

**PERFORMANCE ANALYSIS BASED ON EXPERIMENTAL
DATA OF BACKPRESSURE STEAM TURBINE
FOR COGENERATION IN SATURATED STEAM
APPLICATIONS**

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A b s t r a c t

This paper aims to validate the performance capabilities of a Pressure Reducing Turbine (PRT) with respect to initial predictions based on analytic calculations. The designed equipment was installed in a beverage facility, located in Brazil. The validation procedure consists of analyzing the data collected in several periods of PRT's operation, accessed remotely via an online server. The analysis of empirical data identifies the behavior of two key variables: generated power and effective efficiency. However, the observed boundary conditions differed significantly from expected values, forcing the turbine to operate in off-design conditions. The turbine model was hence refined and used to predict the PRT's performance in such conditions. Results showed satisfactory accuracy for both power and efficiency predictions.

Keywords: steam turbine, cogeneration, energy efficiency, heat waste, saturated steam

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1. INTRODUCTION

Brazil has many industry sites, with a wide range of specialties, such as soy processing, automobilist, food and drinks, among others. Thus, the industrial sector is responsible for the highest electricity consumption in the Brazilian energy scenario, with 86×10^3 GWh/year of energy conservation potential [1]. According to official reports, Brazil has an electrical matrix of predominantly renewable origin, with emphasis on the water source that accounts for 64.9% of the domestic supply. Renewable sources account for 83.0% of the domestic supply of electricity in Brazil, which is the result of the sum of the amounts referring to domestic production plus imports, which are essentially of renewable origin [2]. Nonetheless, electricity consumption in the country presented an increase of 1.3% in 2019, represented by a 4.1% increase within the energy sector. In the same year, the total emissions associated with the energy sector reached a value of 419.9 Mt of CO₂-eq [2]. Brazil occupies the 46th position on the Environmental Performance Index on the world [3], meaning there is a great opportunity for the implementation of new technologies and policies for energy recovery and generation in the industrial sector. According to the International Energy Agency [4], the energy sector and industrial processes was responsible for the emission of 35 Gt of CO₂ in 2020, while there is a lack of new energy efficiency policies.

Considering that direct heating and process heat represent 77% of industry's overall energy consumption [1], the opportunity for new solutions is evident. Most of these plant sites have a steam generator designed to supply steam at a high-pressure level, followed by pressure reducing valves (PRV), which adjust the pressure according to process needs. This device operates by applying a throttling section to the steam flow, promoting a pressure drop. Since it is a fast process, there is no sufficient time to consider heat transfer – in that matter, the throttling can be modeled as an isenthalpic process. In that case, an entropy increase occurs [5], meaning the destruction of useful energy [6].

Thomazoni et al. [7] presented an alternative to the PRV, the Pressure Reducer Turbine (PRT). The PRT consists of a compact turbo generator that controls the pressure reduction according to the process needs. It has the added benefit of electricity generation, contributing to the whole plant's energy efficiency, alongside with cost reduction regarding energy consumption. The authors presented a detailed description of both the analytical and numerical approaches for the design of the PRT. The two approaches presented an absolute deviation of 5% in calculated power output amongst each other. In the numerical approach, simulations using the Finite Element Method (FEM) were conducted to further increase the design's reliability. It was predicted that the PRT, operating with a steam flow of 22 ton/h could save up to € 336.5 k in a year, generating 455.2 kW of power with a global efficiency of approximately 55.6%. The authors concluded

that the use of a PRT is complex due to the great variation of steam supply conditions – variables such as seasonality of consumption, operating time and production planning highly affect the steam generator operation.

Spirax Sarco [8] presented an alternative model of steam microturbine, with a capacity of 100 kW, providing savings on the scale of £ 75000 (€ 84000) when operating 8000 hours a year (based on 2014 figures). There is the understanding that the use of microturbines for controlling and reducing steam pressure for cogeneration is a brand-new technology, with recent applications on the energy market. Therefore, to the best of our knowledge, there is a limited offer of microturbines with the purpose of steam pressure modulation to this date.

The present paper is divided in four main parts, besides the current introduction and the conclusions. Section 2 begins with a contextualization of the advantages of the use of steam turbines for different cogeneration applications, followed by a presentation of the case study for this paper. The main characteristics of the turbine are given, as well as a general description of the installation layout. Section 3 briefly describes the instrumentation that was used for controlling and monitoring the PRT operation. The uncertainties associated with the measurements of key variables for efficiency calculations are also shown. Section 4 presents the methodology for calculating efficiency. Two methods are shown: the calculation of efficiency from the measured data collected from PRT operation, and the performance prediction by the use of a computational model. For the second case, some well-known methods for predicting steam turbine performance are referenced to support the discussion and the choice of the methodology applied in this paper. Lastly, the results are presented in section 5. The relation between power and efficiency from measured data is shown for a wide operation range. Next, a specific operation condition was observed in order to compare the model's efficiency prediction to the efficiency derived from real data. Then, the relation between pressure drop and generated power is shown for a range of measured mass flow values. In this case, only the average mass flow was used in the PRT model, to verify if this approximation has a significant impact on the model's accuracy.

2. THE CASE STUDY

Steam turbines have been used for power generation for the past 100 years, on account of its efficiency and cost advantages. Additionally, it has several levels of complexity regarding its construction, capacity and application [9]. This equipment is largely used for cogeneration applications in utility systems, for balancing the steam requirements, generating electricity or driving rotary equipment [10]. In order to achieve the best possible efficiency, the control of inlet steam flow must compensate for the process variations.

As previously described, the current global energy scenario is what motivates the search for solutions to reduce energy waste in industries. One attractive alternative to increase overall energy efficiency is the use of steam turbines for cogeneration in power plants. The cogeneration happens when there is a simultaneous production of electricity and heat. Generally, the heat production is achieved by extracting steam from one or more stages of the turbine. For these types of applications, the turbines are usually large scale and are composed of several stages. The heat extracted in cogeneration can be used in many different processes that require heat, but one important limitation is the temperature of extraction, since it depends on the place of extraction and size of the power plant. Several arrangements of steam turbines used for cogeneration with district heating are shown in [11]. Different extraction configurations are analyzed for the case of heat production from nuclear power plants, with a case study of a 1000 MW Light Water Reactor (LWR) power plant. Similarly, [12] presents a detailed analysis and simulations of a 250 MW power plant used for cogeneration with district heating, located in Copenhagen. For the different operation scenarios, a generic isentropic efficiency curve was adopted in steam turbines part-load calculations. The goal of reducing emissions leads to an increasing interest on the study of electrical and thermal power mix in nuclear power plants in many countries, in order to accommodate for the growing renewable intermittent energies. The work of [13] presents a study on the effects of different load rates in a Gen2 Pressurized Water Reactor under variable temperature grade cogeneration for district heating and water desalination. Results show that, under low temperature cogeneration, the power plant originally designed for full electricity production doesn't need to be revised nor adapted. High temperature cogeneration, however, shows a significant increase in irreversibility due to part-load operation of the cycle components. It was also shown that isenthalpic throttling valves are responsible for a major source of exergy destruction in the cycle.

The advantages of the PRT are also based on the concept of cogeneration, however, it is different from the conventional heat extraction from turbine groups in power plants previously described. In industries such as food and beverages, pulp and paper, and others, there are several processes that require steam in different pressure levels. The most common solution used by these companies is to generate steam at the highest pressure, and to use pressure reducing valves in order to adjust the pressure needed in each process. As said, these valves are responsible for a considerable share of energy waste in this sector. Alternatively, when they are replaced by PRTs, the controlled expansion of the steam provides the required pressure drop while passing through the turbine, which is coupled with an electricity generator. This is a very beneficial type of cogeneration, since the process heat production is not affected by the electricity generation, and the heat waste is reduced.

The PRT case study that will be discussed in this paper is a single-stage Rateau backpressure turbine whose inlet mass flow is controlled by a single valve. The turbine's interior is divided in three chambers, each one directing the steam flow towards a different set of nozzles. The valve is responsible for controlling the partial arc admission through a programmable logic controller (PLC), enabling or preventing steam flow through the chambers. Fig. 1 shows the operation modes.

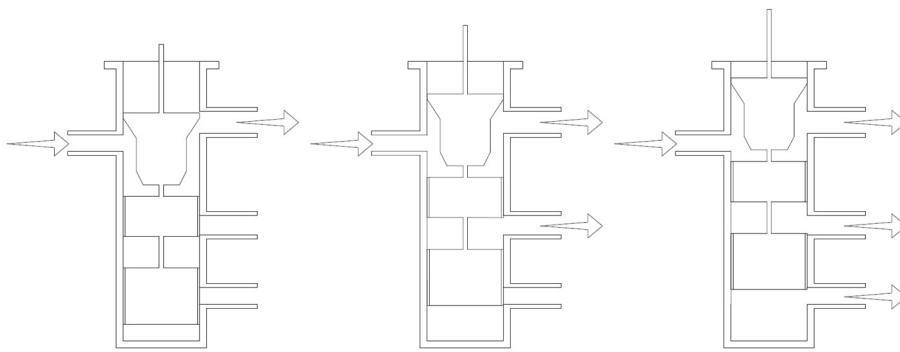


Fig. 1. Schematics of turbine's admission valve, controlling steam passage. The first, second and third admission levels are shown from left to right

The valve rod moves vertically and is composed of an upper solid plug and two hollow plugs, in the middle and lower sections. In the left scheme, the turbine is in startup mode, where only the upper passage is enabled and the other two are blocked by the plugs. In intermediate load mode, the rod is displaced upwards (middle scheme), unblocking both upper and middle passages while the lower passage is closed. The right-most scheme in Fig. 1 shows the valve position for the full arc operation mode, in which all three steam passages are enabled.

The governor regulates the level of partial admission, depending on load requirements, maintaining a constant shaft rotation speed. The percentage of the partial arc admission to each opening level is shown in Table 1.

Table 1. Partial arc admission characteristics. The values indicate the percentage of available nozzle area used per admission level

| | 1 st level | 2 nd level | 3 rd level |
|-----------------------|-----------------------|-----------------------|-----------------------|
| Partial arc admission | 45% | 80% | 100% (full arc) |

The turbine start-up occurs with 45% of the total nozzle area. With load increase, the turbine starts operating in second level admission, which is a significant change, since it almost doubles the passage area. With further increase in load, the

third admission level is used. Switching from the second to the third level represents a much lower change, as there is only a 25% increase in nozzle area. In order to transform shaft work in output electricity, the turbine is connected to a synchronous generator through a speed reducer. Furthermore, the turbine is equipped with a PLC control system. The PLC is desirable for steam turbines coupled with synchronous generators, because of its several features, such as remote communication, allowing real-time monitoring [14]. The controller acts to maintain the constant outlet pressure required by the factory's production, meaning that there is a low impact on steam conditions when replacing the PRV with a PRT.

The PRT described in this paper was installed in a beverage facility in Brazil, replacing three pressure reducing valves. Fig. 2 shows a schematic of the plant's layout.

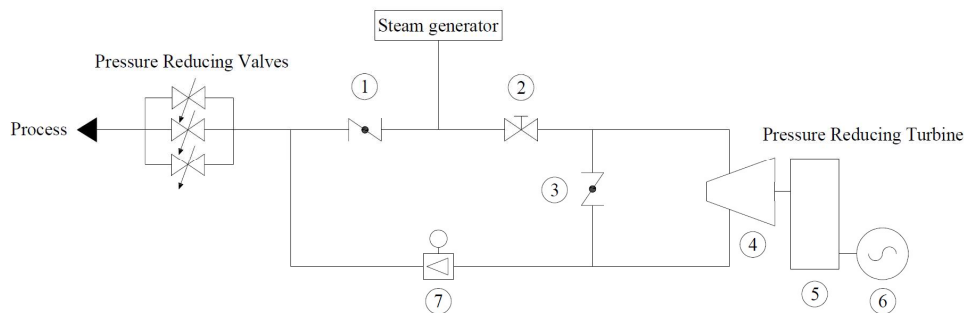


Fig. 2. Plant layout schematic. Numbers 1 and 3 are butterfly valves. Number 2 is a manual slide valve. Numbers 4, 5 and 6 are the turbine, speed reducer and synchronous generator, respectively. Number 7 indicates the vortex flow meter

A group of valves, allow the reconfiguration of the steam path, permitting easy access for maintenance interventions. In standard operation, the flow should be directed entirely to the PRT, and no pressure reduction should take place in the PRVs. An exception exists when great changes in the flow parameters occur. In this case, the PRVs and the PRT can be simultaneously used to prevent turbine damaging.

Real turbine operation is commonly far from designed conditions, deviating from the rated conditions, due to process parameter adjustments, and utility system variations [10]. For this reason, a series of preventive actions were programmed to take place to avoid shutdown in the event of great fluctuations in operating conditions. An important protective measure is the opening of bypass valve 3 when steam consumption in the factory is high. In this case, a portion of the flow is diverted directly to the turbine outlet, in order to stabilize the pressure.

3. INSTRUMENTATION

After the design process, once the equipment is installed, it must be submitted to a validation procedure. Thus, acquiring detailed measurement data from the operating plant to be used in the equipment validation has the utmost value. Those measurements require effort and cooperation between the turbine manufacturer and plant owner [15]. Laboratory trials and plant feedback are appreciated but they are not enough to predict the equipment's impact on the plant's operation, because even the best testing facility for steam turbines does not reach full scale [14].

In order to maintain precise surveillance of the turbine performance and operation conditions, several sensors are used to provide the required measurements. This section aims to explain the instrumentation of the PRT, as well as the associated instruments accuracy.

The turbine control system comprehends two PLCs. The first is mainly used for the control of the turbine, while the second is responsible for power export. The turbine's governor is controlled by a Proportional Integral Derivative controller (PID), integrated into the first PLC. Moreover, the control system is connected to an online server that allows remote data extraction. The exported data consists of steam flow, temperature and pressures, rotation speed, governor output, power export, oil pressure, bearing vibration and temperatures.

Adjacent to the PRT, a Human Machine Interface (HMI) provides real-time information to the operator, assisting in start-up and shutdown procedures. Turbine and oil system conditions are some of the information displayed. Fig. 3 displays some of the turbine's key components.

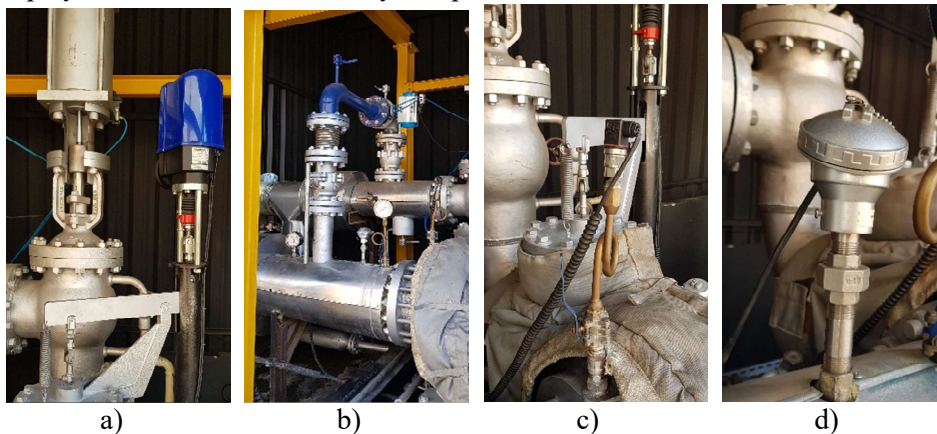


Fig. 3(a) shows the 25 kN linear actuator, which moves the admission valve through a lever with position feedback. Fig. 3(b) illustrates the bypass system, where the pneumatic butterfly valve is also seen in Fig. 2 (number 3). Fig. 3(c) and Fig. 3(d) show the pressure and temperature sensors, respectively

The measurements of pressure, temperature and mass flow are used for computing efficiency. Therefore, only the uncertainty relative to these measurements will be discussed. The information was obtained from the instrumentation's datasheets. Moreover, all these sensors are connected to the PLC, through a 4-20 mA signal. Table 2 shows the associated uncertainties.

Table 2. Uncertainties related to performance evaluation

| Sensor Type | Uncertainty (%) |
|-------------------------|-----------------|
| Pressure transmitter | ±0.50 |
| Temperature transmitter | ±0.55 |
| Mass flow meter | ±1.00 |

The vortex flow meter is connected to the plant's central PLC since it was integrated before the installation of the PRT. The temperature and pressure monitoring are done by a PT-100 sensor and a pressure transmitter.

The power generation is measured by the export system. Coupled to the synchronous generator, an Automatic Voltage Regulator (AVR) is responsible for controlling the voltage, maintaining 380 V for the power export. The PLC reads the voltage and current on the generator terminals and calculates the generated power.

4. PRT EFFICIENCY CALCULATION PROCEDURE

There are several ways to calculate the efficiency of a turbine, varying on the level of detail and precision that is needed. The chosen approach can also change depending if it is done during early design stages or if it is used for diagnosis purposes. The type of operation (full-load or part-load) and the size of the turbine (nominal power and number of stages) are other relevant aspects that can influence the way the performance is calculated. Hence, the first step of the performance analysis is to determine which model is the most suited for the given case.

4.1. Performance from measured data

The effective efficiency considered in this paper is calculated with two methods. The first one consists in calculating the effective efficiency of the turbine-generator from the measured data points obtained from PRT real operation. In this case, Equation (4.1) can be used since the inlet and outlet states, as well as generated power and mass flow, are known. .

$$\eta_{measured} = \frac{h_i - h_o}{h_i - h_{s,o}} = \frac{\dot{W}}{\dot{m}(h_i - h_{s,o})} \quad (4.1)$$

Where:

| | |
|-------------------|---|
| $\eta_{measured}$ | effective efficiency of the PRT, calculated from measured data, |
| \dot{W} | generated electric power, kW, |
| \dot{m} | mass flow through the turbine, kg/s, |
| h_i | specific enthalpy at turbine's inlet, kJ/kg, |
| h_o | specific enthalpy at turbine's outlet, kJ/kg, |
| $h_{(s,o)}$ | specific enthalpy at turbine's outlet considering an isentropic expansion, kJ/kg. |

In this first method, the efficiency calculated from measured data is denoted by $\eta_{measured}$. In the middle term, efficiency is given by the ratio between real and isentropic enthalpy drops. However, since there is no temperature sensor in the outlet of the turbine, efficiency could not be calculated using the middle term in Equation (4.1), because h_o can't be determined. Alternatively, the right-most term was used in an equivalent manner, since power and mass flow are measured. In this case, \dot{W} is obtained through the generator terminals and the measurements collected with the flowmeter are used in \dot{m} . The pressure measurements at the turbine's inlet are used as input in Engineering Equation Solver (EES), which has built-in thermodynamic tables. These tables are used to obtain h_i and the specific entropy at inlet s_i for saturated steam. The plant's steam generator is set to deliver dry steam at the turbine's inlet. To simplify the problem, the assumption of fully saturated steam is taken, instead of considering the minor temperature variations due to steam generator real operation. Since the steam is considered as fully saturated (dry steam) at the inlet, h_i and s_i are a function of the inlet pressure P_i , as shown in (4.2).

$$h_i = f(P_i) \quad (4.2)$$

$$s_i = f(P_i)$$

Finally, the isentropic enthalpy drop in the denominator of Equation (4.1) must be determined to calculate $\eta_{measured}$. In an isentropic expansion, the specific entropy at the outlet $s_{(s,o)}$ is equal to s_i . The outlet pressure measurements P_o are then used with $s_{(s,o)}$ in order to obtain the tabled values of $h_{(s,o)}$, as shown in (4.3).

$$s_i = s_{s,o} \quad (4.3)$$

$$h_{s,o} = f(P_o, s_{s,o})$$

On-sight measurements are highly dependent on sensors calibration and correct installation. Oftentimes, incorrect measurements occur due to some external influence or interference. In this procedure, the data must be filtered to remove any inconsistent measurements or outliers that would lead to incoherent efficiency results.

4.2. PRT Model

A second method is needed for estimating the performance of the turbine-generator, given the known boundary conditions. In this case, a mathematical model is required to accurately predict the PRT efficiency at the observed conditions.

A classical approach for estimating turbine-generator performance in full load or partial load conditions is presented by Spencer, Cotton and Cannon [16]. In this work, a methodology is developed for predicting the efficiency of each individual stage or a group of stages. The influence of exhaust losses, packing and valve steam leakage flows, mechanical losses, and generator losses are considered. The calculation procedure is based on the analysis of expansion lines, and the use of a set of curves and tables for correcting the given base efficiency, to account for each given loss. The work of [17] presents the development of a computational tool for calculating efficiency drop in steam turbines due to cogeneration. The study was based on the methodology of Spencer, Cotton and Cannon, which was modified to account for intermediate extractions. However, this methodology is validated for turbines larger than 16.5 MW, which is bigger than the case studied in the present paper. For this reason, this approach could not be used for calculating PRT efficiency.

The Ellipse Law is another well-known method for predicting the behaviour of turbines in off-design conditions. In this approach, simple algebraic equations are used to model the expansion in a group of stages as an expansion in a single nozzle, a simplification known as nozzle analogy. Variations for choked flow and low number of stages (down to single stage) were also developed, which makes this method applicable for a broad range of situations. Cooke [18] presents the basis of the method for controlled and uncontrolled expansions and a comparison to real turbine operation data in different scenarios. The Stodola Ellipse Law is used in [19] to model a single-pressure condensing steam turbine without reheat operating in a 150 MW combined-cycle power plant. The study focused on the utilization of the combined-cycle to compensate the variable electricity production of renewable energy sources. In such conditions, the components of the cycle are submitted to transient and off-design behaviors, which were modelled by the Stodola Ellipse Law, in the case of the steam turbine. Unlike in [16], Cooke's approach doesn't seem to be restrictive over smaller turbines, meaning it could be used for estimating PRT efficiency. Nevertheless,

since [18] doesn't account for the several dynamic turbine losses that are present during part-load operation, this methodology was also not applied in the subsequent analysis.

Schegliáiev [20] is also well-known for his comprehensive work on steam turbine design. He presents detailed information and calculation procedures for several turbine sizes and applications. Regarding the estimation of a real non-isentropic flow through the turbine blades, three main energy loss coefficients are considered in his work, relative to: the nozzle blades, the rotor blades, and the velocity leaving the stage. The efficiency obtained when considering these coefficients is denoted as relative efficiency of the blades. Furthermore, in a turbine stage, there are generally four other complementary losses that are considered: ventilation and friction effects, partial admission, steam leakage, and steam wetness. The efficiency that considers all these effects is called relative internal efficiency, which is also the final isentropic efficiency of the turbine, for the case of the PRT. The work of [21] presents a review of some known equations for performance prediction of steam turbines, including some of the equations presented in [20]. The calculations are compared with existing turbine data in order to propose a modified version of Troyanovskii's equations to better fit the observed data. This was done because the original equations provide only an upper estimate for the turbine performance.

In [22], a comparison is made between different load control modes for a turbine in off-design operation. For the initial calculations, the pressure relation found in [20] and the Stodola's model [18] were used. The results showed high accordance between the two approaches for a four-group turbine with reheating and two extractions. Since the two approaches presented similar accuracy, the authors decided to apply Schegliáiev's model for the load control calculations, since it achieved convergence more easily when compared to Stodola's model.

The model used for the performance prediction is an in-house extension of the PRT model initially presented by [7] and is based on the Schegliáiev's extended methodology described in [20]. The main addition of the present work to the model used by [7] is the parametrized calculation of off-design performance of the PRT. This approach was chosen since all dimensional characteristics of the turbine are known, meaning that a detailed analysis of turbine losses could be done. In this case, the effective efficiency is denoted by η_{model} . With the established boundary conditions and dimensional characteristics, a series of primary energy losses related to geometric characteristics are considered to determine the flow velocity coefficients. Thereafter the turbine's velocity triangles are constructed, and several secondary flow-related losses are considered. The Equation used in this paper is shown in (4.4).

$$\eta_{rp} = 1 - \xi_f - \xi_r - \xi_{vs} \quad (4.4)$$

$$\eta_{ir} = \eta_{rp} - \xi_a - \xi_p - \xi_l - \xi_h$$

$$\eta_{model} = \eta_m \eta_g \eta_{ir}$$

Where:

| | |
|----------------|--|
| η_{model} | estimated effective efficiency of the PRT, calculated with the turbine model, |
| η_{rp} | relative efficiency of the blades, which considers geometry-related energy losses, |
| ξ_f | energy loss coefficient of the nozzle, |
| ξ_r | energy loss coefficient of the rotor blades, |
| ξ_{vs} | energy loss coefficient relative to the velocity leaving the blades, |
| η_{ir} | isentropic efficiency, which also accounts for flow-related energy losses, |
| ξ_a | energy loss coefficient relative to ventilation and friction effects, |
| ξ_p | energy loss coefficient relative to partial admission, |
| ξ_l | energy loss coefficient relative to steam leakage, |
| ξ_h | energy loss coefficient relative to steam wetness, |
| η_m | mechanical efficiency, |
| η_g | synchronous generator efficiency provided by the manufacturer. |

The losses are estimated by the use of empirical equations and by correcting the results obtained from generalized experiments found in the literature. An example of some generalized results can be found in [20]. The consideration of the steam's energy losses also leads to estimating the thermodynamic state at the outlet, and consequently the steam quality, after a non-isentropic expansion.

Since the calculation process can be challenging and deals with several non-linearities, simplifying assumptions are taken to facilitate the procedure. The main assumptions considered in this paper are the following: steam is saturated at inlet (steam quality equals to 1), flow parameters are considered in mid-height of the blades and do not vary radially, velocity entering the nozzles are estimated from average velocity entering the turbine, the average velocity vector is considered to be orthogonal to nozzle inlet, velocity deviation due to shockwave effects are not considered.

It is worth mentioning that performance prediction of turbines is still a field under development, with numerous new approaches being proposed. More recent turbine models make use of neural networks for performance prediction and monitoring of gas and steam turbines [24, 25, 26]. However, these methods were not considered in the present work.

5. PERFORMANCE DATA

The turbine that will be discussed in this paper was designed for an average mass flow rate of 22 t/h, ranging from a minimum of 14 t/h to a maximum of 29 t/h. In the designed conditions, saturated steam enters the PRT with an average pressure of 16.5 bar(g) and is returned to the process at 8.4 bar(g).

However, the factory's real operation diverged from the expected values and showed high fluctuations, leading to PRT functioning in off-design conditions. Initially, more than 96000 data points were collected, consisting of 20 days of PRT's operation. Table 3 shows the difference between real and designed operating conditions, disregarding start-up, shutdown and other transient events.

Table 3. Turbine's boundary conditions

| Variable | Units | Design Conditions | Measured Data (average) |
|-----------------|--------|-------------------|-------------------------|
| Mass flow | t/h | 22 | 16.7 |
| Inlet pressure | bar(g) | 16.5 | 11.9 |
| Outlet pressure | bar(g) | 8.4 | 7.4 |

The highest variation is observed in mass flow measurements, in which the minimum and maximum values are 2.1 t/h and 27.6 t/h, with an average of 16.7 t/h. Although inlet pressure measurements show greater stability, their average values are 28.3% lower than the designed values, which also lead to an 11.9% decrease in outlet pressure. However, the measured inlet temperature showed a maximum deviation of $\pm 2\%$ from the saturation temperature for each measured pressure, including the associated uncertainties. This demonstrates that the steam entering the turbine can still be considered as saturated, since it varies slightly between wet and supersaturated conditions.

The discrepancy in Table 3, imposed a challenge to this paper's goal, that is the performance validation relative to initial predictions. Since the equipment does not operate under designed conditions, the turbine model was refined to predict the off-design performance. The model was created using EES software and was based on the analytical expressions and empirical equations found in the traditional literature for the design of steam turbines [20, 23, 6].

When the outlet pressure is lower than 7 bar(g), valve 3 in Fig. 2 opens to maintain the outlet pressure, splitting the inlet flow. Since the flowmeter is positioned downstream of the bypass, it is unable to capture how much of the flow is passing through the turbine in this case. The flow measurement data relative to this situation were hence discarded. Unfortunately, that prevents the calculation of η_{measured} for such data, which contains the highest values of generated electrical

power, reaching a maximum of 550 kW. Fig. 4 shows the relation between power generated and η_{measured} for several operating conditions of the PRT.

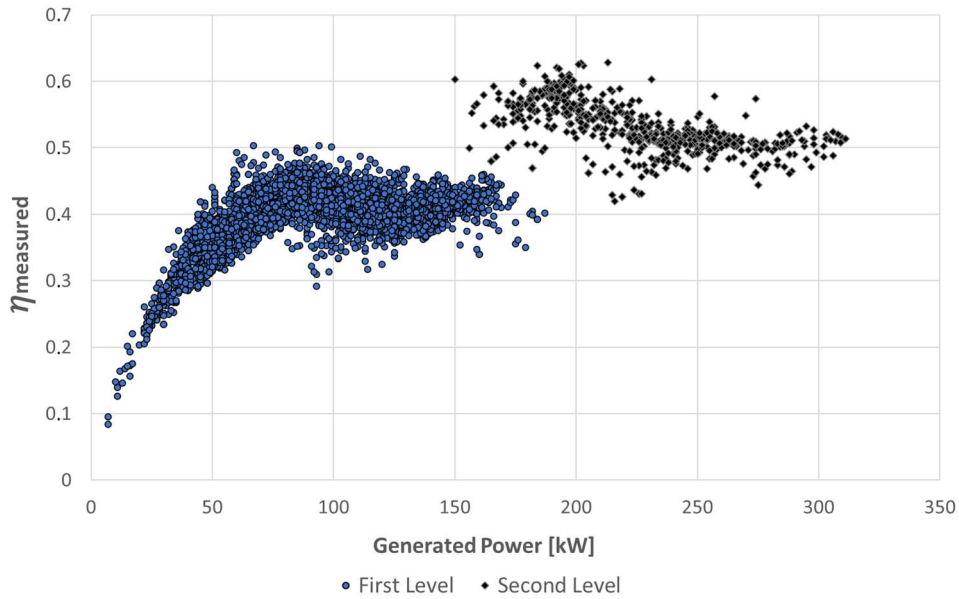


Fig. 4. Electrical power generation versus effective efficiency (η_{measured}) for steady state operation of the PRT. Inlet/outlet pressure ratio ranging from 1.2 to 1.8. Mass flow ranging from 7 to 23 t/h. First and second levels of nozzle governing are shown

In Fig. 4, it is possible to distinguish between two levels of nozzle arc opening. For low power generation, the PRT has lower levels of η_{measured} , which rapidly increases and stabilizes around 40%. If there is an increase in load, the second admission chamber is opened, causing a shift in the power-efficiency curve, achieving values around 50%. This significant shift in the curve is explained by the almost 80% nozzle area gain when passing from the first to the second level. The third nozzle arc opening level was not represented in Fig. 4, because most of the full arc operation occurs when the bypass in Fig. 2 is opened. However, the lower number of nozzles in the third chamber would result in a minor shift in the curve, when passing from second to third admission level.

In order to analyze full arc admission and compare the model's prediction to the measured data, a particular case with the following characteristics was studied: Inlet pressure $P_{\text{in}}=11.8$ bar(g) and outlet pressure $P_{\text{out}}=7.4$ bar(g). Only the full arc mode in steady state was considered, minimizing the impact of parameters fluctuation due to transient states and process variations. The influence of mass

flow on efficiency and power is analyzed in Fig. 5, for the described conditions. In this case, both η_{measured} and η_{model} are shown.

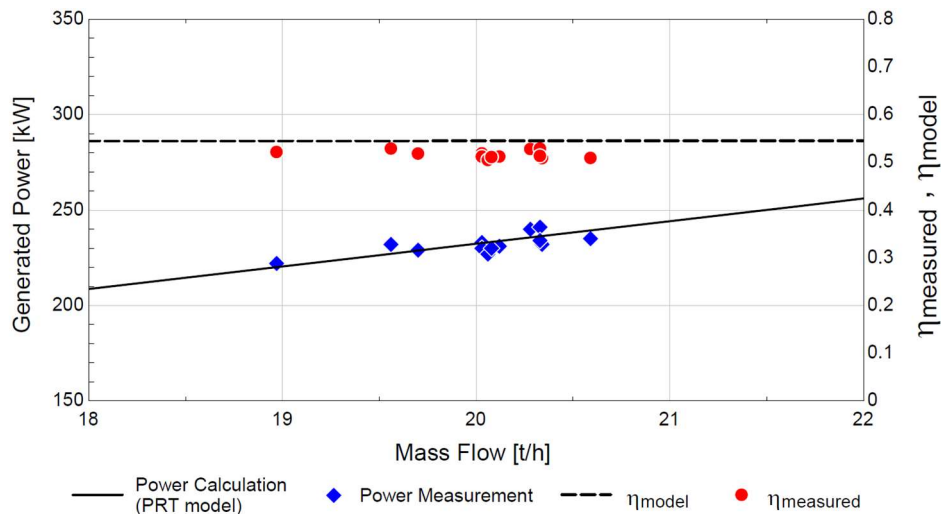


Fig. 5. Comparison between measured data (points) and model prediction (line) with $P_{\text{in}}=11.8$ bar(g) and $P_{\text{out}}=7.4$ bar(g). Both generated power and efficiency are shown. The efficiency was computed using the equations for η_{measured} and η_{model} . Turbine operating with full arc admission

For the selected conditions, the PRT model was able to capture the behavior of generated power, with errors below 3%. The almost linear relation between mass flow and power, known as is Willans Line, is often studied in turbomachinery topics. On this matter, [10] presented a model based on the Willans Line used to estimate steam turbine performance for full-load and part-load conditions.

Fig. 5 also reveals an almost constant value of efficiency. The turbine model presented slightly overestimated values of η_{model} , with errors below 10%. Considering there is no change in the nozzle arc opening level and the selected power range, a small change in efficiency is indeed expected. This can be observed in Fig. 4, where η_{measured} tends to stabilize for loads above 100 kW, within the same nozzle arc opening level.

To further analyze PRT performance, Fig. 6 presents the relation between the pressure drop in the turbine and generated power. For this analysis, the third arc opening level is not considered.

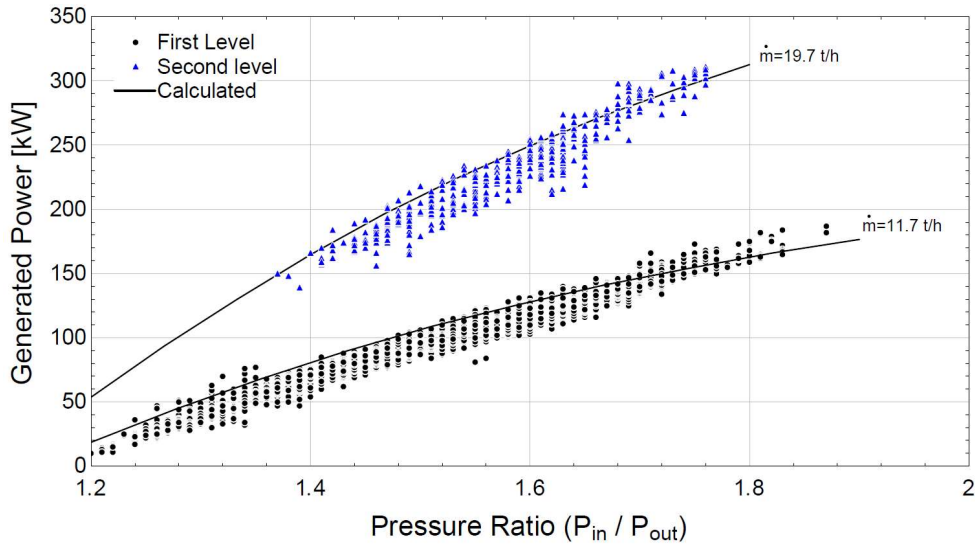


Fig. 6. Comparison between measured data (points) and model prediction (line). Inlet pressure ranges from 10.9 to 13.3 bar(g). In measured data, mass flow ranges from 7 to 23 t/h

The mass flow varies from 7 to 23 t/h in measured data, with an average of 11.7 t/h and 19.7 t/h for the first and second admission levels. These average values were used as constants in the PRT model. Considering this approximation, the calculated response presents a close resemblance to the real behavior, for both arc opening levels. The model's precision for the present case is approximately $x_{-17\%}^{+5\%}$, where x is the measured power.

6. CONCLUSIONS

The goal of this paper is to validate the performance of the pressure reducing turbine installed in an industrial plant in Brazil, with respect to initial performance predictions. However, the measured steam pressure in the beverage factory diverged considerably from the expected values. Moreover, the high fluctuation in mass flow also contributed to the operation of the PRT in off-design conditions. For this reason, the PRT model was used to estimate the performance in part-load operation, which was then compared to the on-site measurements.

The effect of the turbine's partial admission can be observed in the relation between generated power and η_{measured} . However, due to the flow meter's location, most of the mass flow measurements with full arc operation could not be considered. Therefore, this analysis focused mainly on the first two admission

levels. When only the first set of nozzles are active, the PRT can generate up to 175 kW with η_{measured} between 35 to 45%. When the second admission level is active, power is generated between 175 and 300 kW, with η_{measured} varying from 45 to 60%. The measurements data also shows that the PRT can generate a maximum of 550 kW, with full arc admission.

The PRT model presented high accuracy when estimating power and efficiency, for a fixed value of pressure drop in the turbine. The maximum errors obtained were 3% and 10% for power and η_{model} , respectively.

The relation between generated power and pressure drop in the turbine was successfully estimated by the PRT model. Even though the average values of mass flow were used in the calculations, the result showed a maximum error of approximately 17%.

Generally, the performance of the pressure reducing turbine in off design conditions was correctly estimated by the applied model. In most conditions, a slight performance overestimation was detected. For future studies, mass flow data should be acquired over the complete operating range in order to fully capture the PRT's performance. Moreover, the loss coefficients of the turbine can be adjusted to better fit the measured data, increasing its estimation accuracy.

ADDITIONAL INFORMATION

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