

Efficiency calculation on 10 MW experimental steam turbine

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Abstract. The paper deals with defining flow path efficiency of an experimental steam turbine by using measurement of flow, torque, pressures and temperatures. The configuration of the steam turbine flow path is briefly described. Measuring points and devices are defined. The paper indicates the advantages as well as disadvantages of flow path efficiency measurement using enthalpy and torque on the shaft. The efficiency evaluation by the help pressure and temperature measurement is influenced by flow parameter distribution and can provide different values of flow path efficiency. The efficiency determination by using of torque and mass flow measurement is more accurate and it is recommended for using. The disadvantage is relatively very complicated and expensive measuring system.

1 Introduction

Measurement of experimental turbine flow path efficiency is a rather complex, but at the same time a routine activity. However, attention must be paid to measuring all parameters influencing the efficiency calculation relation. It is necessary to adjust calculation relations so that they describe better the real behavior of the machine, and also to use more precise measuring equipment to calibrate individual sensors and to optimize the measuring chain in order to lower uncertainties and in some cases also measurement errors. Last but not least the experiment must be constructed so that the sensors, pressure samplings or probes are located in a suitable position to measure the required steam flow parameter typical for the whole measured cross section. It means, for example, that the thermometer in the blading outlet must be located at such a blade height and circumferential position where the expected temperature is very similar to the mean temperature in the whole cross section. It is obviously a very complex task not directly related with defining measurement uncertainties. The experiment designer will use his experience as well as some preparatory numerical simulations of the flow. On the other hand, the designer is limited by many construction limitations and a limited number of measuring points. These limitations often complicate data evaluation. At this moment numerical simulations must be used whose boundary conditions are more exactly set using the already measured values. A number of

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measuring places in the flow path serves only for verification whether at specific points the steam has the same parameters that were measured by the sensor. If so, then the CFD results can be taken as credible and usable for further analyses [1] serves as a suitable example.

Steam turbine efficiency can be defined by measured pressures and temperatures that can determine the steam enthalpy. This way of measurement is the only one possible when it is necessary to define efficiency of individual stages placed in one rotor. An example of such an experiment is shown in [2]. However, problems occur when wet water steam appears in the last stage outlet cross section or even before it. In this case the only possibility to determine flow path efficiency is to measure the mass flow through the flow path and the torque. Distribution of efficiency among individual stages is in all cases very complicated. This turbine configuration is often used when the last stage or group of stages is placed on a special rotor with a brake for defining only the torque of this last group of stages, e.g. [3, 4]. The design of flow path efficiency for the situation when wet steam is found in the last or the last four out of five stages is shown in [5].

The submitted paper aims to describe the flow path of a 10 MW experimental steam turbine as well as the measuring points and their positions. In the next part of the paper equations are described for defining efficiency using the flow, the torque and measurements of steam state. Equation members are briefly described. Advantages and disadvantages of efficiency determination using both types of equations are provided.

2 Description of experimental rig

Experimental measurement was carried out on the steam turbine with the configuration of the flow path named “Boiler Feed Pump Turbine (BFTP)”. It is a special design of flow path, when the steam, after passing through the radial inlet sieve, turns to the axial direction and flows through both stages. Behind the second stage one half of the steam continues to the third (and at the same time the last) stage and the diffuser. The second half of the steam is diverted before the third stage and flows through a bypass to the fourth stage and the following diffuser.

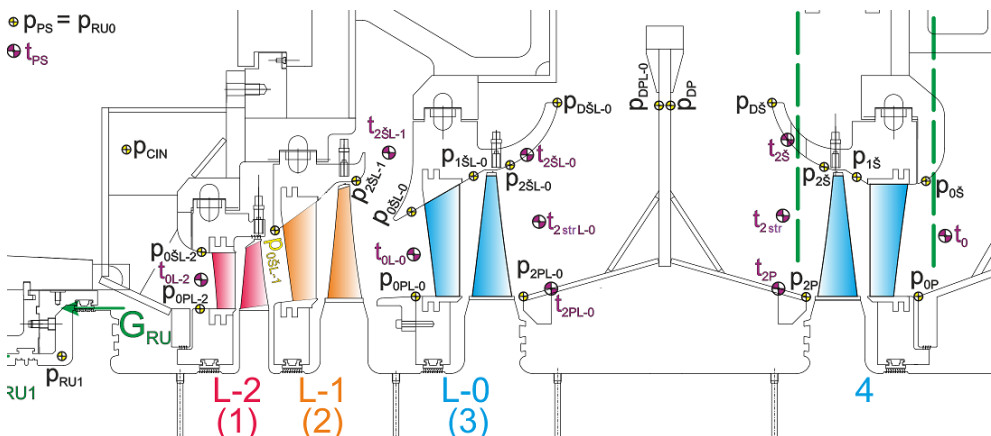


Fig. 1. Flow path with measuring places description.

In the outlet casing the steam from the last two stages unites. In the flow part and on the outer gland casing a total of 54 static pressure sample collecting places are located. All the pressures are doubled. It means that in each particular place, e.g. on the tip limiting wall behind the stationary blade the pressure sample is on the left and the right side of the turbine about 10° below the dividing plane. In case of one measuring place malfunction,

it is possible to use the second pressure. When both pressure pipes are working, the obtained data are averaged.

In the flow path also a total of 37 thermometers are placed. Again, the thermometers are circumferentially doubled. It is worth mentioning that outlets of both last stages are equipped with many measuring points, as it is necessary to measure the distribution of temperatures along the blade length for small turbine outputs, in the so called ventilation regimes.

In order to obtain complete knowledge of the flow path behaviour it is necessary to measure the torque twice (using the torque meter and the water brake) and the rotor speed. Last four important data are measurements of flow times of specific oil tank, main condensate and gland and vent steam condensates. Using known tank volume and water density it is possible to calculate mass flows.

Schematic section of the flow path is in Figure 1. All names of pressure samples are labelled with letter „p“. All temperatures are labelled with letter „t“.

The second scheme in Figure 2 shows basic steam or water flows important for calculation of efficiency using flow and torque measurement. During the measurement it is necessary to determine the steam flows using orifice of admission steam (flow G_{pc}) and gland steam (G_{vv}). The amount of water injected into the admission steam in order to regulate the inlet temperature G_{vv} is evaluated as well as the amount of water measured in the condensate tank G_k and the amount of water condensed after the passage through the gland system. Specifically, it is the amount that goes through the vent and gland steam condenser.

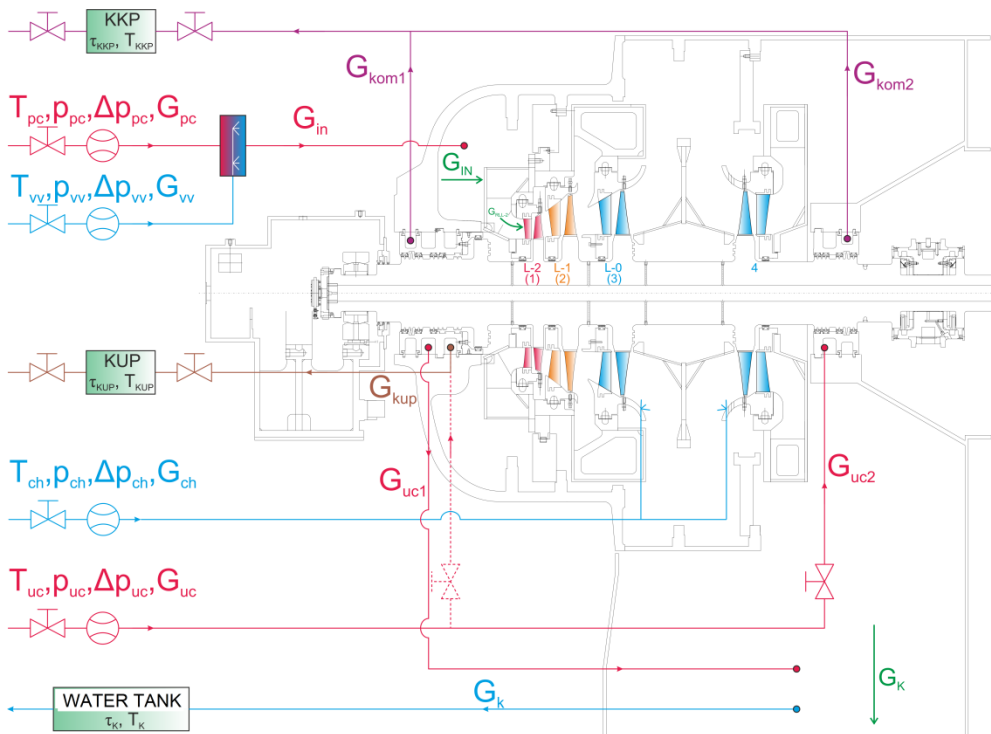


Fig. 2. Water and steam flow in the turbine.

3 Relationships for calculation

3.1. Calculation of efficiency based on temperatures and pressures

For calculations that use equation (1) it is expected that the whole flow path is located in the superheated steam area. In this case it is possible to define the relation for efficiency calculation static-static (in the inlet - index 0 – in the outlet – index 2 – from blading are considered static states) as follows:

$$\eta_{ss} = \frac{h_{0-2}}{h_{is0-2}} = \frac{i_0 - i_2}{i_0 - i_{2is}} \quad (1)$$

In practice the relation is considered with directly measured data:

$$\eta_{ss} = \frac{IAPWS_97(p_0, t_0) - IAPWS_97(p_2, t_2)}{IAPWS_97(p_0, t_0) - IAPWS_97(p_2, s_0)} \quad (2)$$

It is evident that for calculation of water steam qualities water steam tables are used in the industrial version IAPWS_97. Using these tables even entropy in the flow path inlet can be simply calculated as $s_0 = f(p_0, t_0)$.

It may seem that the easiest choice for efficiency calculation using temperatures and pressures are textbook relations. The static pressure in the blading inlet is measured on the tip and root limiting wall (altogether 4 samples). The temperature is measured in the middle of the blade length (altogether 2 temperatures). In both last stages the pressure is measured again on the tip and root limiting wall (altogether 4 samples) and temperature is measured even in the middle of the blades length (there are 6 temperature measurements behind each last stage).

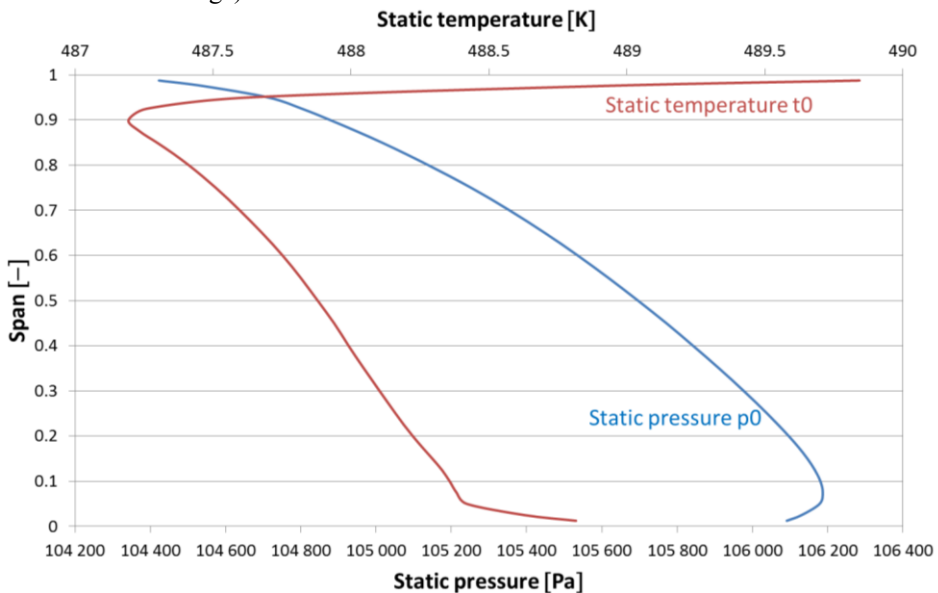


Fig. 3. Static pressure and temperature distribution in the turbine inlet.

Now it is possible to take a look at the results of numerical simulation of the flow with regard to pressure and temperature distribution along the blade height in the flow path inlet (in the plane labelled in relations by index 0). They are in Figure 3.

It is evident that both parameters are in no case evenly distributed. In Table 1 the calculation of flow part efficiency by various ways is shown providing the parameters in the blading outlet are constant and at the level $p_2 = 0.22 \text{ bar}$ and $t_2 = 89 \text{ }^\circ\text{C}$. All efficiencies are calculated using relation (2). It is in fact an expression of measurement error that can occur in an inappropriate location of the static pressure and temperature samples.

Table 1. Possibilities of efficiency calculation when changing inlet parameters.

Measured parameters	η_{ss}
Average value of static pressure (= 1.056 bar) and temperature (=215 °C) from CFD calculation	0.818
Static pressure value on the inner and outer limiting wall, thermometers are placed in the specific blade length 0.07 and 0.93 – i.e. about 10 mm in the steam flow.	0.819
Static pressure value on the inner and outer limiting wall, thermometers are placed in the specific blade length 0.15 and 0.85.	0.819
Static pressure value on the inner and outer limiting wall, thermometer is placed in the specific blade length 0.50. This configuration is used in the experiment	0.819
Static pressure value in the inlet is average from the probe measurement with step 0.1 dimensionless blade length; thermometer is placed in the specific blade length 0.50.	0.818

From Table 1 it is evident that the method of measurement of steam flow inlet parameters is suitable and even for an incorrect position of thermometers the efficiency values are not distorted. It is also obvious that it is not necessary to traverse the inlet pressure field distribution.

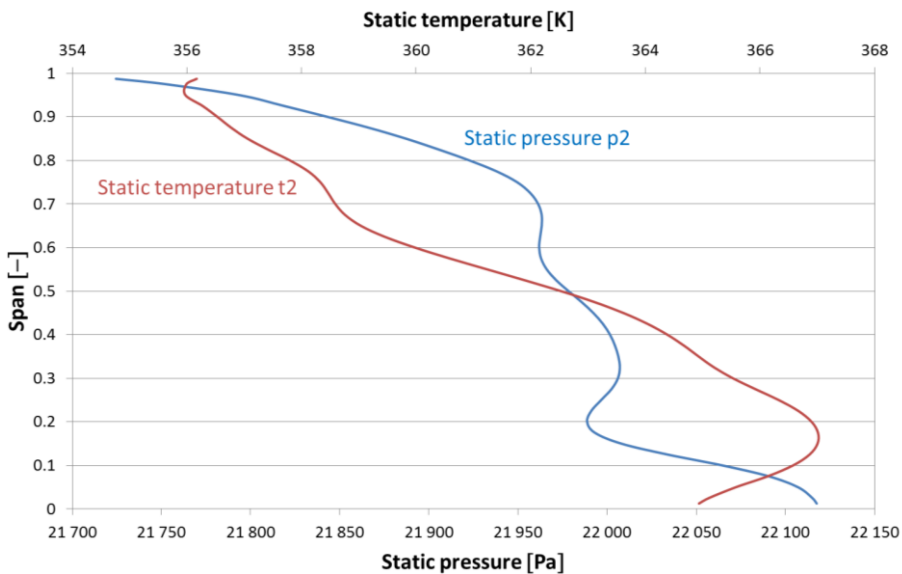


Fig. 4. Static pressure and temperature distribution in the turbine outlet.

The situation when there is constant pressure and temperature in the inlet (which is according to the data in Table 1 quite justifiable) and there will be changes in the pressure and velocity field behind the third stage) is presented in Table 2. In this case pressure $p_0 = 1.056$ bar and temperature $t_0 = 215$ °C. Distribution of steam flow parameters behind the 3rd stage is in Figure 4. It is evident that while the static pressure distribution is only with the difference of 400 Pa, the difference in minimal and maximal temperature in the flow field is up to 11 °C.

Table 2. Possibilities of efficiency calculation when changing outlet parameters.

Measured parameters	η_{ss}
Average value of static pressure (= 0.22 bar) and temperature (=89 °C) from CFD calculation	0.815
Static pressure value on the inner and outer limiting wall; thermometers are placed in the specific blade length 0.03 and 0.97 – i.e. about 10 mm in the steam flow.	0.823
Static pressure value on the inner and outer limiting wall; thermometers are placed in the specific blade length 0.05 and 0.95.	0.822
Static pressure value on the inner and outer limiting wall; thermometers are placed in the specific blade length 0.15 a 0.85.	0.814
Static pressure value on the inner and outer limiting wall; thermometers are placed in the specific blade length 0.03; 0.50 and 0.97. This configuration is used in the experiment	0.819
Static pressure value on the inner and outer limiting wall; thermometer is placed in the specific blade length 0.50.	0.810
Static pressure value in the inlet is average from the probe measurement with step 0.1 dimensionless blade length; thermometer is placed in the specific blade length 0.50.	0.811

From Table 2 it is evident that outlet parameters have considerably higher influence on efficiency than inlet parameters. At the same time the outlet flow field is very unbalanced and the location of thermometers positions is very important. In general, the longer the blade and the lower the steam parameters behind it, the higher a possible measurement error is caused by the influence of wrong location of measuring sensors.

3.2. Efficiency calculation based on flow and torque measurement

For the above mentioned reasons it is advisable to try to measure efficiency in a different way. It is necessary to measure also the flow through the blading and the torque. However, this is a significant technical complication for the whole experiment design and implication. The relation for the efficiency calculation is presented here:

$$\eta_{t-b} = \frac{3600 \cdot N_{nl}}{G_{in} \cdot (h_{1zp} + h_{is0-2})} \quad (3)$$

Where N_{nl} is the output on turbine blades, G_{in} is the flow mass in the first stage inlet. Then h_{1zp} is enthalpy drop calculated from inlet axial velocity, density and flow cross

section or using total pressure probe. Drop h_{celk_is} is the total drop between the static state in the inlet and the total state in the blading outlet.

It is necessary to realize that the output on blades N_{nl} is the sum of outputs (torques) measured by torque meter or on the brake, torque loss caused by discs' friction and cylindrical rotor surfaces, and in the bearings. The torque loss in bearings is measured using oil heating and flow through the bearings. Here thus fundamental measurement error cannot be expected. The problem is to determine the disc friction losses. There are many relations available [6]. In general, it can be said that for the given example of the flow part the friction losses are, because of low steam density and a small rotor diameter, at the level of 0.1 % of the total torque. More important errors can occur in the situation when low turbine outputs are measured with high pressure steam, only slightly higher than total losses in bearings and by friction.

The calculation of total iso enthalpy drop h_{is0-2} is done using the relation of denominator in relation (2). As mentioned above, to determine the value of the bracket in relation (2) should not be a basic problem.

However, it is complicated to define the steam flow to the turbine. It can be done in two ways. Firstly by measuring the admission steam flow G_{pc} through the steam orifice and using orifice for injected water G_{vv} . Secondly by measuring the condensate flow using the filling time of the specific tank G_k . Theoretically, both ways of measurement should provide the same flow amount. In practice total agreement was not reached.

The difference in flows can be caused mainly by the amount of steam passing to the steam pipes drainage between the orifice and the turbine. It is a flow that cannot be exactly measured. For the flow through the specific tank it is necessary to realize that the steam from the rear gland casing G_T gets to the tank, but this steam did not go through the flow part. Thus it must be deduced from the flow measured by the tank. The scheme of the rear gland casing is in Figure 5. On the other hand it is necessary to deduce from the flow measured by orifices the steam leaving the front gland casing G_{ru} . At the same time the flow through the front gland casing whose scheme is in Figure 6, drained to the gland steam condenser G_{kup} . The flow through the front part of the gland is regulated by gland steam G_{uc1} . A part of the steam is drained to the vent steam condenser. According to Figure 2 the whole system is a sophisticated complex of flow measurements with the assumption of its tightness.

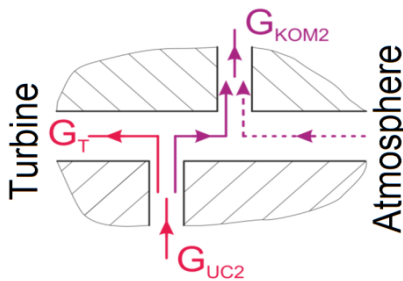


Fig. 5. Scheme of rear gland casing.

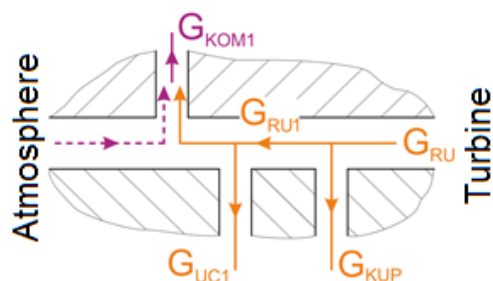


Fig. 6. Scheme of front gland casing.

From the flow balances it can be calculated that based on measured data G_{ru} (measured using flow through the gland), $G_{kom} = G_{kom1} + G_{kom2}$ (measured using vent steam condenser), G_{uc} (measured using gland steam orifice) and G_{kup} (measured using gland steam condenser), it is possible to derive the following equation:

$$G_{uck} = G_{ru} - G_{kom} + G_{uc} - G_{kup} \tag{4}$$

Amount G_{in} measured using condensate tank can be then defined as:

$$G_{ink} = G_k - G_{ch} - G_{uck} + G_v \tag{5}$$

In this equation the amount G_{ch} appears for cooling the turbine outlet parts and amount G_v , which represents the flow of saturated water steam to the vacuum pump. For drop $h_{is_celk} = 293$ kJ/kg, inlet drop $h_{lzp} = 4$ kJ/kg and output 4 460 kW two values of flow were measured.

Table 3. Influence of flow measurement on efficiency.

Flow defining	Flow value [kg/h]	η_{ss}
By orifice	65 900	0.820
By tank	64 500	0.838

It is evident that the difference in efficiency calculation using two ways of flow measurement can be (and usually is) in per cents. It is again a measurement error. The efficiency measurement uncertainties are often lower: less than 1 %. When measuring efficiency using water brake according to relation (3) it is possible to define steam enthalpy i_2 in the case when wet water steam occurred in the outlet cross section. The relation for enthalpy i_2 calculation is as follows:

$$i_2 = i_0 - \eta_{tt_b} \cdot h_{is0-2} \tag{6}$$

From the knowledge of enthalpy i_2 and pressure p_2 it is possible to determine outlet wetness x_2 .

4 Conclusions

In the paper an outline is provided of measurement errors analysis when defining the flow path efficiency of BFPT turbine. Two relations are described for efficiency calculation. The first one is simpler, usable mainly in the situation when short blades are measured, without considerable changes in the temperature or pressure field along the blading height. This relation is also the only one that enables efficiency of individual stages in a multistage rotor to be defined.

The second relation is more complex. It defines efficiency using flow measurement through blading and using the torque. Flow measurement is done using condensate tanks or orifices. The common difference between defining efficiency from orifice or tank is 1 – 3 % of the efficiency value. The differences are caused mainly by a complicated assessment of supporting flows through the glands without a possibility to verify the tightness of the gland and vent systems.

In Doosan Škoda Power the value of efficiency defined using measurements from condensate tank flow is considered credible also because of the fact that it is possible to define the outlet wetness x_2 based on output measurement using water brake.

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