaeval GmbH, Altsagenstrasse 3, 6048 Horw, Switzerland
Email: andre.abgottspon@etaeval.ch

Marc Briggeler
Hochschule Luzern, Technikumstrasse 21, 6048 Horw, Switzerland
Email: marc.briggeler@hslu.ch

Thomas Staubli
Hochschule Luzern, Technikumstrasse 21, 6048 Horw, Switzerland
Email: thomas.staubli@hslu.ch

ABSTRACT

Corrective terms to improve the accuracy of the thermodynamic efficiency measurements are proposed in section 14.6 of the IEC standard 60041 – 1991. The following contribution focuses on the corrective term for the specific mechanical energy. The physical background of this correction will be discussed, the equations will be derived and the range of validity of empirical numbers will be given. On behalf of examples it will be demonstrates how the corrective terms can be determined. The used additional instrumentation such as an anemometer and a hygrometer will be specified.

According to the IEC standard, the arithmetical sum of the corrective terms should not exceed 2 percent of the specific mechanical energy, otherwise the measurement shall not be considered valid. The largest correction evaluated in the example of the measurements recently performed in the hydro power plant of Oberems, Valais, Switzerland, was due to direct exchange with the ambient air and amounted to 13.2J/kg. This corrections lead to an increase of 0.16 percent of the evaluated hydraulic efficiency.

1. CORRECTIVE TERM FOR SPECIFIC MECHANICAL ENERGY

The specific mechanical energy is, referring to IEC standard 60041 section 14.2 [1], generally defined as:

$$E_m = \bar{a} \cdot \left(p_{abs11} - p_{abs21}\right) + \bar{c}_p \cdot \left(\vartheta_{11} - \vartheta_{21}\right) + \frac{v_{11}^{\;2} - v_{21}^{\;2}}{2} + g \cdot \left(z_{11} - z_{21}\right) + \delta E_m$$

The last variable δE_m is the sum of several corrective terms. The following corrective terms, referring to IEC standard 60041 section 14.6 [1], should be taken into account in cases of imperfect measurement conditions, secondary phenomena etc:

- Variations of temperature (Section 2)
- Extraneous heat exchange
    - Heat exchange through the walls (Section 3.1)
    - Direct exchange with the ambient air (Section 3.2)

The sum of the corrective term δE_m is always added to the specific mechanical energy independent whether the machine operates in turbine or pump mode, but the individual corrective terms have different signs in the equations for a turbine or a pump. This has to be carefully taken into account especially in the case of reversible pump-turbines.

The following table gives an overview on the corrective terms; the important variables will be explained subsequently:

Tab. 1: Formulas for the corrective terms in the case of a turbine and of a pump

	Turbine $\eta_h = \frac{E_m}{E}$	**Pump** $\eta_h = \frac{E}{E_m}$
Variations of temperature	$\delta E_m = \overline{c}_p \cdot \frac{\Delta \vartheta}{\Delta t} \cdot (t_a - t - t_b)$ $\rightarrow \eta_h$ *increases for negative* $\frac{\Delta \vartheta}{\Delta t}$ $\rightarrow \eta_h$ *decreases for positive* $\frac{\Delta \vartheta}{\Delta t}$	$\delta E_m = \overline{c}_p \cdot \frac{\Delta \vartheta}{\Delta t} \cdot (t_a + t - t_b)$ $\rightarrow \eta_h$ *increases for negative* $\frac{\Delta \vartheta}{\Delta t}$ $\rightarrow \eta_h$ *decreases for positive* $\frac{\Delta \vartheta}{\Delta t}$
Heat exchange through the walls	$\delta E_m = +\frac{1}{(\rho \cdot Q)_1} \cdot A \cdot P_{a-w} \cdot (\vartheta_a - \vartheta_w)$ $\rightarrow \eta_h$ *increases for* $\vartheta_a > \vartheta_w$ $\rightarrow \eta_h$ *decreases for* $\vartheta_a < \vartheta_w$	$\delta E_m = -\frac{1}{(\rho \cdot Q)_1} \cdot A \cdot P_{a-w} \cdot (\vartheta_a - \vartheta_w)$ $\rightarrow \eta_h$ *increases for* $\vartheta_a > \vartheta_w$ $\rightarrow \eta_h$ *decreases for* $\vartheta_a < \vartheta_w$
Direct exchange with ambient air (Pelton turbine)	$\delta E_m = +\frac{\rho_a \cdot Q_a}{(\rho \cdot Q)_1} \cdot [c_{pa} \cdot (\vartheta_H - \vartheta_{20})]$ $\rightarrow \eta_h$ *increases for* $\vartheta_H > \vartheta_{20}$ $\rightarrow \eta_h$ *decreases for* $\vartheta_H < \vartheta_{20}$	-

Variations of temperature:

$\frac{\Delta \vartheta}{\Delta t}$: temperature gradient

t_a : is the time, taken by the water to pass from the high pressure tapping point to the corresponding measuring vessel

t : is the time, taken by the water to pass through the machine between the two measuring sections

t_b : is the time, taken by the water to pass from the low pressure tapping point to the corresponding measuring vessel

Heat exchange through the walls:

ϑ_a : is the ambient air temperature

ϑ_w : is the temperature of water in the turbine or the pump

Direct exchange with ambient air:

ϑ_H : is the temperature of aspirated air

ϑ_{20} : is the temperature of water in the measuring section 20

2. VARIATION OF TEMPERATURE

The determination of the efficiency in hydraulic machines by the thermodynamic method is normally applied in case of steady state flow. Usually this condition focuses on the pressure and the flow rate of water. For thermodynamic efficiency the steadiness of the temperature of the water at the inlet of a pump or a turbine is most important. Such fluctuation in temperature affects the accuracy of the measurements, since there is time delay of the water flowing through the machine between inlet and outlet. If the temperature changes at the inlet the change will be measured with same delay at the outlet.

IEC standard 60041 [1] allows a slow and continuous variation of temperature of less than 0.005K per minute during one run. To increase the accuracy of a measurement the specific mechanical energy E_m should be corrected with the formula for the correction due to temperature variation as given in the previous section, even when the temperature gradient $\frac{\Delta\vartheta}{\Delta t}$ is smaller than the allowed value.

This correction is valid for a constant temperature gradient. That means that tche formula should be applied only when the temperature is a linear function of time (see Fig. 1). The derivation of the formula given in the IEC standard 60041 for the correction due to temperature variation is described by Borel [2].

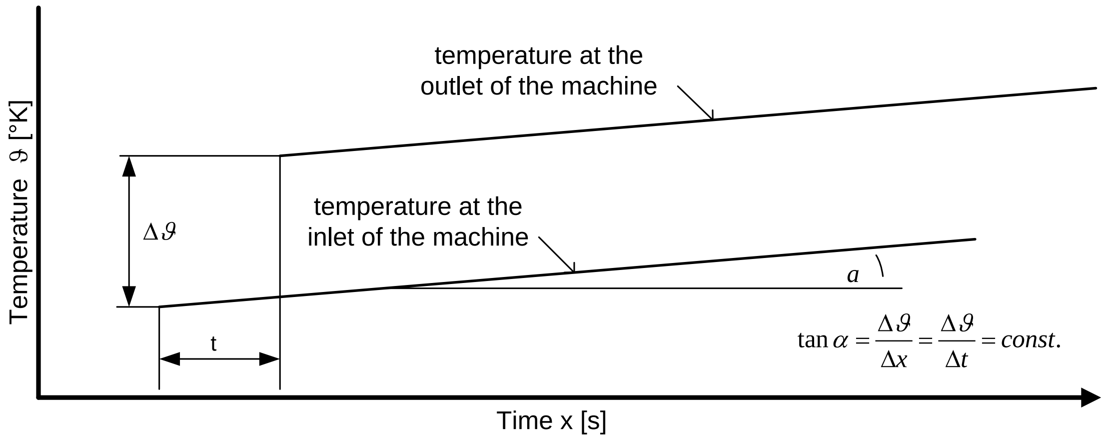


Fig. 1: Linear change of temperature - constant temperature gradient

If the temperature curve is not a linear function (see Fig. 2), the correction should be calculated according to Borel [2] with the following formula:

In case of a turbine: $$\delta E_m = \overline{c}_p \cdot \left[f\left(x - t - t_b\right) - f\left(x - t_a\right)\right]$$

In case of a pump: $$\delta E_m = \overline{c}_p \cdot \left[f(x - t_b) - f(x - t - t_a)\right]$$

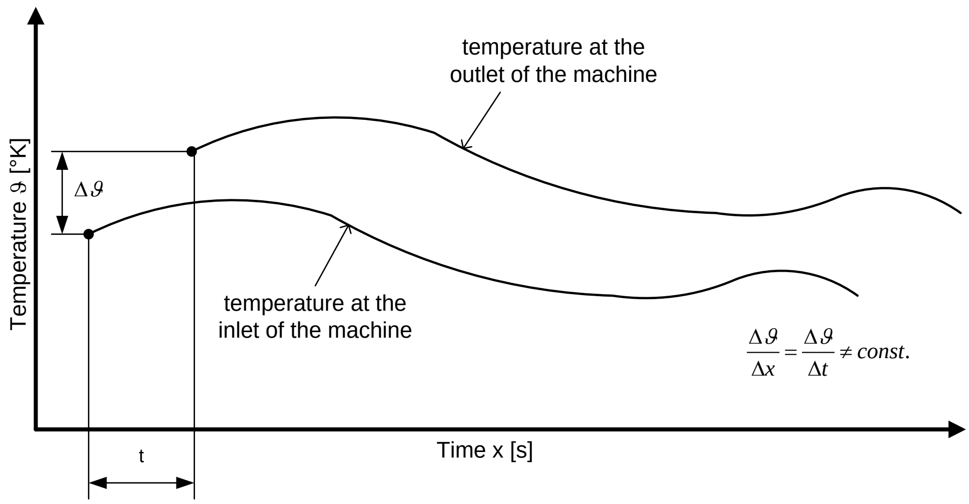


Fig. 2: Non-linear variation of temperature

In practise it is difficult to determine the function of the temperature in case of a non-linear temperature gradient. Polynomial approximation might be considered. For non-linear temperature variations the chances are high that the instantaneous temperature gradient exceeds the allowed value (0.005K/min), while the average gradient remains far below the limit. It depends much on the chosen measuring time if this point can be used or not. IEC standard 60041 [1] does not give recommendations on how to treat instantaneous gradients.

To avoid to strong temperature variation, it is advisable to take measurements during periods when the conduit or the intake (high pressure side of turbine, low pressure side of pump if existing) is not exposed to strong sunlight, e.g. during the night. Moreover, it is also recommended that secondary inflows into the penstock or into the surge tank should be avoided. If no perfect mixing can be achieved the chances for temperature variations will be high.

The corrective term due to temperature variation gets small when the gradient is small, but also for short time terms. That is for small volumes of water and high velocities within the machine.
zero correction for a turbine for $t = t_a - t_b$ and for a pump for $t = t_b - t_a$

2.1 Influence of the temperature gradient on the corrective term

The correction due to temperature variation can be positive or negative, depending on the prefix from the temperature gradient. The value in the brackets (see below) is negative for a turbine and positive for a pump. The time t (taken by the water to pass through the machine between the two measuring sections) is normally longer than the difference between t_a and t_b.

The outcome of this is a negative value for a turbine and a positive value for a pump:

Turbine: $(t_a - t - t_b) < 0$ Pump: $(t_a + t - t_b) > 0$

This leads to following value for the corrective term due to temperature variation for the different temperature gradients:

Turbine: $\frac{\Delta\vartheta}{\Delta t} > 0 \rightarrow \delta E_m < 0$ $\frac{\Delta\vartheta}{\Delta t} < 0 \rightarrow \delta E_m > 0$

Pump: $\frac{\Delta\vartheta}{\Delta t} > 0 \rightarrow \delta E_m > 0$ $\frac{\Delta\vartheta}{\Delta t} < 0 \rightarrow \delta E_m < 0$

2.1.1 Example for a negative temperature gradient in a turbine

A large temperature variation was observed during efficiency measurements in the hydro power plant in Solis, Graubünden, Switzerland. The reason was a secondary inflow from a waste water treatment plant, which was fed intermittently into the penstock. This led to extraordinary temperature variation and gave good examples for positive and negative temperature gradients to show its effect on the determination of the specific mechanical energy.

In Fig. 3 an example from a measurement point with a negative temperature gradient is given. The gradient at this point is -0.0077K/min and is higher than the allowed value referring to IEC standard 60041. The expression in the bracket is negative. The time t_a (is the time taken by the water to pass from the high pressure tapping point to the corresponding measuring vessel) is in this case 0.2s and is much smaller then the time t, which was in this case 16.43s (calculated with the volume between measuring section one and two and the flow rate). The time t_b (is the time taken by the water to pass from the low pressure tapping point to the corresponding measuring vessel) is zero. For Pelton turbines, the low pressure measuring section is in tail water. Therefore, the water temperature is taken directly at the measuring section.

It is shown in Fig. 3 that the measured temperature difference is higher than the temperature difference due to energy dissipation in the machine. This leads to a smaller value of the specific mechanical energy and therefore to a smaller value of the hydraulic efficiency without a correction. A negative temperature gradient and a negative value in the bracket lead to a positive value of the correction due to the temperature variation. In this measurement point, the correction amounted to 0.154% of the specific mechanical energy. Although the gradient exceeds the maximum value with 0.0027K/min (|-0.0077K/min|-0.005K/min=0.0027K/min) , the percentage of this correction term to the specific mechanical energy is much smaller than the allowed 2% of the sum of all correction terms in proportion to the specific mechanical energy.

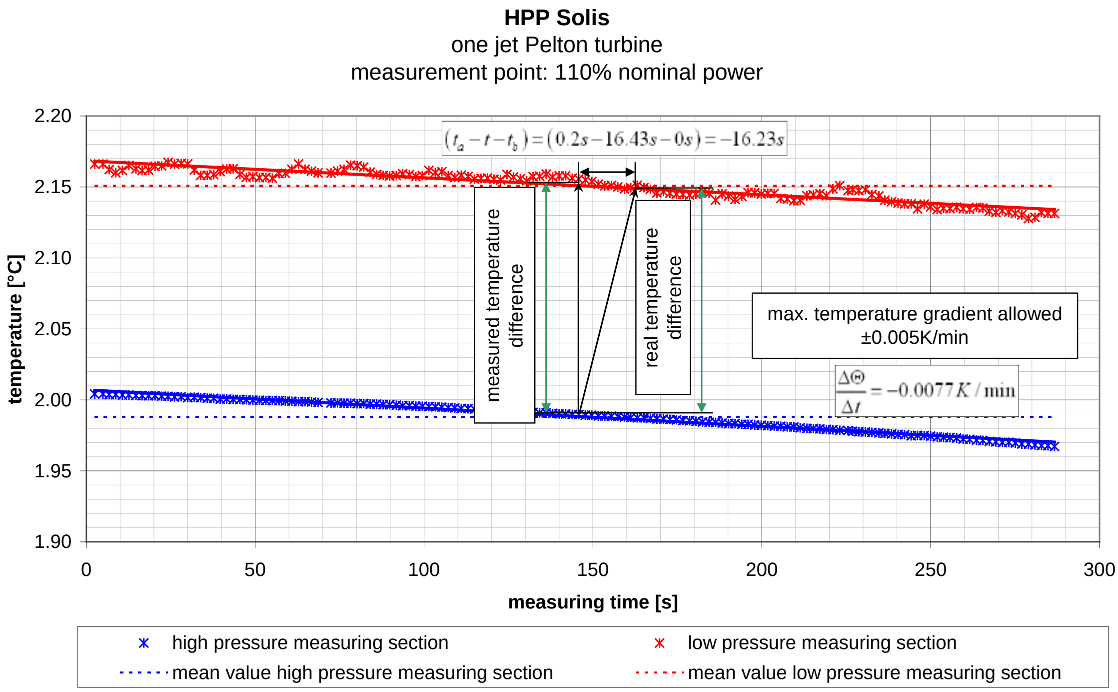


Fig. 3: Negative temperature gradient and its effect on temperature difference between high and low pressure measuring section

2.1.2 Example from a positive temperature gradient in a turbine

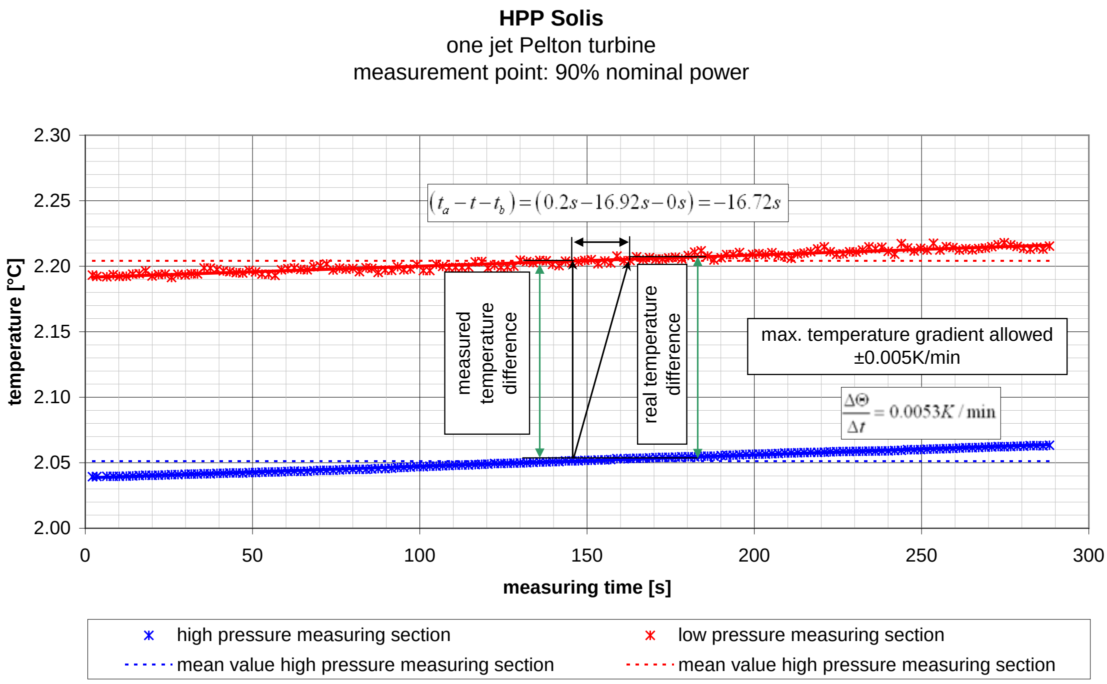


Fig. 4: Positive temperature gradient and its effect on temperature difference between high and low pressure measuring section

In Fig. 4 an example of a measuring point with a positive temperature gradient in the same power plant is given. The gradient is with 0.0053K/min just at the limit of the allowed value. The value in the bracket (time difference) is in case of a turbine again negative. It is shown, that the measured temperature difference is smaller than the temperature difference due to dissipation. Therefore the specific mechanical energy and also the hydraulic efficiency would be too high without a correction. A positive temperature gradient and a negative value in the bracket lead to a negative correction due to the temperature variation. The correction goes also for a positive gradient to the right direction.

The correction due to temperature variation amounted to 0.108% of the specific mechanical energy at this measurement point. This is again far away from the maximum allowed value from 2% (sum of all corrections terms in percentage of the specific mechanical energy).

2.2 Time difference

2.2.1 Calculation of the time difference

The normal way to calculate the times in the bracket (t_a, t, t_b) from the formula of the corrective term due to temperature variation is with the volume and the flow rate.

This comes from the calculation of the mean hydrodynamic residence time defined as:

$t = \frac{m}{\dot{m}}$ with m = mass and $\dot{m}$ = mass flow and with constant density $t = \frac{m}{\dot{m}} = \frac{\rho \cdot V}{\rho \cdot \dot{V}} = \frac{V}{\dot{V}}$

Generally some mixing occurs within the machine. This is shown in Fig. 5 with an example of an indicator inserted into the water flow in the section 1. The same mixing occurs with the water temperatures between two measuring sections.

The mean residence time can be calculated either with the formula above, or be determined by a measurement with an indicator.

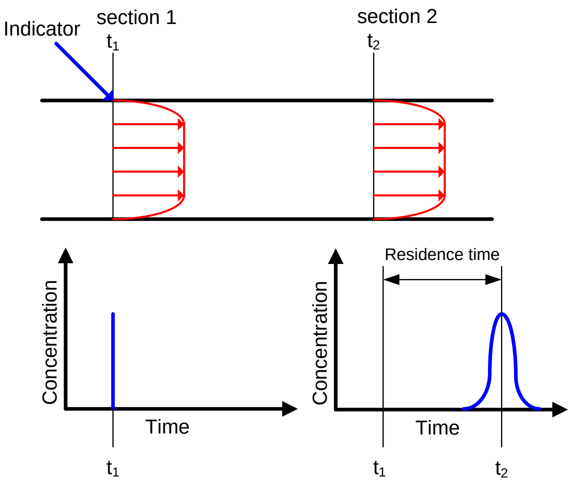


Fig. 5: Residence time in a pipe flow

2.2.2 Comparison time difference calculated and read out from a temperature curve

Two examples with non-linear temperature variations of measuring points during field tests in a hydro power plant in Solis are given in Fig. 6 and 7. These examples can be used to determine the residence time.

In Fig. 6 there is at the beginning a negative temperature gradient till it is zero. Afterwards the temperature gradient changes to a positive value. The residence time determined from the time delay of the measurements in this example is 20.16s. It is about 17% larger than the calculated time difference from 17.28s.

In the second example shown in Fig. 7 the time difference determined with the shift of the curves is 22.75s. This value is about 29% larger then the value from the calculated residence time. To determine the time difference more exactly from the measurements, longer periods of data should have be acquired.

From the analysis of these two examples it can be concluded that the uncertainty of the correction term due to temperature gradients easily can be of the order of 20%.

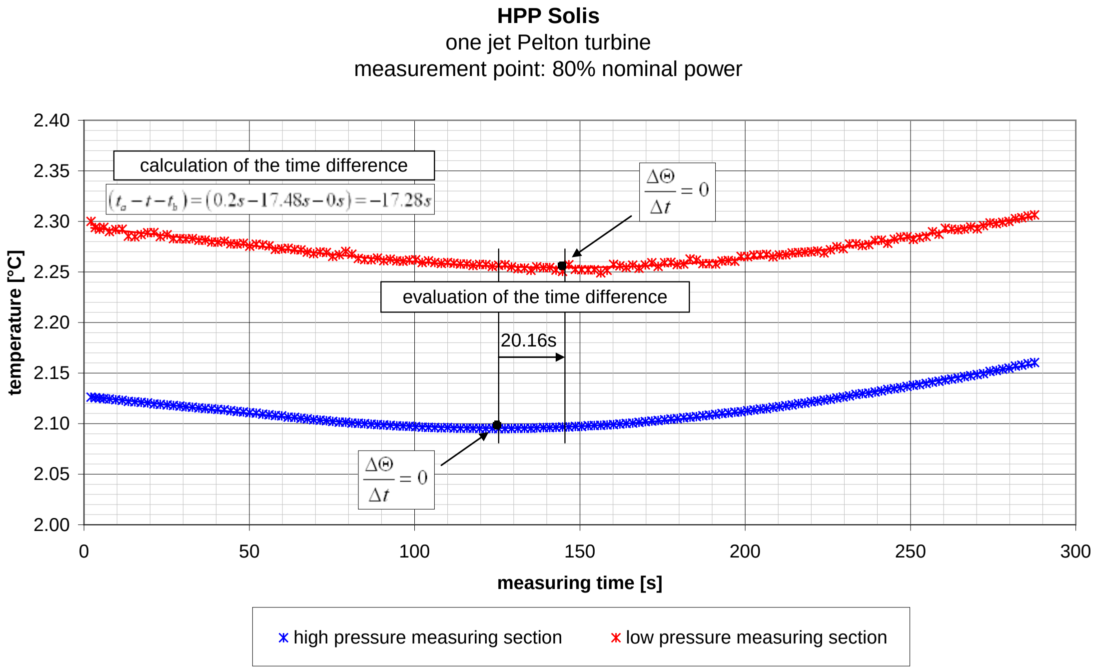


Fig. 6: Example 1: Non Linear Temperature Variation

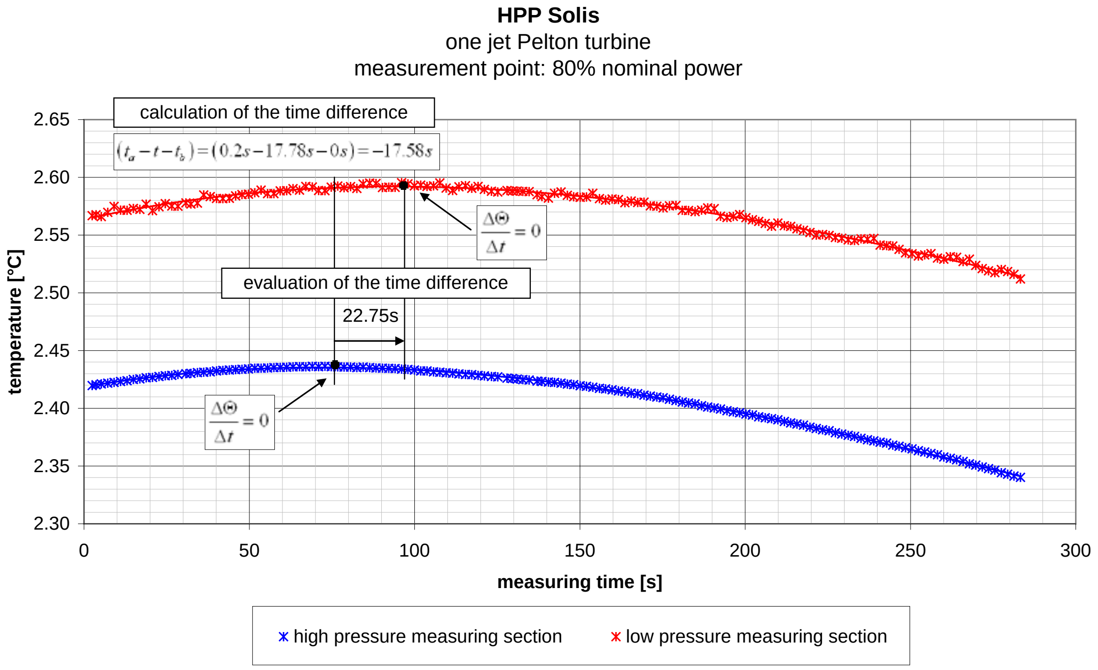


Fig. 7: Example 2: non linear temperature variation

3. EXTRANEOUS HEAT EXCHANGE

3.1 Heat exchange through the walls

Heat exchange through wall is basically proportional to the surface area, assuming all other parameters being constant, thus proportional to the square of the length scale. Power of a machine increases with the fifths power of the length scale, assuming constant rotational speed. Accordingly, the influence of the heat exchange will drastically decrease with increasing length scale. For Pelton turbines, however, where the circumferential speed is given by the head of the machine, the power increases also with the square of the length scale thus the relative influence of the heat transfer will be the same for small or large Pelton turbines.

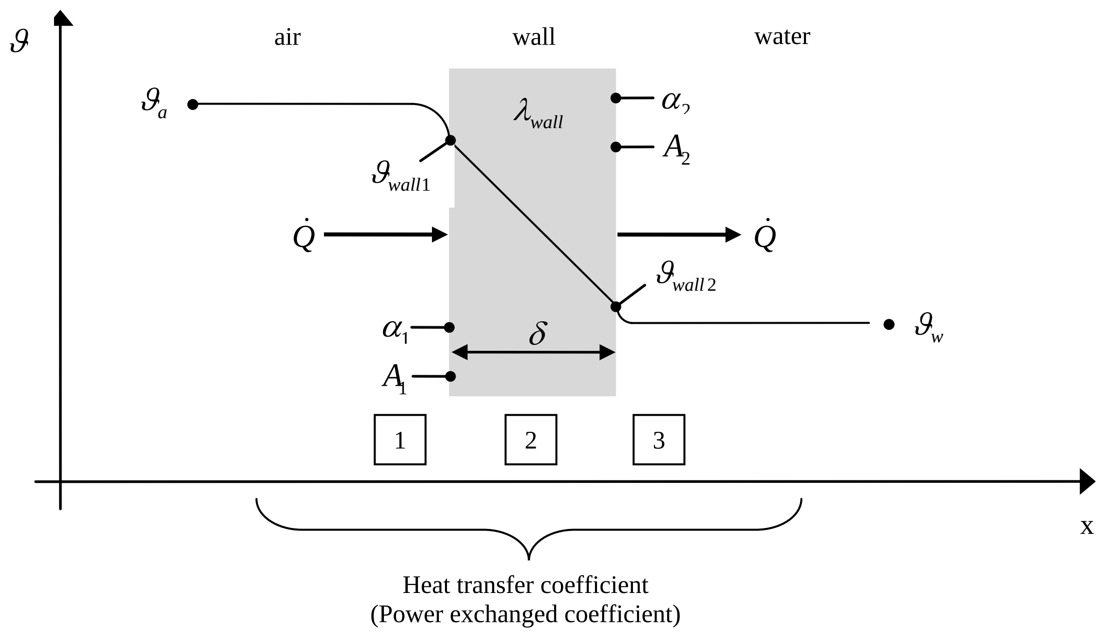
3.1.1 No condensation on the machine surface

The formula to be used is given in Tab. 1. It is a semi-empirical expression to keep the procedure for estimation of the heat transfer simple.

The equation takes into account that the specific total heat transfer over an interface is equal to the area times the transfer coefficient and times the potential change.

The basic idea behind is that the potential change (temperature difference) acts as driving factor for the interfacial heat transfer. In IEC standard 60041 section 14.6.2.1 [1] the heat transfer coefficient is called power exchanged coefficient and is considered to be equal to $10 W/m^2K$. The following consideration shows that this value is realistic for a typical application in a hydro power plant.

A steady heat transfer through a wall is a combination of a convective heat transfer from the fluid to the wall (1), followed by the conductive heat transfer through the wall (2) and a convective heat transfer from the wall to the fluid (3):



(1) With α, the convective heat transfer coefficient, the heat flux from the air to the wall is [3]:

$$\dot{Q} = \alpha_1 \cdot A_1 \cdot \left(\vartheta_a - \vartheta_{wall1}\right)$$

(2) With λ, the conductive heat transfer coefficient, the heat flux through the wall is [3]:

$$\dot{Q} = \frac{\lambda_{wall}}{\delta} \cdot A \cdot \left(\vartheta_{wall1} - \vartheta_{wall2}\right)$$

(3) The convective heat flux from the wall to the water is [3]:

$$\dot{Q} = \alpha_2 \cdot A_2 \cdot \left(\vartheta_{wall\,2} - \vartheta_w\right)$$

With the three formulas above, the wall temperatures can be eliminated and the equation can be rewritten as:

$$\dot{Q} = P_{a-w} \cdot A \cdot \left(\vartheta_a - \vartheta_w\right)$$

With the overall heat transfer coefficient [4]: $P_{a-w} = \dfrac{1}{\dfrac{1}{\alpha_1} + \dfrac{\delta}{\lambda_{wall}} + \dfrac{1}{\alpha_2}}$

For more than one wall layer or coat of painting the heat transfer coefficient is [4]:

$$P_{a-w} = \frac{1}{\dfrac{1}{\alpha_1} + \sum_i \dfrac{\delta_i}{\lambda_i} + \dfrac{1}{\alpha_2}}$$

The convective heat transfer coefficient α can only be solved analytically for laminar problems. For any other case the convective heat transfer coefficient must be determined experimentally. However, for a great number of problem situations coefficients can be found in literature. To determine the order of magnitude of the heat transfer coefficient, the values given in a physics handbook [5] are taken:

- air with very low velocity < 0.5m/s (free convection) α_1 : 10W/m^{2}K
- water with a velocity of 1m/s (free convection) α_2 : 2500W/m^{2}K (conservative)

The conductive heat transfer coefficient λ is also determined by experiments and available for most materials, we set [5]:

- cast iron (at 20°C) λ_{wall} : 40W/mK
- iron steel (at 20°C) λ_{wall} : 45W/mK
- concrete (at 20°C) λ_{wall} : 2.1W/mK
- coats of painting (epoxy resin) λ_{wall} : 0.2W/mK [7]

Therefore the heat transfer coefficient is for a turbine with a casing of cast iron with 40mm thickness and two coats of painting of 560µm thickness at the inner side and 230µm thickness at the outer side [thickness for coats of painting from 7]:

$$P_{a-w} = \frac{1}{\dfrac{1}{10\dfrac{W}{m^2 \cdot K}} + \dfrac{0.040m}{40\dfrac{W}{m \cdot K}} + \dfrac{0.00056m}{0.2\dfrac{W}{m \cdot K}} + \dfrac{0.00023m}{0.2\dfrac{W}{m \cdot K}} + \dfrac{1}{2500\dfrac{W}{m^2 \cdot K}}}$$

$$= \frac{1}{0.1\dfrac{m^2 \cdot K}{W} + 0.001\dfrac{m^2 \cdot K}{W} + 0.0028\dfrac{m^2 \cdot K}{W} + 0.0012\dfrac{m^2 \cdot K}{W} + 0.0004\dfrac{m^2 \cdot K}{W}} = 9.5\frac{W}{m^2 \cdot K}$$

The heat transfer coefficient is determined to be P_{a-w}= 9.5W/m^{2}K. Without the second (conductive heat transfer through the casing) and fifth term (convective heat transfer from the wall to the water) in the denominator of above equation only a minor difference of approximately 1 percent is found in the result. Obviously, the fraction of the conductive heat transfer through the casing and the convective heat transfer from the wall to the water are almost negligible. Neglecting also the conductive heat transfer through the coats of painting a heat transfer coefficient of 10W/m^{2}K is calculated, corresponding to the magnitude of the power exchanged coefficient proposed in the IEC standard 60041.

This example shows that with a value of 10W/m²K the corrective term due to heat exchange through the wall is eventually overestimated in the order of 5 percent if the casing is coated on the inner and outer side.

In the IEC standard 60041 is stated that the heat exchange through concrete and rocket walls is negligible. Assuming a draft tube which lies 2m under the ground floor in an iron steel tube of 10mm thickness and one coat of painting inside the tube, the following heat transfer coefficient would be found:

$$P_{a-w} = \frac{1}{\dfrac{1}{10\dfrac{W}{m^2 \cdot K}} + \dfrac{2m}{2.1\dfrac{W}{m \cdot K}} + \dfrac{0.010m}{45\dfrac{W}{m \cdot K}} + \dfrac{0.00056m}{0.2\dfrac{W}{m \cdot K}} + \dfrac{1}{2500\dfrac{W}{m^2 \cdot K}}}$$

$$= \frac{1}{0.1\dfrac{m^2 \cdot K}{W} + 0.952\dfrac{m^2 \cdot K}{W} + 0.0002\dfrac{m^2 \cdot K}{W} + 0.0028\dfrac{m^2 \cdot K}{W} + 0.0004\dfrac{m^2 \cdot K}{W}} = 0.95\frac{W}{m^2 \cdot K}$$

That is approximately 10 times smaller than without concrete wall. For this reason this heat transfer may be neglected in most of the cases since the concrete wall thicknesses usually are not smaller than 0.5m.

3.1.2 Condensation on the machine surface

If on a machine surface condensation is visible, an adjustment of the corrective term due to heat exchange through the walls is necessary. The effect of condensation can be included by increasing the “dry” heat exchange in a certain proportion related to the water and air temperatures and the humidity conditions of the air (maximal 400%).
Condensation will take place if the surface is sufficiently cool and the surrounding warm enough. The “dry” heat transfer flux $\dot{Q}$ will not be changed noticeably. The heat resistance through the condensate film is negligible. But the condensation means that the condensating steam gives its heat of evaporation to the wall [6].
The proportion factor is defined in the IEC standard 60041 Section 14.6.2.1 [1] as:

$$\psi = \frac{1}{1 - \dfrac{k \cdot x}{\Delta i}}$$

Δi : is the specific enthalpy difference of the air in the surroundings and the water, in J/kg
x : is the difference of relative water content of the air, in kg/kg
k : specific heat of vaporization of water, in J/kg

The specific heat of vaporization of water k is 2500kJ/kg, which is a definition to use the Mollier diagram. To get the rest of the variables, measurements (for example with a hygrometer) of the air temperature and relative air humidity are necessary:

Specification of the used hygrometer:

Temperature range:	-10 – 50°C
Permissible tolerance:	±0.5°C
Rel. humidity range:	0 – 100% rH
Permissible tolerance:	±2.5% rH

Fig. 8: Relative humidity measurement close to the heat exchanging wall

The evaluation of the difference of relative water content of the air x and the difference of specific enthalpy Δi can be shown in a Mollier diagram:

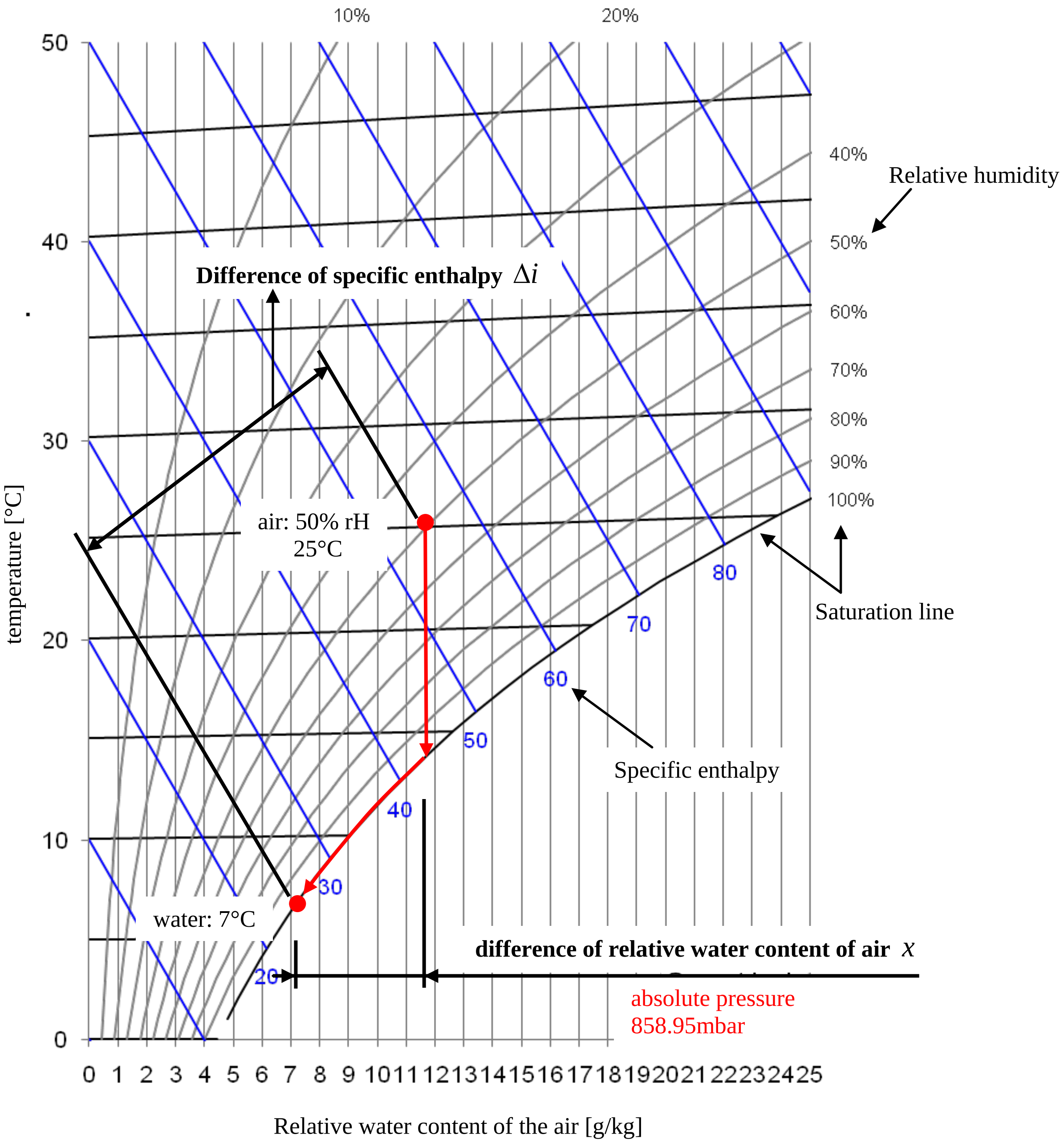


Fig. 9: Mollier diagram with the condensation process (red line)

In Fig. 9 the condensating process from air with a relative humidity of 50% and a temperature of 25°C to saturated air with a relative humidity of 100% and a temperature of 7°C is demonstrated. The temperature of the saturated air on the metal surface is assumed to be the same as the temperature of the water inside the housing (conductive heat transfer through the casing is negligible, see section 3.1.1).

From the Mollier diagram above the difference of relative water content of air and the difference of specific enthalpy can directly determined graphically. Therefore the proportion factor can be evaluated as:

$$\psi = \frac{1}{1 - \frac{k \cdot x}{\Delta i}}$$

$$\psi = \frac{1}{1 - \dfrac{2.5\cdot 10^{6}\,\dfrac{J}{kg}\cdot 4.5\cdot 10^{-3}\,\dfrac{kg}{kg}}{30\cdot 10^{3}\,\dfrac{J}{kg}}} = 1.6$$

The “dry” heat exchange has to be increased for this example by approximately +60%.

3.2 Direct exchange with the ambient air

The formula to be used is given in Tab. 1. It is a simplification of the original equation in the IEC standard 60041 [1]:

$$\delta E_m = +\frac{\rho_a \cdot Q_a}{(\rho \cdot Q)_1} \cdot \left[c_{pa} \cdot (\vartheta_H - \vartheta_{20}) + K_w \cdot (\alpha_a - \alpha_{20})\right]$$

The last term of the equation above is negligible because the ratios between steam and air masses at the injection point α_a and in section 20 α_{20} are almost the same. The equation in Tab. 1 describes the ratio between the mass flow rate of the injected air and the water through the Pelton turbine times the specific heat of air (approximately 1005J/kg·K) times the temperature difference (potential change) between the aspirated air and the water in a Pelton turbine.

To get the variables, measurements (for example with an anemometer with build in temperature probe) of the mass flow rate of air and their temperature are necessary:

Specification of the used anemometer:

Temperature range	: 0 – 50°C
Permissible tolerance	: ±0.5°C
Air velocity	: 0.3–20m/s
Permissible tolerance	: ±0.1m/s + 1.5%

Fig.10: Temperature and air velocity measurements at the inlet of the air aspirated by the Pelton Turbine

The density of air can be calculated with the ideal gas law (R = universal gas constant of air 287J/kg·K):

$$\rho_a = \frac{p_{amb}}{R \cdot (273.15K + \vartheta_H)}$$

The flow rate of air can be calculated with the measured mean velocity times the area (see Fig. 10):

$$Q_a = v_a \cdot A_a$$

4. MEASUREMENTS IN HYDRO POWER PLANTS

- Temperature gradient

The following table shows the results of the corrective terms only due to temperature variations in a proportion of the total specific mechanical energy in selected hydro power plants in Switzerland:

Tab. 2: Summary of the influence on the specific mechanical energy due to temperature variation

Hydro power plant	Time t_a	Time t	Time t_b	Temperature gradient	Temperature variation corrective term	Specific Mechanical Energy	Proportion
	t_a	t	t_b	$\frac{\Delta \vartheta}{\Delta t}$	δE_m	E_m	
	[s]	[s]	[s]	[K/min]	[J/kg]	[J/kg]	[%]
Oberems (Pelton)	0.2	31.6	0	0.000’000’7	-0.10	8435.6	0.001
Soazza (Pelton)	0.2	2.1	0	-0.000’086’9	0.70	6101.4	0.011
Frisal (Pelton)	0.2	8.7	0	-0.000’008’8	0.31	4167.0	0.008
Sedrun (Pelton)	0.2	5.2	0	-0.000’002’2	0.05	5120.6	0.001

The temperature gradient is almost zero in stable measuring condition. As a result the correction term due to temperature variations is smaller than 0.02% in proportion to the specific mechanical energy.

- Heat exchange through the walls

The following table shows the results of the corrective terms only due to heat exchange through the walls in a proportion of the total specific mechanical energy in selected power plants:

Tab. 3: Summary of the influence on the specific mechanical energy due to heat exchange through the walls

Hydro power plant	Heat exchange through the walls		Resulting heat exchange through the walls	Specific mechanical energy	Proportion
	No condensation	Proportion factor			
	δE_m	ψ	δE_m	E_m	
	[J/kg]	[-]	[J/kg]	[J/kg]	[%]
Oberems (Pelton)	3.9	1.2	4.8	8435.6	0.057
Soazza(Pelton)	1.7	3.0	5.0	6101.4	0.082
Frisal (Pelton)	1.6	-	1.6	4167.0	0.038
Sedrun (Pelton)	2.7	-	2.7	5120.6	0.052

If there is no condensation on the machine surface, the corrective term is small in comparison to the specific mechanical energy. To determine the proportion factor ψ, measurements with a hygrometer are essential. A visual estimation (as it was done in the case of the hydro power plant of Soazza) is very inaccurate and can lead to further errors or overestimations of the specific mechanical energy.

- Direct exchange with the ambient air

The following table shows the results of the corrective term due to the direct exchange with the ambient air in a proportion of the total specific mechanical energy in selected power plants:

Tab. 4: Summary of the influence on the specific mechanical energy due to direct exchange with the ambient air

Hydro power plant	Direct exchange with the ambient air	Specific mechanical energy	Proportion
	δE_m	E_m	
	[J/kg]	[J/kg]	[%]
Oberems (Pelton)	13.2	8435.6	0.156
Soazza (Pelton)	0.05	6101.4	0.001
Frisal (Pelton)	6.3	4167.0	0.152
Sedrun (Pelton)	0.9	5120.6	0.018

For the hydro power plant of Oberems a proportion of 0.16 percent of the specific mechanical energy is determined, this increases directly the result of the hydraulic efficiency by the same amount.

- Heat exchange through the shaft

Another heat source comes from the bearings close to the turbine casing. A heat flux through the shaft into the turbine casing occurs. An estimation of this heat flux was done in the hydro power plant of Sedrun, Switzerland. Shortly after the runner was stopped a temperature of 30°C was measured on the shaft close to the bearing. The distance from the measuring point to the turbine was approximately 1.3m and the shaft diameter was 600mm. With a average water temperature of 6°C the following heat flux results:

$$\dot{Q} = \frac{\lambda}{\delta} \cdot A \cdot \Delta\vartheta = \frac{45 \frac{W}{m \cdot K}}{1.3m} \cdot (0.6m)^2 \cdot \frac{\pi}{4} \cdot 24K = 235W$$

For this turbine with a power output of 25MW the fraction is approximately 0.001 percent. For another calculated turbine with a power output of 4.2MW and the same potential change in temperature the fraction is 0.001 percent too (the reason is explained in section 3.1). Therefore a correction is not necessary for Pelton turbines. In the IEC standard 60041 [1] such a heat flux through the shaft is not considered.

5. CONCLUSIONS

The following table shows the results of the sum of the corrective terms in relation to the total specific mechanical energy in the selected power plants:

Tab. 5: Summary of the influence on the specific mechanical energy

Hydro power plant	Sum of the corrective terms	Specific mechanical energy	Proportion
	δE_m	E_m	
	[J/kg]	[J/kg]	[%]
Oberems (Pelton)	17.9	8435.6	0.212
Soazza (Pelton)	5.7	6101.4	0.094
Frisal (Pelton)	8.2	4167.0	0.198
Sedrun (Pelton)	3.6	5120.6	0.071

The sum of the corrective terms led to a correction of the hydraulic efficiency by a maximum of 0.21 percent for the investigated hydro power plants. This is far away from the critical limit of 2 percent defined in the IEC standard 60041. But especially for Pelton turbines where the corrective term due to direct exchange with the ambient air is effective, detailed investigations for the corrective terms are recommendable. Compared to the uncertainty level which was calculated to be in the order of 0.8 percent for the selected power plants the sum of the corrective terms can reach values of the order of 1/4 of this uncertainty level.

A recommendation for the correction of the IEC standard 60041 is to rename the variable P_{a-w} to heat transfer coefficient instead of power exchanged coefficient.

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AUTHORS

André Abgottspon graduated in Mechanical Engineering from the Hochschule Luzern in Horw, followed by three years as research assistant at the CC Fluid Mechanics and Hydro Machines in the field of hydro power plant efficiency measurements and Pelton jet studies. In 2009 he co-founded the etaeval GmbH, a company specialized in hydro power plant measurements.

Marc Briggeler graduated in Mechanical Engineering from the Hochschule Luzern in Horw. Since 2008 he is research assistant at the CC Fluid Mechanics and Hydro Machines in the field of model turbine performance testing and hydro power plant efficiency measurements.

Thomas Staubli graduated in Mechanical Engineering from the Swiss Federal Institute of Technology (ETH) in Zürich. After two years of post-doctoral research in the field of flow induced vibration at Lehigh University, Pennsylvania, he worked in experimental fluid mechanics at Sulzer Hydro (now Andritz Hydro) in Zürich. He then headed the Hydromachinery Laboratory at the ETH Zürich. During this period he directed research projects in the field of hydraulic machinery. Since 1996 he is professor in Fluid Mechanics and Hydro Machines at the Hochschule Luzern.