COMPARISON BETWEEN THE PRESSURE-TIME AND THE THERMODYNAMIC METHOD ON A 52 M NET HEAD PLANT

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ABSTRACT

This paper presents a comparison of turbine efficiency measurements performed with the pressure-time method and the thermodynamic method at Gråsjø power plant. In addition, index performance tests were made using the Winter-Kennedy method. Gråsjø power plant is equipped with one vertical Francis turbine, and has a net head of 52 m. In 2012, the power plant was refurbished, and the turbine runner was replaced. The paper also presents the efficiency test performed on the previously installed runner before refurbishment. The efficiency test of the old runner was performed with the pressure-time method. The test conditions were good during all tests. The results from the thermodynamic test were used to verify the contractually guaranteed efficiency. The pressure-time and thermodynamic results agree well. The difference between the efficiency curves are below 0.5 percentage point (pp) for the whole measured range and below 0.15 pp for relative powers (P/P*) between 0.5 – 1.15.

INTRODUCTION

The pressure-time and thermodynamic methods are both methods for measurement of absolute turbine efficiency in the test standard IEC 60041:1991-11 [1]. However, the standard limits the application of the thermodynamic method to heads in excess of 100m, except for tests under highly favourable conditions.

Nevertheless, at Gråsjø power plant, which has a net head of 52 m, the turbine refurbishment contract specified the use of the thermodynamic method for comparison with turbine efficiency guarantees after installation of a replacement runner. To verify the results obtained using the thermodynamic method, it was decided to measure the efficiency using the pressure-time method in addition to the thermodynamic method. The efficiency was measured before runner replacement using the pressure-time method only.

Similar comparative tests of these two methods have been presented previously. Lévesque and Néron [2] presented a comparison of several methods, including the pressure-time and thermodynamic methods. However, the results of the thermodynamic tests were not presented in detail, but it was found that the thermodynamic method had a systematic bias of 1.5%. Dahlhaug et al [3] found results with similar efficiency curve shapes, with a maximum difference of 1.5 percentage points (pp) for one turbine, and 2.5 pp for another turbine, between the two methods. The best results of Dahlhaug had a systematic bias between the methods for most of the operating range, the efficiency measured using the pressure-time method being about 1 pp higher than the efficiency measured using the thermodynamic method. However, on the other turbine, the systematic bias was reversed - the efficiency obtained using the thermodynamic method was higher.

The differences in the above mentioned papers are significant, which merits the presentation of further comparative tests of these methods.

The uncertainty analyses of the tests is not object of discussion, and are not presented in this paper.

1 DESCRIPTION OF GRÅSJØ POWER PLANT

One Francis turbine is installed in Gråsjø power plant, which is owned by Statkraft AS. The turbine has a nominal net head of 52 m, a nominal power of 14 MW and a specific speed defined by Kjølle [4], $\boldsymbol{\Omega*}$ (Eq. 1), of 0.9. The water way consists of a trash rack, a tunnel of about 250 length followed by a horizontal penstock of about 120 m leading to the turbine. The turbine, which has a vertical axis, has a butterfly main inlet valve, a full spiral case, and discharges into an elbow draft tube. The draft tube is a single conduit which discharges into a short length of river and into the lower reservoir.

$$\Omega^* = \frac{\omega\sqrt{Q^*}}{(2gH_n)^{3/4}} \tag{1}$$

The power plant is shown in Figure 1, which also indicates the locations of the measurement sections.

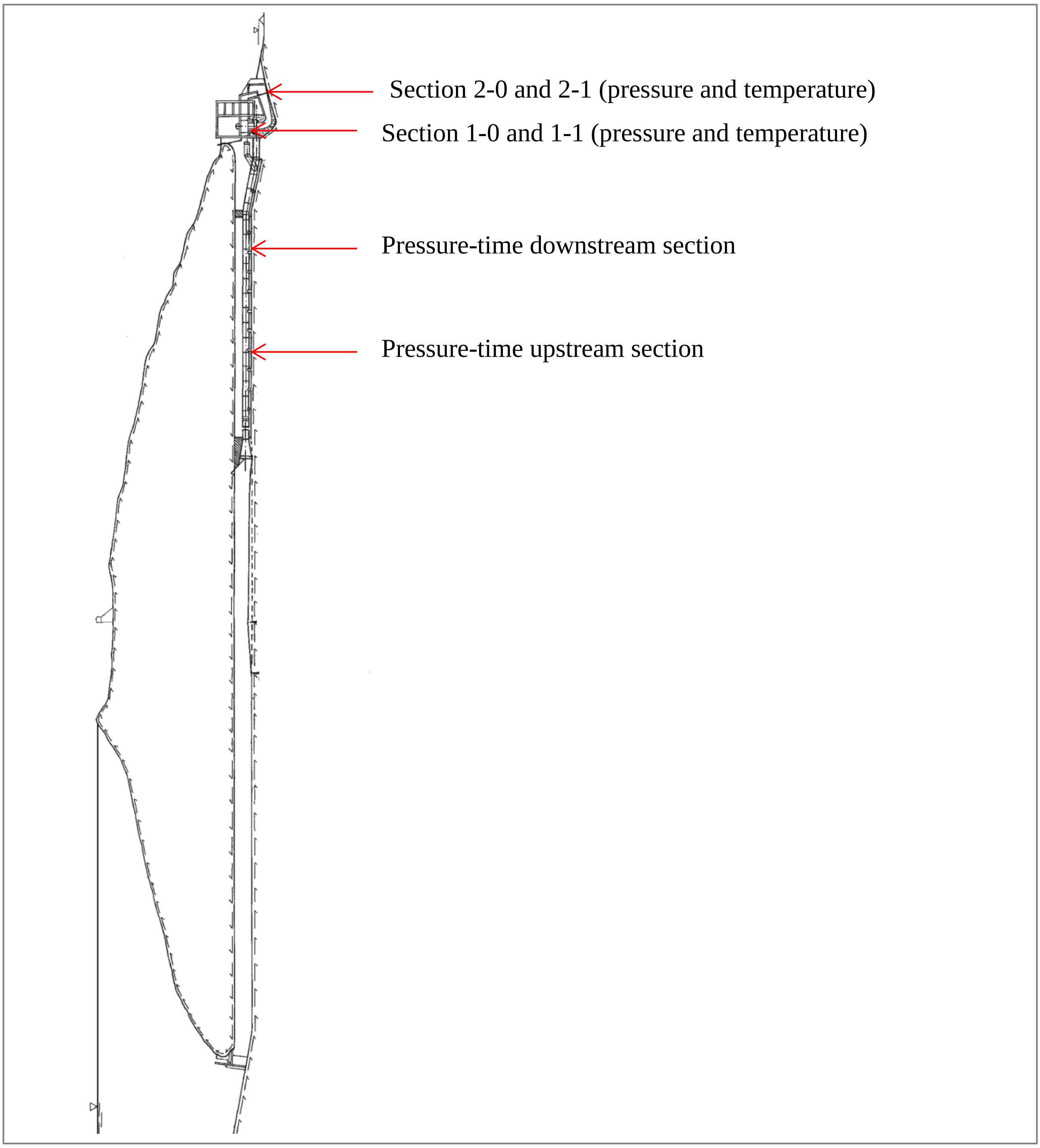

Figure 1 Lengthwise cross-section of Gråsjø power plant

2 DESCRIPTION OF EFFICIENCY TESTS

The efficiency test before refurbishment was done on 29th to 30th November 2011.

The efficiency tests after refurbishment were carried out in the period from 11th to 14th November 2013. Two days were used for the thermodynamic test, and two days were used for the pressure-time test. The Chief of Test of the thermodynamic test was Mr. E. Nilsen, and the Chief of Test of the pressure-time test was Mr. L. Parr. For both tests, the same equipment and setup were used for inlet and outlet pressure and level measurements, Winter-Kennedy differential pressure measurements, as well as generator power measurements.

3 PRESSURE-TIME METHOD

The penstock is equipped with two measurement sections with two pressure taps in each section. The length between the pressure taps is 30 m, and the flow area is 5.28 m^2, which gives a pipe factor F [1] of 5.7 m^{-1}. The product of the length between the pressure taps and the full load velocity is 170. The length and area were measured on site during refurbishment.

Two pressure transducers were mounted directly on the pipe wall in both measurement sections, one transducer for each pressure tap. The transducers were sampled simultaneously with at a rate of 320 samples/s through a 16 bit D/A converter. The transducers were calibrated before and after the test and found to be better than 0.04% of full-scale range. The nominal accuracy of the transducers is 0.06% of full-scale range.

The pressure-time measurements were made according to normal procedures. The discharge is calculated using a special software which uses the iterative procedure described in [3]. The turbine flow is the average of the four flows calculated based on the four combinations of differential pressure between the individual transducers.

The leakage flow through the wicket gate was determined using two separate methods. The first method consists of using the pressure-time method for integrating the (small) pressure-rise at closure of the main inlet valve. This was done for all measurement points, and averaged. The second method consists of measuring the differential pressure across the equalizing pipe of the main inlet valve when the main valve is closed. By applying theoretical loss factors to the pipe components of the equalizing pipe, a leakage flow is computed. The leakage flow calculated by the two methods agreed within 2%. The leakage flow is 0.3% of the full load turbine flow.

4 THERMODYNAMIC METHOD

The temperature at the inlet of the turbine was measured using one measuring vessel, which extracts a continuous small water sample which flows past the thermometer.

The temperature at the outlet of the turbine was measured using three thermometers located in three pipes. The pipes were perforated with small holes upstream and downstream in the flow direction, allowing for water flow past the thermometer at a low velocity. The pipes were fixed between the draft tube gate (which was located in the upper position) and the draft tube floor. The thermometers were raised and lowered manually to three different vertical positions for each test, which give a total of nine temperature sampling points at the turbine discharge. Due to limitations of time and cost, the corresponding velocity distribution was not measured. The efficiency is calculated based on the average of the nine differential temperatures.

High-quality PT100 elements read through a high-precision multiplexer were used for the temperature measurements.

The leakage flow through the upper labyrinth seal was re-introduced into the main flow in the draft tube below the runner. Therefore, the loss in the upper labyrinth seal did not need to be measured separately.

5 WINTER-KENNEDY METHOD

Winter-Kennedy differential pressure was measured during both tests after refurbishment. The Winter-Kennedy constant k and n (Eq. 2) are calculated based on the best fit of both the thermodynamic results and the pressure-time results. Unfortunately, the presence of Winter-Kennedy pressure taps was not discovered when the efficiency test was made before refurbishment. Therefore, no Winter-Kennedy differential pressure measurements exist for the test before refurbishment.

$$Q = kh^n \tag{2}$$

6 OTHER MEASUREMENTS

The inlet pressure was measured upstream of the spiral casing using four connected pressure taps which were led to a high-accuracy pressure transducer. The outlet pressure was measured as the level of the free water surface in the draft tube gate surge shaft.

The generator power output was measured at the output terminals using the permanently installed voltage and current transformers and kilowatt-hour meter. The generator losses used to calculate turbine power were measured at site in 1971, when the power plant was commissioned. No major revisions or changes have been made to the generator since commissioning.

7 RESULTS

The measurement conditions were very good for both the pressure-time tests and the thermodynamic test, with highly stable operation of the unit and stable inlet pressure readings.

For the thermodynamic test, the temperature at the inlet was stable during the test. At the outlet, the temperature readings at low loads were influenced by the discharge of the cooling water into the draft tube surge shaft. No temporary re-routing of the cooling water was possible. For these points, the temperature registrations in the upper position have been discarded.

The main results are presented in three figures. In all figures, the normalized efficiency is plotted versus the turbine power referred to nominal net head divided by the turbine power at the best efficiency point (P/P*). The turbine power has been chosen for the x-axis, as this parameter is measured by the same equipment, and calculated in the same way for both test methods. The best efficiency point is taken to be the highest level of the polynomial based on the pressure-time test after refurbishment. The efficiency is normalized by lifting the best efficiency of the polynomial based on the pressure-time test after refurbishment to 100%. The difference between this level and the actual efficiency is added to all other efficiencies. This is done to preserve the information of the turbine supplier.

The efficiency curves are represented by fourth order polynomials, except for the efficiency curve before refurbishment, which is of third order. The polynomials are fit using the method of least squares.

Figure 2 presents a comparison of the efficiency measured using the pressure-time and thermodynamic method. The fitted polynomial efficiency curves of the pressure-time and thermodynamic tests after refurbishment are almost identical, except for at the lowest load. The maximum difference between the curves is 0.5 pp at the lowest load. At loads between relative powers (P/P*) of 0.5 and 1.15, the difference between the curves is less than 0.15 pp.

Figure 3 presents a comparison of the results of the Winter-Kennedy test to the pressure-time and thermodynamic efficiency curves. The Winter-Kennedy test is comprised of measurement points taken at the same tiime as both the pressure-time and thermodynamic tests.

Figure 4 presents a comparison of efficiency tests made before and after refurbishment and runner replacement.

The tests made after refurbishment were made at net heads between 50.8 – 53.4 m. The test made before refurbishment was made at net heads between 54.7 – 56.1 m.

The uncertainty in efficiency was ±1.4% for the pressure-time test and ±1.7% for the thermodynamic test. The uncertainty in turbine power was ±0.9%.

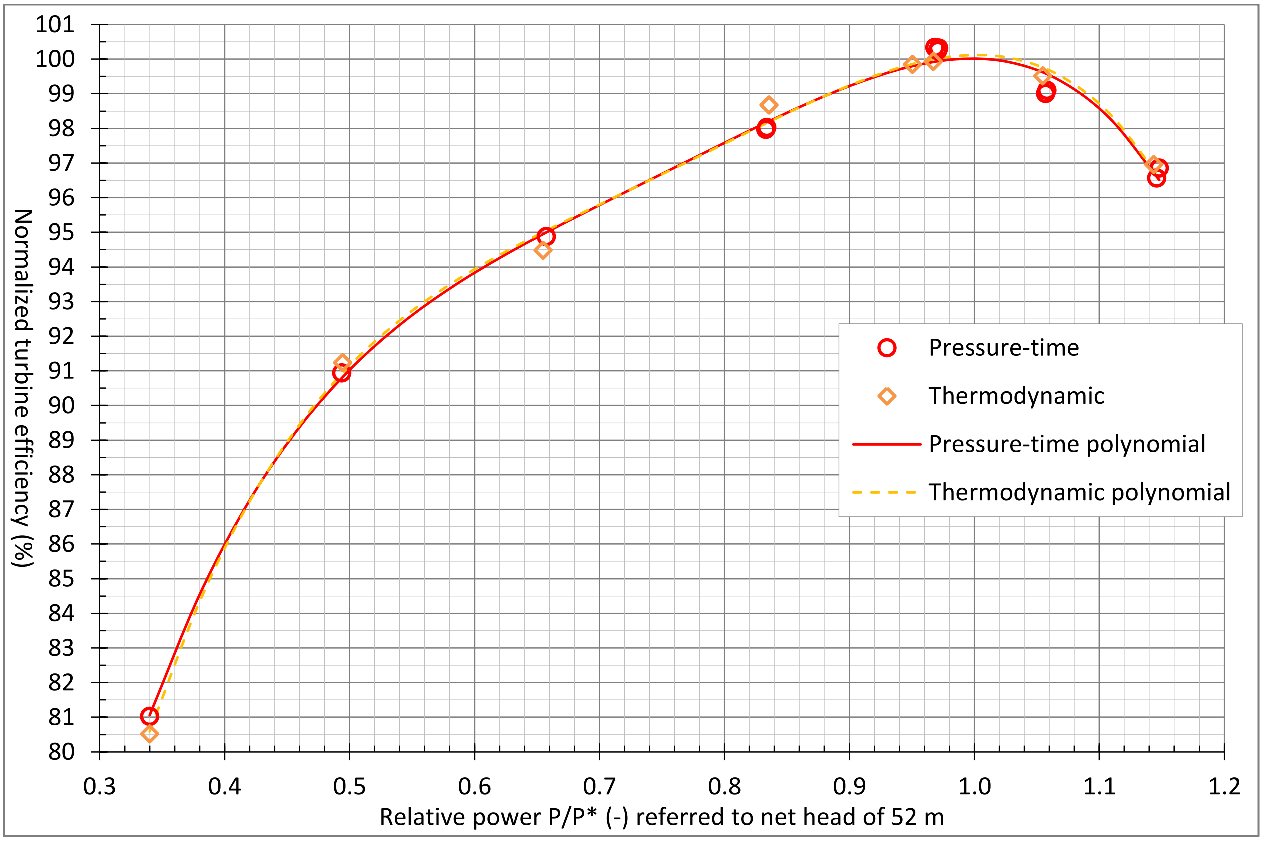

Figure 2 Comparison of pressure-time and thermodynamic tests

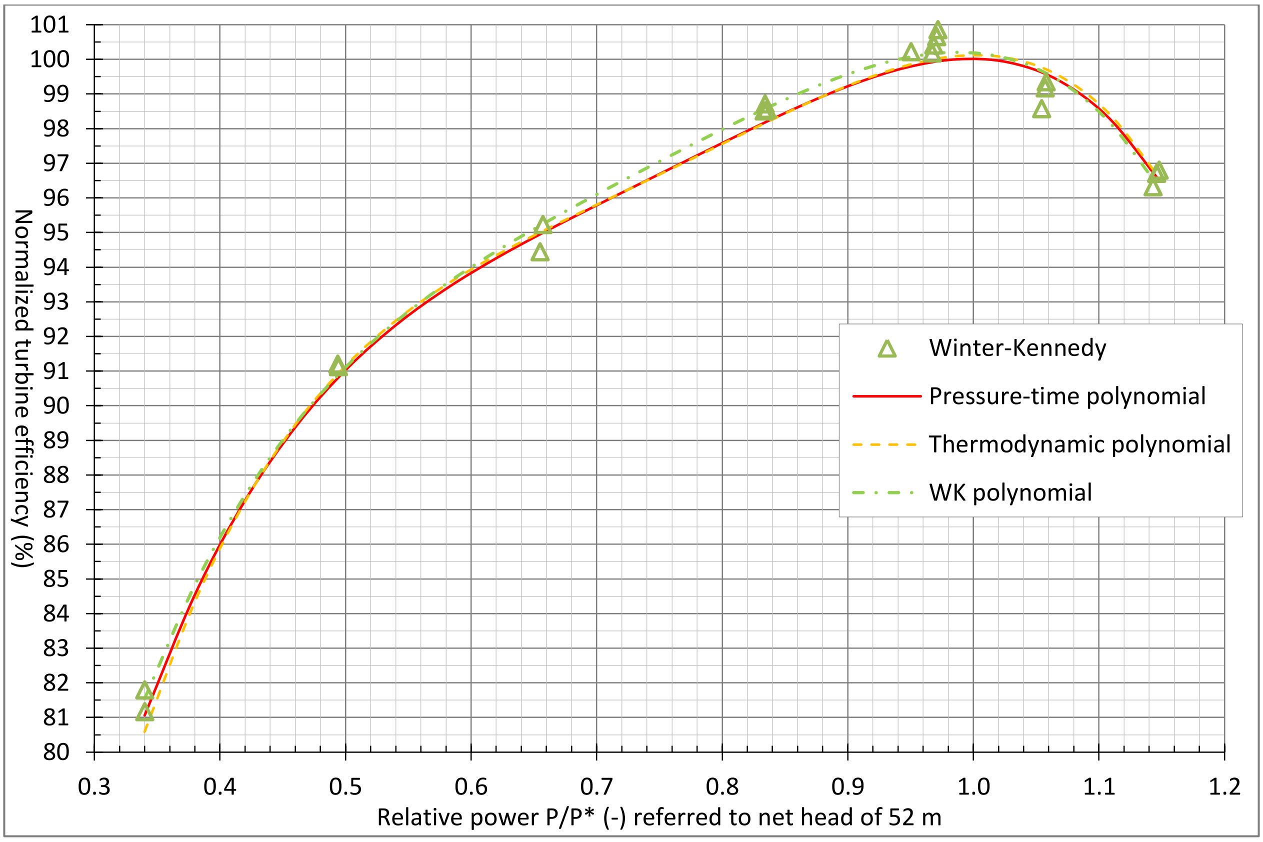

Figure 3 Comparison of Winter-Kennedy test to pressure-time and thermodynamic polynomials

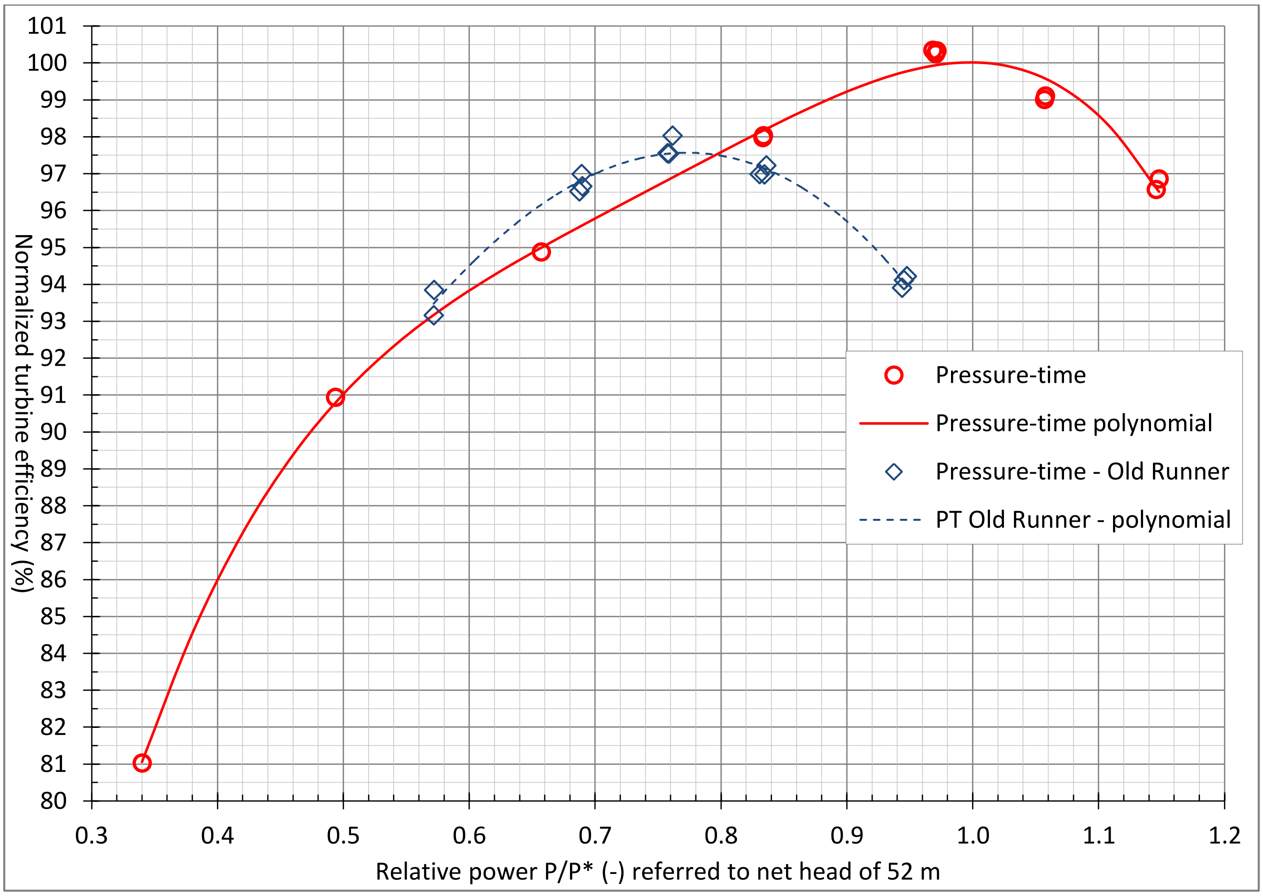

Figure 4 Comparison of efficiency of old and new runner

The numerical values of relative power and normalized efficiency for the tests after refurbishment are presented in Table 1. The difference in percentage points between efficiencies measured by the pressure-time method, the thermodynamic method, and the Winter-Kennedy method are also given in the table. To produce the points in the table, the test runs taken at the same relative power have been averaged. For the pressure-time test, the number of repeated test runs and their standard deviation is also given in the table.

Table 1 Relative power and normalized efficiency for tests after refurbishment

Relative power P/P*	(-)	0.34	0.49	0.66	0.83	0.97	1.06	1.15				
Efficiency - Pressure-time	(%)	81.02	90.94	94.88	98.00	100.30	99.05	96.71				
Number of runs - PT	(-)	1	1	1	2	3	2	2				
Standard deviation - PT	(pp)	-	-	-	0.03	0.05	0.07	0.20				
Efficiency - Thermodynamic	(%)	80.52	91.24	94.48	98.67	99.93	99.52	96.96				
Efficiency - Winter-Kennedy	(%)	81.48	91.16	94.84	98.59	100.53	99.03	96.61	avg	min	max	stdev
Difference PT - thermo	(pp)	0.50	-0.30	0.40	-0.67	0.37	-0.46	-0.25	-0.06	-0.67	0.50	0.47
Difference PT - WK	(pp)	-0.46	-0.23	0.04	-0.59	-0.22	0.02	0.11	-0.19	-0.59	0.11	0.27

The temperature distribution in the draft tube outlet for the thermodynamic method is presented in Figure 5, in which the efficiency has been calculated separately for each measured temperature in the draft tube outlet. This is shown for the best efficiency point.

Draft tube outlet seen against flow direction

-1.7 pp	0.2 pp	0.8 pp

Thermodynamic method at best efficiency point - deviation of efficiency from average based on individual draft tube temperatures

-2.1 pp	-1.8 pp	0.7 pp

P/P* = 0.97, Average efficiency = 99.9 %
Standard deviation = 1.5 pp
Maximum - minimum = 3.9 pp

0.7 pp	1.3 pp	1.8 pp

Figure 5 Temperature distribution given as difference of efficiency from average in draft tube outlet

8 CONCLUSIONS

The main conclusion of this paper is that the efficiency curves produced by testing with the pressure-time method and with the thermodynamic method are nearly equal, within ± 0.15 pp for relative powers 0.5 – 1.15, even though the deviation of individual test points is greater.

Given the significant pipe factor, the pressure-time method is well suited for Gråsjø power plant, except for the fact that the measurements of the generator losses date back 42 years to commissioning. However, for the purpose of measuring efficiency increase of runner replacement, this factor is not of importance. The repeatability of test runs at the same power is good, with a standard deviation of 0.05 pp for three repeated runs at the best efficiency point.

Regarding the thermodynamic method, the distribution of temperature in the draft tube gives a higher variation in calculated efficiency between two locations than what is accepted in the standard [1], which is 1.5%. Regardless of this fact, the resulting efficiency curve is satisfactory. The stable inlet temperature and the relatively small draft tube dimensions contributes to increase the accuracy of the method. The stability of the inlet temperature is influenced by the reservoir temperature distribution and layers, and the location of the intake. The test was done in late fall, and the reservoir was partly covered with ice. The intake is well submerged at the bottom of the reservoir. Both factors typically contribute to a stable inlet temperature.

It is worth noting that both the pressure-time and the thermodynamic tests take about two days at site including tests and installation (given that the site is already prepared with pressure taps etc). The cost of performing the tests using either method is in this case about equal.

The Winter-Kennedy test fits well with the absolute efficiency tests, using an exponent n of 0.5. Unfortunately, no Winter-Kennedy test was made before runner replacement.

The tests made before and after refurbishment show that the efficiency increase between the the best-efficiency points was 2.4 pp.

NOMENCLATURE

Ω	Specific speed	(-)	**Subscripts**	
Q	Flow	(m^3/s)	*	At best efficiency point
ω	Rotational speed	(rad/s)	n	Net (head)
g	Acceleration of gravity	(m/s^2)		
H	Head	(m)		
F	Pipe factor	(m^{-1})		
k	Winter-Kennedy constant	(m^3/sm^a)		
h	Winter-Kennedy differential pressure	(m)		
n	Winter-Kennedy exponent	(-)		
P	Turbine power	(W)		

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