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Evaluating the efficiency of low pressure part of steam turbines based on probing measurements

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Abstract

A methodology oflow pressure part turbine efficiency evaluation based on measurements of the steam flow parameters in the interspaces between neighbouring stages is described. Specially manufactured probes have been applied carry out such measurements. The efficiencies of the stages operating in the superheated steam zone result directly from experimental values of pressures, temperatures, and flow angles. To complete the efficiency evaluation for the stages operating in the wet steam region, a relevant estimation of blading system losses has been proposed and validated. This evaluation of losses is compatible with the measurement results. Adaptation of a comparative error analysis makes it possible to show the advantages of the methodology over the thermal balance applied during performance tests. The low pressure turbine efficiency evaluation methodology has been applied to numerous steam turbine power units of 200–500 MW output.

Keywords: Wet steam; Steam turbines; Measurements; Numerical calculations

Nomenclature

c	–	velocity
d_{32}	–	Sauter droplet diameter
g, G	–	mass flow rate
h, H	–	enthalpy, enthalpy drop
k	–	Baumann coefficients

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K	–	kinetic energy at the stage
k_1, k_2	–	coefficients
L	–	length
p	–	pressure
r	–	radial coordinate of the probe position
s	–	specific entropy
t	–	temperature
v	–	specific volume
x	–	any thermal or flow parameter
y	–	steam wetness

Greek symbols

α	–	inclination angle
γ	–	meridional angle
Δh	–	loss of enthalpy
η	–	efficiency
ξ	–	relative loss
ρ	–	density
σ	–	error
ω	–	circumferential angle

Subscripts

c	–	condenser
i	–	internal
in	–	inlet
out	–	outlet
LP	–	low pressure outlet
n	–	number of stage (group of stages)
max	–	maximum
r	–	rotor row, radial coordinate
s	–	stator nozzle
T	–	total parameters

1 Introduction

Designing the low pressure (LP) stages of large power steam turbines is one of the most difficult engineering challenges. Ever more advanced numerical methods are being used for this purpose, along with the most recent design and technological techniques. Unfortunately, turbine designers still face problems with the experimental verification of the guaranteed operational parameters, mainly the efficiency.

This situation arises from the fact that the efficiency can be precisely evaluated only for stages operating within the superheated steam regime for which we can assess the steam enthalpy from direct pressure and temperature measurements.

For stages working in the wet steam regime, which is the case of LP turbine stages and where the isotherms and isobars are coincident, the steam wetness has to be measured additionally. Unfortunately, this measurement cannot be conducted with sufficient precision in the turbine (see: Krzyzanowski [14]; Kleitz and Dorey [12]). Most turbine producers and users use the guarantee measurement results for this purpose (ASME Standards, 2005). In that case the efficiency of the LP part results from the balance of mass and energy measured for particular turbine components. According to the results of relevant analyses, this balance is weighted with a relatively high uncertainty by Namieśnik and Gardzilewicz [18].

The above factors were the motivation for searching for other methods of measurement-based efficiency evaluation for the stages operating in the wet steam regime. One of best known methodologies of this kind has been proposed by Moore [16]. This methodology is based on the steam angular momentum, which makes it possible to determine the stage efficiency from probe measurements of pressure distributions along traverse lines between the blade rows of the turbine. The measured data are processed using the Euler equation and are complemented by simplified calculations of the radial loss distribution.

High costs, problems with measurements, especially in the stator/rotor plane where supersonic velocities exist, along with relatively primitive methods of loss evaluation are the reasons why this methodology has not been widely accepted in in practice for LP turbine assessment.

The paper presents another, more simplified, approach in which probe measurements are performed only in the interstage planes. This approach does not affect the efficiency evaluation of the stages working in the superheated steam regime but eliminates the need for experimental verification of the calculated angular momenta of the rotors working in the wet steam regime. This lack of data is compensated by more precise calculation of stage losses which are realised using computer codes solving the 3D Navier-Stokes equations and validated by probe measurements.

2 Measuring equipment

A typical arrangement of the measuring station installed on a 360 MW turbine is shown in Fig. 1 (see: Gardzilewicz *et al.*, [7]). Probes designed by Marcinkowski [15] are applied. By using these special measuring probes it is possible to record the distributions of the steam parameters in control areas between the stages. These probes allow:

- the measurement of total and static pressures and temperatures, and flow angles, and

- visualisation of the flow and localisation of the beginning of condensation.

The probes can be mounted without interrupting the turbine operation in special probe seats prepared earlier during a turbine overhaul. The probes are usually 2–3.5 m long and provide an opportunity for performing traverses in the radial direction only. At the turbine exit, where a large circumferential asymmetry of the flow parameters is observed, three probes are simultaneously used as shown in Fig. 1. Moreover, the results of the measurements are complemented by static pressure measurements obtained at a series of points evenly distributed along the inner and outer flow passage perimeter.

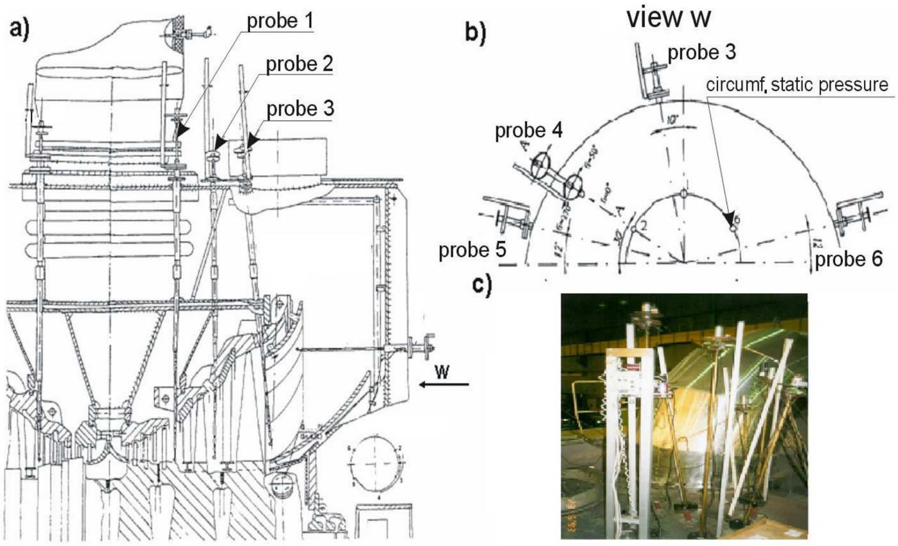

Figure 1: Schematic diagram of a 360 MW turbine with measuring equipment.

The head of the probe referred to as the disc-probe is shown in Fig. 2. The meridional angle, γ, is measured according to the pressure indication in the four front holes, Fig. 2. The yaw angle, α, is measured by rotating the probe until the static pressure measured on the left disc is equal to the static pressure on the right disc. This type of probe is insensitive to water films inundation. The diameter of the head is 20 mm. Its miniaturisation has been made possible by applying a permanent bleed flow through the measuring duct covered with teflon to protect the measuring tubes against the creation of internal water locks. The bleed flow is stopped only for the short time needed for a pressure measurement.

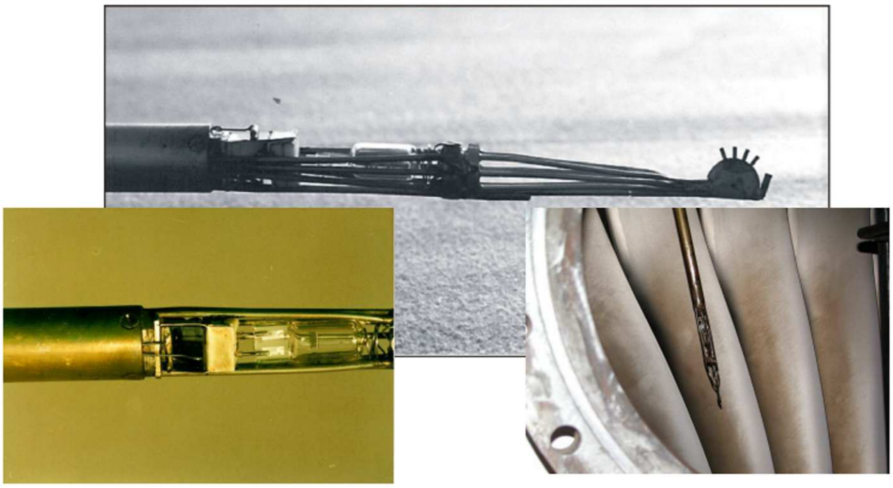

Figure 2: The probe with optical system for thermal and flow measurements.

Before each series of measurements the probes are calibrated in laboratory rigs. This procedure, along with stabilisation of the turbine-generator operating parameters, provides the opportunity for high-accuracy measurements. The temperature is measured with an uncertainty of ± 0.5 K and the pressure with an uncertainty of $\pm 0.5\%$. The uncertainty in measuring the flow angles was $\pm 2^\circ$ in the circumferential plane and $\pm 10^\circ$ in the meridional plane. The probe is equipped with a ruler and a ring with two protruding pins to secure the proper alignment and required depth of probe immersion in the flow path. The measuring procedures are described in detail by Gardzilewicz *et al.* [6,8] and Blazko *et al.* [3].

3 Methodology of LP turbine efficiency evaluation

A detailed description of the methodology is presented by Gardzilewicz and Marcinkowski in [5]. The efficiency of the LP part of the steam turbine is related to the assessment of flow losses. In the reported methodology the loss assessment procedure is divided into two parts, valid for stages operating in the superheated and wet steam regimes, respectively (Blazko *et al.*, [3]). This is illustrated by the shape of the expansion line on the Mollier diagram (Fig. 3).

For the stages working in the superheated steam regime, the loss is calculated

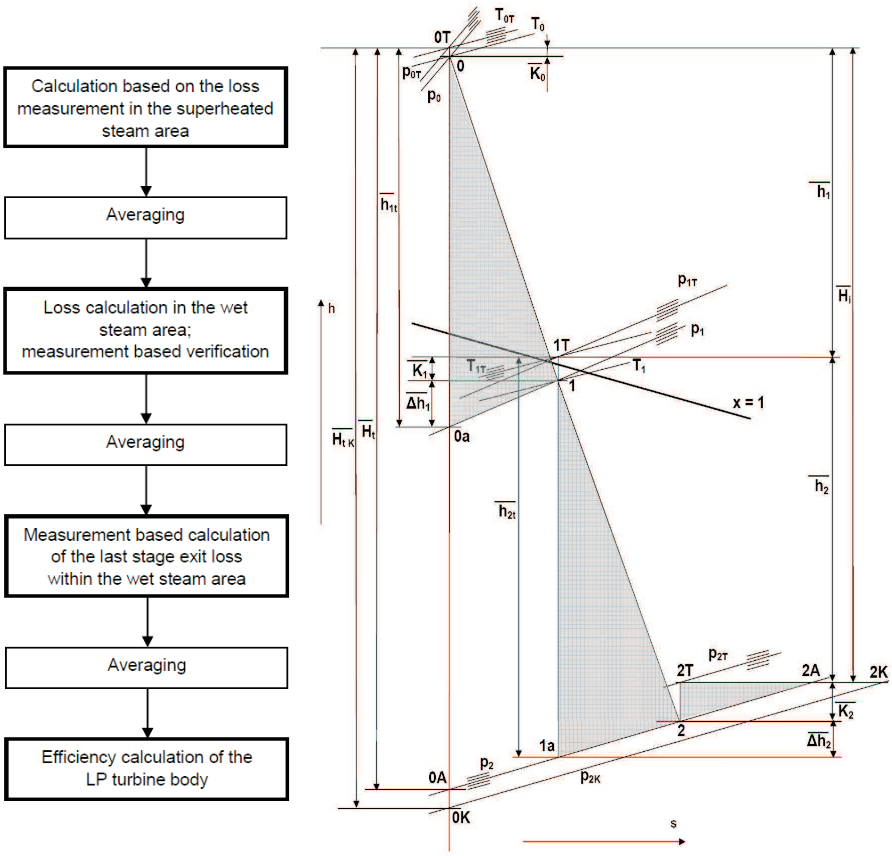

Figure 3: The expansion line for a low pressure turbine.

from the averaged enthalpy values as (see Fig. 3):

$$\overline{\Delta h}_1 \;=\; \overline{h}_{1t} \;-\; \overline{h}_1 \tag{1}$$

The enthalpy values are obtained directly from the measured pressures and temperatures at the inlets and exits of the examined stages. The procedure for their mass averaging is accomplished by applying the following general equation,

$$\overline{x} \;=\; \frac{1}{G(r)} \int_{r_{in}}^{r_{out}} g(r)\, x(r)\, 2\pi\, r dr \; . \tag{2}$$

The distribution of mass flow rate per unit area along the radius, $g(r)$, is calculated from the experimentally measured steam velocity and density, and,

$$G(r) \;=\; 2\pi \int_{r_{in}}^{r_{out}} r\, g(r)\, dr \; . \tag{3}$$

The loss evaluation for those stages working in the wet steam region is also divided into two parts: Eq. (1) the energy loss in particular rows of the blading system, and Eq. (2) the exit loss, see Fig. 3.

In the absence of a wetness measurement, the enthalpy at point 2 is determined from the calculated energy loss in a particular row. These loss calculations use the known stage geometry, the thermodynamic parameters of the superheated steam at turbine inlet and the static pressure at the exit. Additional information which is used includes the circumferential asymmetry of the thermodynamic and flow parameters at exit (see: Szymaniak and Gardzilewicz [19]). The calculations were performed using the computer code *FlowER*, which solves the three-dimensional Navier-Stokes equations in the turbine geometry. The code has been validated on numerous sets of experimental flow data recorded in both laboratory turbine stages and in full-scale turbines in operation in Poland and abroad (Yershow *et* al., [24]).

The basic features of *FlowER* are:

- numerical integration of the unsteady Reynolds-averaged Navier-Stokes equations,
- two-equation k-ω SST turbulence model,
- implicit Godunov ENO scheme with second order of accuracy,
- circumferential averaging of flow parameters in the interrow spaces to calculate stage and multistage turbine flows,
- perfect gas or local properties of steam may be used depending on the aim of the calculation.

Due to the absence of the information of steam wetness, relevant corrections were made using the modified Baumann formula [2], by usage of additional coefficients [10,11,22,25].

$$\zeta_{wet} \;=\; k_1\, y_{in} \;+\; k_2 \left(\frac{y_{in} \;+\; y_{out}}{2} \right) \; . \tag{4}$$

The first term represents the energy loss for accelerating the primary droplets in the flow, while the second term determines the rotor blade braking loss resulting from the impact of large secondary droplets. The coefficients were evaluated from experimental investigations they were assessed at the level of $k_1 \approx 0.7$–0.9 , $k_2 \approx$

0.25–0.5 (Gardzilewicz *et al.* [8]).

The calculations usually involved two stages and a measure of their correctness was obtained by comparing the calculated results with the values measured in the interstage planes (as these did not form part of input data). In this case, the final loss $\Delta\overline{h_2}$ is the sum of the losses (including also the leakage losses) of the following stator and rotor blade rows operating in the wet steam area,

$$\Delta\overline{h_2} \;=\; \sum_{j=1}^{n}(\overline{\xi_s}\, h_s \;+\; \overline{\xi_r}\, h_r)\ , \tag{5}$$

The exit loss is proportional to the square of the steam velocity leaving the turbine. In the present methodology the exit velocity is calculated from averaged measurements of the static and total pressure at exit. This is related to the calculations of enthalpy at point 2 (see Fig. 3),

$$K_2 \;=\; \frac{\overline{c_2}^2}{2} \;=\; \overline{h_{2T}} \;-\; \overline{h_2}\ . \tag{6}$$

Finally, the efficiency of the LP turbine is calculated from the relationship,

$$\eta_{i\,LP} \;=\; \frac{H_i}{H_t} \;=\; \frac{H_t - \Delta h_1 - \Delta h_2 - K_2}{H_t}\ . \tag{7}$$

It is noteworthy that, in the investigations, the steam mass flow rate, which is needed for the averaging of flow parameters, was additionally verified by measuring the volume flow rate of the condensate in pipelines beyond the condenser and before the LP turbine regenerative heat exchangers.

The research completed by Gluch and his team focusing on the application and improvement of power cycle calculation methods has facilitated the determination of efficiency of the whole steam cycle and its components such as the steam turbine [9].

The accuracy of the probing method for LP turbine stages working in the superheated steam regime depends on the accuracy of the pressure and temperature measurements. Based on our own analyses, the maximum inaccuracy of this method was estimated to be 1.5–2.5% depending on the enthalpy drop in the stage group (Namiesnik and Gardzilewicz, [18]).

Evaluating the errors made in the efficiency calculations for stages operating in the wet steam regime is more complicated. Krzyżanowski solved this problem using approximate computer simulations [14]. The result depended not only on the pressure and flow angle measurements but also on the accuracy of the evaluation of the blade row losses (it is worth noting that all required thermal and flow parameters are properly averaged using the procedure described above and illustrated in Fig. 3)

$$\eta_{i\,LP} = f(p_{T\,in}, p_{in}, T_{T\,in}, \rho_{in}, p_{T\,out}, p_{out}, \gamma_{in}, \gamma_{out}, \xi_S, \xi_r)\ . \tag{8}$$

For individual stages the maximum inaccuracy was assessed at a higher level based on our own experiments

$$\sigma_{\max} \approx 3\text{-}6\% \tag{9}$$

Assuming that, for the entire LP turbine, the expected uncertainty is the weighted sum of the elementary inaccuracies of the two above analysed stage groups then, for the same enthalpy drop, the maximum expected errors are lower

$$\sigma_{\max\,LP} \approx 1.5\text{-}3\% \tag{10}$$

These values are comparable to the accuracy of the efficiency evaluation obtained for the guaranteed measurements, but the presented methodology provides designers with much more information on the steam flow in the turbine. Obviously, the standard mean deviations of the methodology may be smaller but multiple repetitions of time-consuming measurements are required to verify this.

4 Sample of results

Selected representative results obtained for the LP part of the 360 MW turbine with a ND37 exit in the Belchatow power plant by Gardzilewicz and Marcinkowski [5] are shown below. They have the form of radial distributions of the measured thermodynamic parameters and flow angles behind the penultimate and the last LP turbine stage (Figs. 4 and 5) operating in the wet steam region. The diagrams also include values of related parameters which are determined by loss calculations. Satisfactory correspondence of the recorded and calculated values at measurement positions can be seen in the mainstream area. However, that agreement gets worse at leakage areas.

Figure 6 shows the measured distributions of static and total pressure around the circumference at the turbine exit. These were used in the 3D flow loss calculations.

The averaged efficiency changes of the LP turbine under investigation are shown in Fig. 7 as a function of the volumetric flow rate. To provide opportunities for direct comparison with balance calculations, these efficiency values are related to the condenser pressure. The efficiency values estimated according to the probing methodology differ insignificantly from the values obtained by applying

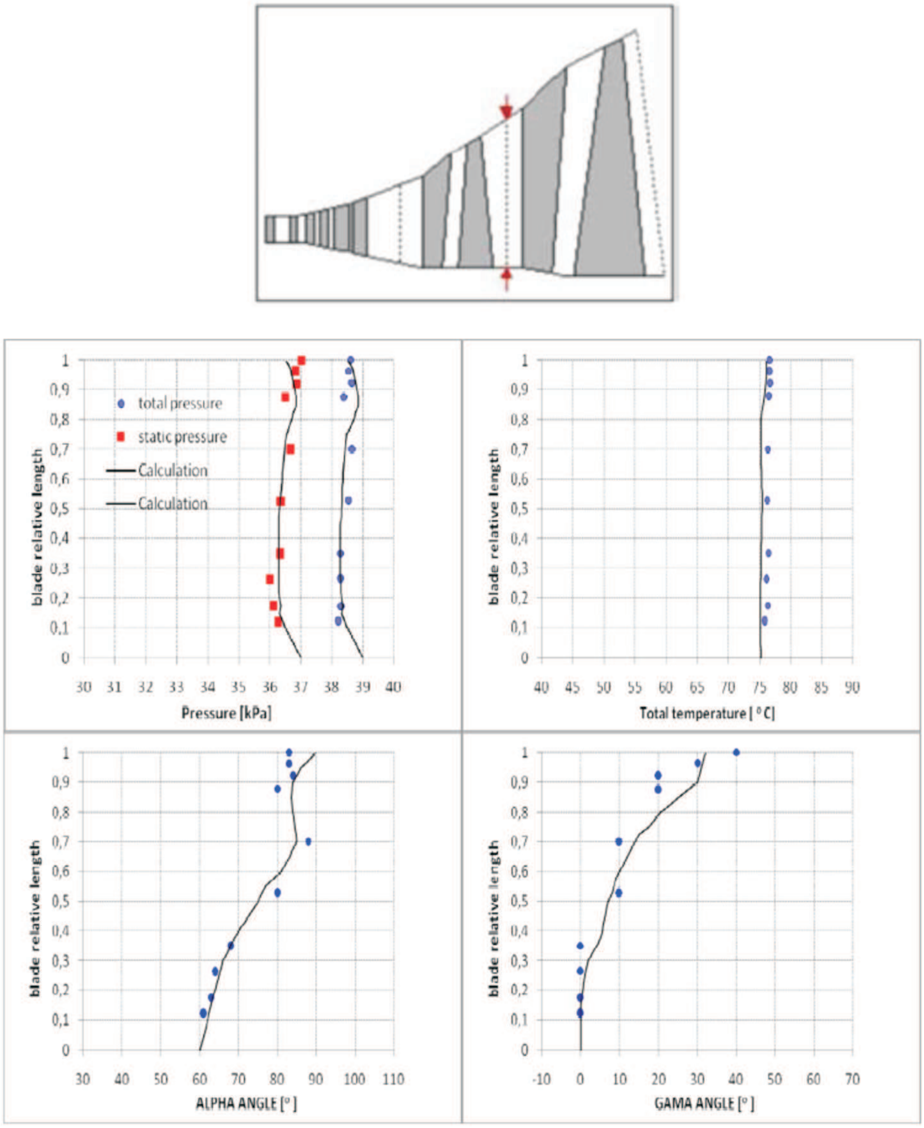

Figure 4: Radial distribution of pressures and flow angles behind the penultimate stage (probe No. 3) used for verification.

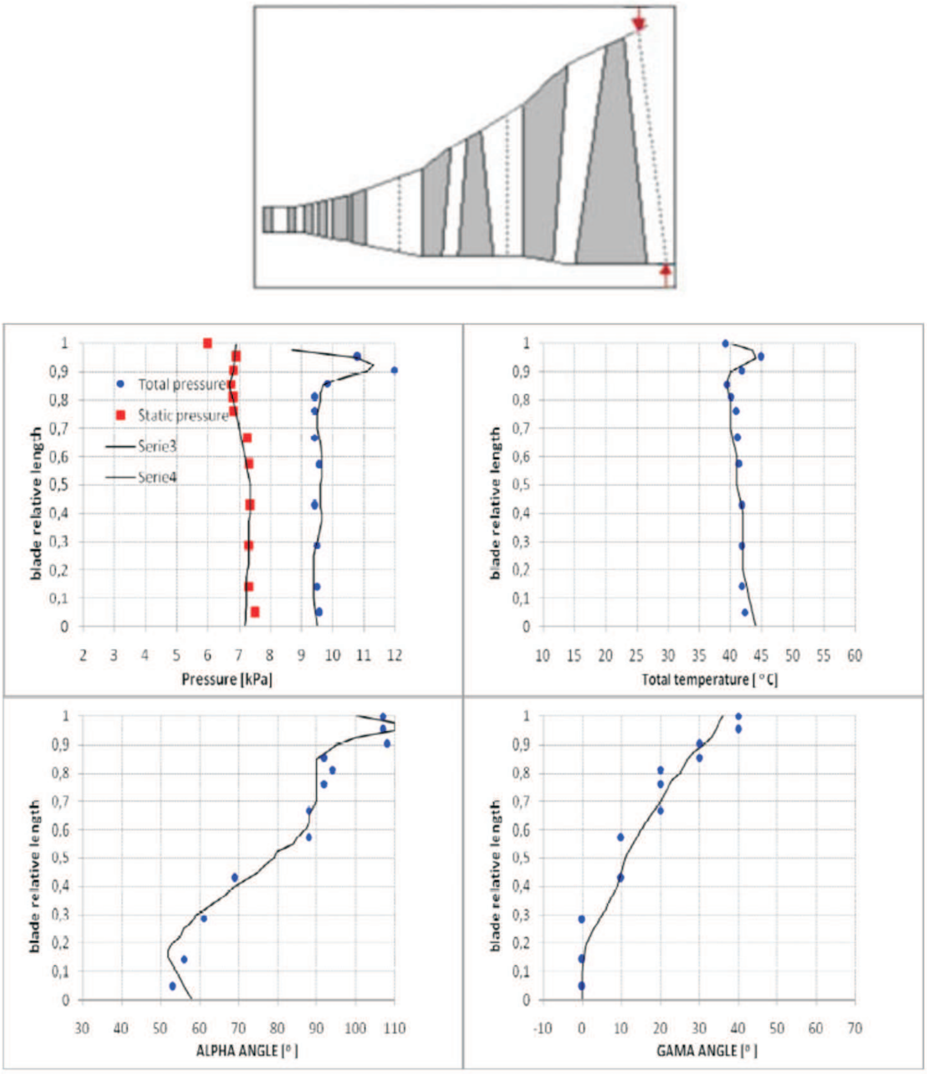

Figure 5: Radial distribution of pressures and flow angles behind the last stage (probe No. 5).

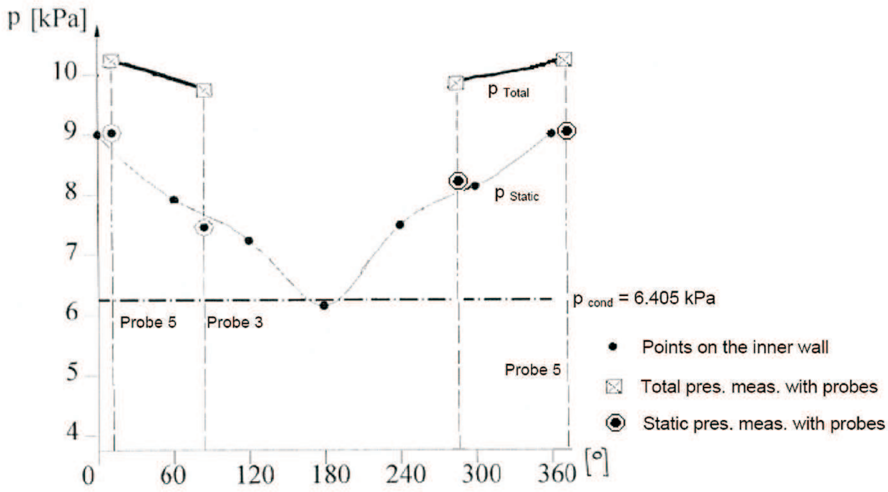

Figure 6: Circumferential distribution of static and total pressure behind the last stage, at turbine exit.

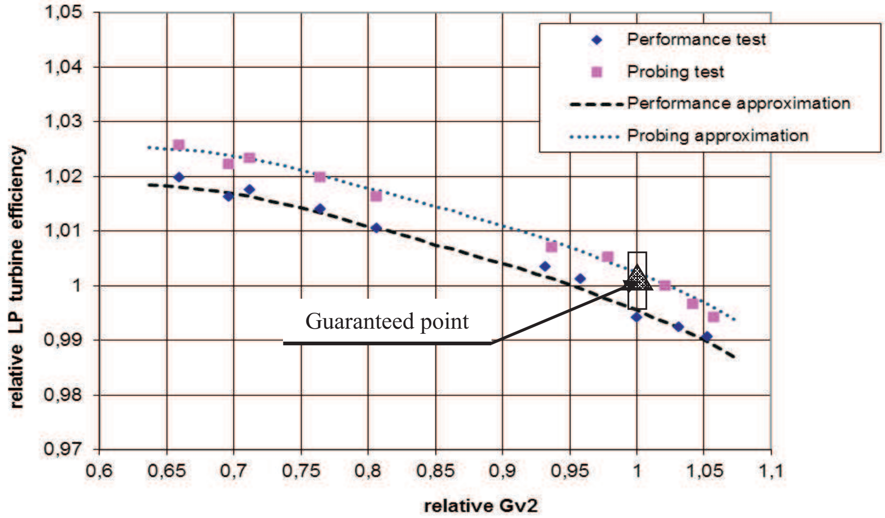

Figure 7: Efficiency of the LP part of a 360 MW turbine as a function of the volumetric flow rate.

the energy and mass balance methodologies. They are situated within the uncertainty bands of both mentioned methods. These facts confirm the predictions of

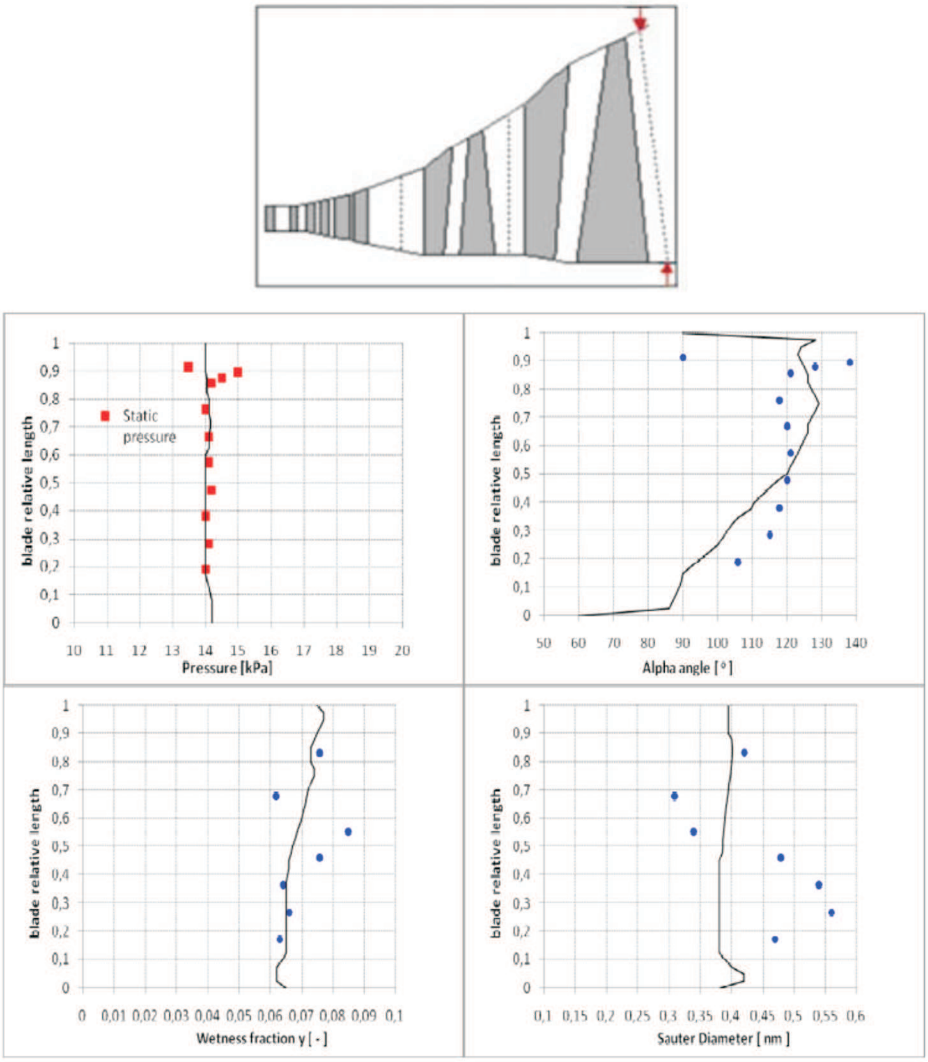

Figure 8: Static pressure, flow angle and wetness fraction behind the last stage. Calculations by Wroblewski *et al.* [23]) and experimental data by Gardzilewicz *et al.* [6] and Kolovratnik and Bartos [13].

the designers.

It is noteworthy that in the examined turbine the wetness level was also evaluated by Gardzilewicz and Marcinkowski [5]. The bases for this evaluation were measurements using an extinction probe of the number density and size of the droplets comprising the primary droplet mist. Selected results of these measurements, pbtained by Kolovratnik and Bartos [13] are shown in Fig. 8 and compared to the results of numerical calculations taking into account the process of homogeneous condensation resulting from the presence of chemical compounds in the steam. The contents of these chemical compounds were determined from tests on the primary condensate. This condensate was sampled by a special component of the previously mentioned probe. The measurement results have been compared with results calculated by the numerical codes developed by Wroblewski *et al.* [23]. The differences between the measured and calculated results are greater than in the previous cases. This is probably because of that the numerical codes do not, as yet, take into account the leakage flows nor the relatively large secondary droplets in the flow by Gardzilewicz *et al.* [6] which are responsible for part of the wetness loss.

5 Conclusions

Despite many theoretical and experimental investigations concerning wet steam flow in turbines, precise evaluation of the losses and efficiency is still a difficult task. The direct source of these difficulties is the problem of modeling the nonuniform structure of the liquid phase in the turbine steam flow. Therefore, the modified Baumann formula is still used by the world's largest steam turbine manufacturers in their design activities.

The relevant probing methodologies are highly labour-consuming and expensive and they do not lead to significant improvement in the accuracy of the efficiency determination. However, they have the advantage of providing designers with much more information on the steam expansion in particular turbine rows. This advantage is clearly visible in the performance analyses conducted by the authors of this paper for numerous large-power steam turbines.

Considering the difficulties in wetness measurements in real turbines, low pressure turbine stage efficiency determination based on energy and mass balances is still applicable. Performed test methodology is an example.

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