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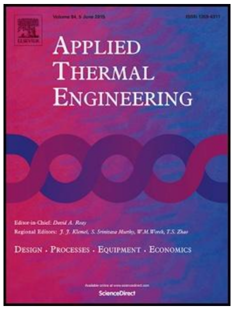

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Thermal balance of wet-steam turbines in nuclear power plants. A case study

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Highlights

- Steam turbine efficiency and power are estimated at a particular NPP.
- The model estimates the cycle parameters from various performance tests.
- Areas where the results of the simulation may be improved are identified.

Abstract
Conventional methodology is applied in this study to estimate the efficiency of nuclear power generation and the electrical power that is generated at a nuclear plant built in the nineteen-sixties. A simulation of the whole power plant is conducted with this methodology. The characteristic parameters from the study are compared with those estimated by the manufacturer for its on-design operation mode. Furthermore, the model calculates the anticipated pressures, temperatures, mass flows and electrical power on the basis of the data from various performance tests. These estimated parameters are then compared with real values measured in the plant. The objective of the present paper is to study whether the conventional approach is acceptable for the simulation of the parameters of the power station, and to identify areas where the results of the simulation may be improved, in order to minimize the deviations between the simulated parameters and the actual measurements.
Keywords: Nuclear Power Plant, Wet-Steam Turbine, Balance of Plant

1. Introduction

The first technical papers on the thermodynamic behavior of steam turbines were published at the beginning of the 20th c. In this period, the publication of Aurel Stodola [1]**Error! Reference source not found.**, a professor at the Polytechnic University of Zurich, the first edition of which dates back to 1903, is remarkable, because of its scientific rigor, depth, and validation through testing. The book remains a basic reference work on the study of steam turbines.

At the start of his paper, professor Stodola proves the validity of considering that steam (also in a wet condition) acts as a polytropic gas, using the equation $pv\gamma = C$, in order to simplify the relationship between pressure and specific volume in expansion processes at constant entropy. From this hypothesis, a complete development of a simplified formulation can be introduced, including what was later known as the Stodola Ellipse, which was studied by Cook in the nineteen-eighties
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Firstly, engineers focused on the design of efficient steam turbines, thus developing the necessary formulation to define the optimum profiles of blades and nozzles, in order to use the maximum available amount of thermal energy in the working fluid: the steam. On this point, the simplicity of an unpublished study by Soderberg in 1949, published at a later date by Horlock [3], should be highlighted. Other significant contributions in the one-dimensional study of steam flows through turbines are those by Ainley and Mathieson [4], improved by Dunham and Came [5], and the more recent and complex study by Wilson [6]. In all cases, different methods were proposed for the estimation of the energy losses through the blades and nozzles, thus introducing several correction factors as a function of the geometry of these components.

Different one-dimensional, as well as three-dimensional methods for the calculation of the steam flow through the blades and nozzles in steam turbines can be found in the book by Dixon [7].

Having designed the different components of the steam path, the next step is to calculate the operation of the whole steam turbine in the process which it will undergo. One of the first papers in this regard was published by Smith in 1938 [8], where the so-called "'reheat factor"' was first introduced: with this factor, the expansion of the steam according to the efficiency of in-design conditions could be corrected by using the operational parameters of the process.

In the case of condensing steam turbines, one of the first calculation problems was to estimate the thermodynamic properties at the last stage that involves wet-steam. Different publications were presented and later on incorporated in a Performance Test Code (PTC) developed by the American Society of Mechanical Engineers (ASME), edited for the first time in 1915: ASME PTC 6. The basis of the method can be found in the revised edition published in 1964, which includes the studies developed by Cotton and Westcott [9] (where the calculations were highly simplified), and by Spencer, Cotton and Cannon [10] (still useful today).

The first nuclear power plant in the world was connected to the grid (Óbninsk, with 5 MWe of electrical power generated with an efficiency of 17%) in June of 1954. Three years later, the United Kingdom built two more. From 1960 to 1988, the number of nuclear power plants increased from 16 to 416. This is a good explanation of why so many papers on the expansion of wet-steam through steam turbines were published in that period [11]. The main problem was how to calculate steam enthalpy in the flow path, as only the properly measured parameters were pressure and temperature [12].

In their design engineering, the main goal was to study the effect of wet-steam, in order to increase the efficiency of these engines [13], and to define new procedures in order to calculate any deviations after installation of the steam turbines in a power plant. The same calculations should be useful to monitor the operation of the unit, by comparing the actual parameters from instrumented readings with the estimated values. However, in this case, the actual mass flow of the steam extracted from the turbines to the feedwater heater has to be known. Due to the lack of instrumentation, the calculation of this flow was proposed using radioactive [14] or chemical tracers [15] during periodic efficiency tests. New papers were then published on the estimation of the power generated and efficiency of wet-steam turbines in nuclear power plants [16].

The ASME PTC 6 has since been revised with the contributions of published papers on fossil and nuclear power plants, up until its most recent edition [17].

Recently, new models have been proposed for the simulation of fossil power plants, but they often include simplifications of the real process. For instance, Colonna and Van Putten [18, 19] presented a model where the parameters in the heat exchangers are corrected as a function of the operational conditions using the Dittus-Boelter and Colburn correlations. However, their study examined the economizer and the superheater of the boiler and not the wet-steam turbines, as in this study.

Chailbakhsh and Ghaffari [20] published a non-linear mathematical model and non-linear functions, which characterize the transient dynamics of steam turbine subsections and estimate the thermodynamic properties of the steam, respectively.

In a paper by Medina Flores and Picón Nuñez [21], a relationship between the temperature and enthalpy of the steam and the extraction pressure of steam turbines with multiple extractions was used to calculate the isentropic efficiency.

The problem of calculating the thermodynamic properties of wet-steam in nuclear power plants is highlighted in the paper by Heo and Chang [22]. Here, a complex model using the PEPSE code is developed.

Sunde and Berg [23] published a paper in which the thermal balance of the steam turbines of a boiling water reactor power plant was modeled using the Cycle-TEMPO software for diagnostic purposes.

Teyssedou et al. [24] provided a complete simulation of the Glentilly-2 nuclear power plant. Temperatures in the feedwater heaters train were calculated by using a simplified procedure derived from the Bell-Delaware method [25]. However, the deviation between the parameters obtained in the simulation and the consigned values were too high, although the formulation included the same operating conditions as the validation process. The reason was perhaps due to their having considered the pressure ratio between turbine sections as a constant. Furthermore, the moisture-content effectiveness in the steam turbine extractions and the mass flow used for the steam seals were not taken into account.

Mochizuki and Tsukamoto [26] published a thermal-hydraulic network analysis of the turbine and feedwater system of the Fugen reactor using the code NETFLOW++. And Punnonen et al. [27] published the modeling of full-speed turbines in the secondary loop of a pressure water reactor, demonstrating that the traditional simplified models can be used to predict total exit losses and plant performance.

	Nomenclature
A	Area, $mm2$.
C	Constant.
C_q	Flow coefficient in the Stodola equation.
c_t	First stage wheel speed, m/s.
EL	Exhaust loss, kJ/kg.
F_M	Correction factor for the average moisture (HP turbines) or for the initial moisture (LP turbines).
F_{speed}	Correction factor for the first stage wheel speed.
F_v	Correction factor for volume flow.
h	Specific enthalpy, kJ/kg.
k_p	Coefficient in the Martin's Equation.

M	Moisture content, %.
N_p	Number of throttlings through the labyrinth.
$\dot{m}$	Mass flow, kg/s.
$\dot{m}_v$	Volumetric flow, $m3/s$.
p	Pressure (absolute), MPa.
s	Specific entropy, kJ/kgK.
v	Specific volume, $m3/kg$.
$\dot{W}$	Turbine thermal power, MW.
x	Quality.
γ	Isentropic exponent.
ε_s	Stage moisture removal effectiveness.
η	Efficiency.
η_i	Turbine base efficiency.
η_{i2}	Corrected turbine base efficiency.
	Subscripts
c	Cold fluid (feedwater).
d	Fluid from the previous heaters or other processes.
e	Steam extracted.
h	Hot fluid (steam).
HP	High-pressure turbine.
HTR	Feedwater heater.
i	Inlet section / Any particular turbine section after the extraction point.
i'	Any particular turbine section before the extraction point.
LP	Low-pressure turbine.
o	Outlet section.
p	Packing.
$-l$	Saturated liquid.
$-v$	Wet-steam (vapor).
2	Intermediate section ($c2$ or $h2$).
$;d$	On-design value.

2. Traditional approach to the simulation of nuclear power plants

2.1. Description of the power plant under study

The Santa María de Garoña Nuclear Power Plant (SMGNPP) is a 466 MWe power station located in the north of Spain that was first connected to the grid in 1971. Designed by General Electric, the steam produced in a boiling water reactor (BWR) generates 1,381 MWt of thermal power that is supplied to the steam turbines with a moisture content of 0.28%. An initial power of 460 MWe was guaranteed by the designer, although, after some improvements and changes in the operating conditions, the consigned gross electric power was raised to 6 MWe.

A simplified diagram of the water and steam flows is shown in Fig. (1). Numbers in the diagram indicate the thermodynamic state of the steam for both the following equations and the model.

Steam leaves the BWR through four pipes connected to the high-pressure (HP) turbine. A by-pass valve is provided, used at low loads to regulate the reactor pressure (and as an overpressure protection device), piping the steam directly to the condenser.

Then, the steam passes through the stop and control valves, one per pipe. The first prevents the flow of steam if necessary, and the control valves regulate the mass flow, in order to achieve an approximately constant pressure in the reactor. The steam in the gap between the valve-stem and the valve-bushing is piped to the feedwater heaters (HTR's) 11A and 11B and to the steam-seal regulator (SSR).

Immediately before these valves, a pipe sends around 2,268 kg/h (5,000 lbm/h) of steam to the steam-jet air ejectors (SJAE).

The steam expands in the HP, a single-flow action turbine with ten stages. The steam in the gap between the rotor and the shell on the inlet side is firstly piped to the HTR's 11A and 11B, then to the SSR, and finally, the mixture of steam and atmospheric air, is piped to the steam-packing exhauster (SPE). At the discharge side, the steam in the gap between the rotor and the shell is firstly piped to the SSR and, finally, the mixture of steam and atmospheric air is piped to the SPE.

After expansion in the HP turbine (10 stages), the steam passes along four pipes, two of which run into low-pressure (LP) turbine "A" and the other two into LP turbine "B".

Between the HP turbine and the LP turbines, the pipes that run into LP turbine "A" are connected to HTR 12A and the pipes that run into LP turbine "B" are connected to heater 12B. After these connections, the pipes run into four Moisture Separators (MS), where the steam quality is improved by moisture removal, and then on to HTR 11A (from one of the MS in train "A" and one of the MS in train "B") and HTR 11B (in the same way).

Then, the steam is expanded in LP turbines "A" and "B". Each one has two opposite flows that go to the condenser (10 stages per flow). The following extractions are in place during the steam expansion through the LP turbines: in stage 12 to HTR's 11A/B, in stage 14 to HTR's 9A/B, in stage 16 to HTR's 8A/B and in stage 18 to HTR's 7A/B (located in the condenser). Several drains are placed in stages 15, 17 and 19, in order to improve steam quality.

The seal-steam piped from the HP turbine and the turbine valves is used for the LP turbines seal. The steam is injected into the clearance between the shaft and the shell on the closest side to the LP turbine discharge side. An amount of this steam goes to the condenser, and the rest, mixed with atmospheric air, is piped in the other direction to the SPE. No air therefore passes into the main condenser.

The condensed and subcooled steam drained from HTR's 12A/B goes to HTR's 11A/B, etc. The steam drained from HTR's 7A/B goes directly to the main condenser.

2.2. Turbine stages and feedwater heaters pressures

The following equation, derived from the Stodola's Ellipse Law, has traditionally been used to estimate the pressure in the steam turbine extraction points, under the hypothesis that the pressure ratio between turbine sections remains constant:

$$\dot{m}_{(i)'} = c_{q(i)} \sqrt{\frac{p_{(i)'}}{v_{(i)'}}} \qquad (1)$$

where, $(i)'$ refers to a particular stage just before the extraction point. Obviously, $p_{(i)'} = p_{(i)}$. The mass flow that goes to the next stage will be $\dot{m}_{(i)} = \dot{m}_{(i)'} - \dot{m}_{(i)e}$, where $\dot{m}_{(i)e}$ is the mass flow extracted to the feedwater heater.

The pressure drop between the turbine stage and the heater, Δp_{HTR}, should be taken into account to calculate the pressure in the feedwater heater shell. In the power plant under study, this value is a 95% for HTR12A/B and 92% for the others.

2.3. HP and LP turbine efficiency and thermal power

Steam turbine efficiency was calculated using the Performance Test Code (PTC) developed by the American Society of Mechanical Engineers (ASME): ASME PTC 6. This code is divided into three parts: ASME PTC 6 [17]**Error! Reference source not found.**, ASME PTC 6A [28] and ASME PTC 6S [29].

The paper by F.G. Baily, J.A. Booth, K.C. Cotton and E.H. Miller [16] is the standard reference for the calculations implemented in those codes, in relation to nuclear steam turbines. The excellent book by K.C. Cotton [30] also includes the applicable calculations for nuclear power plants. The necessary diagrams implemented in the model can be found in these papers.

In the HP turbine, efficiency is calculated from the so-called base efficiency, corrected by three factors affected by the first stage wheel speed, the average moisture content and the exhaust loss.

The base efficiency, $\eta_{i_{HP}}$, is a function of the volumetric mass flow at the inlet section, $\dot{m}_{v_{1;d}} = \dot{m}_{1;d} v_{1;d}$, and the bowl to exhaust pressure ratio, $p_{1;d} / p_{4;d}$, in design conditions.

The correction for the first stage wheel speed (governing stage), F_{speed}, is a function of the first stage pitch wheel speed and the bowl to exhaust pressure ratio with the valves wide open. The first stage wheel speed is calculated with the following equation:

$$c_t = \frac{pitch \cdot \pi \cdot rpm}{60} \qquad (2)$$

In the power plant under study, $pitch = 1.8542$ m and $rpm = 1,500$.

The correction for the average moisture, $F_{M_{HP}}$, is a function of this variable and the first stage bowl pressure. The thermodynamic properties of the wet-steam are unknown at the discharge section, thus an iterative calculation is needed. Firstly, the steam quality value at the discharge section, $x_{4'}$, should be estimated by calculating the average moisture content, M_{1-4}:

$$M_{1-4} = 100\left(1 - \frac{x_{4'} + x_1}{2}\right) \qquad (3)$$

where, x_1 is the steam quality at the inlet. With this value and the first stage bowl pressure, the correction factor can be obtained by using the appropriate diagram. Thus the steam turbine efficiency will be calculated by:

$$\eta_{i2_{HP}} = \eta_{i_{HP}} F_{speed} F_{M_{HP}} \quad (4)$$

The enthalpy at the discharge section will be:

$$h_{2'} = h_1 - \eta_{i2_{HP}} \left(h_1 - h_{4s} \right) \quad (5)$$

where, h_{4s} is the enthalpy at the discharge section after an isentropic expansion:

$$h_{4s} = enthalpy(p = p_4; s = s_1) \quad (6)$$

And, finally, the steam quality $x_{4'}$ will be:

$$x_{4'} = quality(p = p_4; h = h_{4'}) \quad (7)$$

This value should be compared with the one initially assumed in Eq. (3), repeating the calculations until convergence is obtained.

Finally, the exhaust loss, EL_{HP}, is calculated as a function of the discharge pressure. Thus the enthalpy at the discharge section will be:

$$h_4 = h_{4'} + EL_{HP} \quad (8)$$

The procedure for these calculations is shown in Fig. (2).

The HP turbine efficiency will be:

$$\eta_{HP} = \frac{h_1 - h_4}{h_1 - h_{4s}} \quad (9)$$

The enthalpy at the HP turbine bowl is equal to the enthalpy of the steam supplied by the reactor, $h_1 = h_2$, and the enthalpy after the nozzles in the HP turbine admission, used later for the steam seal calculations:

$$h_3 = h_2 + \eta_{HP} \left(h_2 - h_{3s} \right) \quad (10)$$

where:

$$h_{3s} = enthalpy(p = p_3; s = s_1) \quad (11)$$

The LP turbine efficiency is calculated in a similar way. The procedure for the calculations is shown in Fig. (3). Firstly, the base efficiency, $\eta_{i_{LP}}$, should be defined from the bowl pressure and the bowl to condenser pressure ratio. Then, several factors should be applied to correct this efficiency: for volume flow and for initial moisture content.

The stage group efficiency correction for volume flow, F_v, is a function of the inlet volume flow and the bowl to exhaust pressure ratio. The stage group efficiency correction for initial moisture, $F_{M_{LP}}$, is a function of the bowl pressure and the bowl to exhaust pressure ratio. The LP turbine efficiency is calculated by:

$$\eta_{LP} = \eta_{i_{LP}} F_v F_{M_{LP}} \quad (12)$$

The enthalpy of each turbine section can be calculated using this efficiency. At a particular stage (i):

$$h_{(i)'} = h_{(i-1)} + \eta_{LP} \left(h_{(i-1)} - h_{(i)s} \right) \quad (13)$$

where

$$h_{(i)s} = enthalpy(p = p_{(i}); s = s_{(i-1)}) \quad (14)$$

The enthalpy $h_{(i)'}$ is the value of this thermodynamic property after steam expansion. The enthalpy of the steam that goes to the next turbine section, $h_{(i)}$, will be the same if the quality remains constant after the steam extraction to the feedwater heaters, or different (a little bit higher) if the quality is improved.

Thus the Expansion Line End Point (*ELEP*) will be:

$$ELEP = h_{15'} = h_{14} + \eta_{LP}\left(h_{14} - h_{15s}\right) \qquad (15)$$

where

$$h_{15s} = enthalpy(p = p_{15}; s = s_{14}) \qquad (16)$$

In the last stage, the exhaust loss, EL_{LP}, should be taken into account, which is a function of the last bucket velocity. Thus, the Used Energy End Point (*UEEP*) will be:

$$UEEP = h_{15} = h_{15'} + 2.024 \cdot EL_{LP} \cdot x_{15'}\left(0.9935 + 0.0065 x_{15'}\right) \qquad (17)$$

2.4. Steam flows to feedwater heaters

In power plants, steam is extracted from the steam turbines to increase the feedwater temperature. So, the necessary thermal energy in the steam generator can be reduced, increasing overall plant efficiency. Furthermore, using a special configuration of the blades, some of these extraction points in nuclear power plants are used to improve steam quality.

The effectiveness of this moisture removal is a function of the pressure at the extraction point. The moisture removal effectiveness at a particular (*i*) stage will be the relationship between the mass flow of moisture removed, $\dot{m}_{(i)e_l}$, and the moisture content of the steam before the extraction, $(1 - x_{(i')})$, times the mass flow of the steam that goes to the next stage, $\dot{m}_{(i)}$:

$$\varepsilon_{s_{(i)}} = \frac{\dot{m}_{(i)e_l}}{\dot{m}_{(i)}(1 - x_{(i')})} \qquad (18)$$

The rest of the steam that goes to the heater will be wet-steam with the same quality as the steam after the extraction point.

There are two ways in which the quality is improved in LP turbines: by draining the water from the turbine shells and by extracting the steam. In both cases, the fluid goes to the feedwater heaters. In the calculation process, the most complete case is the one in which the steam that goes to the feedwater heater comes from a drain and from an extraction stage, as shown in Fig. (4).

Following the notation of this drawing, the steam goes into the ($i - 1$) stage and expands until stage (*i*). Some steam is drained off this stage to the feedwater heater, (*i*)*e*, thus the steam after the drain point, (*i*), will show an improved quality. The mass flow of moisture removed can be calculated by using the Eq. (18). An additional 0.5% of wet steam is assumed to be drained, with the same quality as the steam after the drain point, (*i*). So:

$$\dot{m}_{(i)e-l} = \varepsilon_{s_{(i)}} \dot{m}_{(i)}(1 - x_{(i')}) \qquad (19)$$

$$\dot{m}_{(i)e-v} = 0.005 \dot{m}_{(i')} x_{(i)} \qquad (20)$$

The steam expands from stage (*i*) to stage ($i + 1$). In this case, the moisture that is removed will be calculated in the same way, but the wet-steam extracted should be calculated from the thermal balance of the feedwater heater. In this component, the steam (in wet-steam condition) coming from the drain point and from the extraction point (*hi*), plus the steam coming from other processes (*d*) is condensed (from the thermodynamic states *hi* and *d* to *h*2), and then sub-cooled (*ho*). Thus the feedwater is heated (from *ci* to *c*2, until *co*). The mass flows will be:

$$\dot{m}_{(i+1)e-l} = \varepsilon_{s_{(i+1)}} \dot{m}_{(i+1)} (1 - x_{(i+1)'}) \qquad (21)$$

$$\dot{m}_{(i+1)e-v} = \frac{1}{h_{(i+1)} - h_{h2}} [\dot{m}_{ci}(h_{co} - h_{c2}) - \dot{m}_{(i+1)e-v}(h_{(i+1)e-v} - h_{h2}) - \sum \dot{m}_d (h_d - h_{h2})] \qquad (22)$$

Once the LP turbine efficiency has been calculated, this efficiency should be applied to each turbine section, correcting the quality in each case by using the moisture removal effectiveness. Furthermore, the mass flows going through each section should be corrected by using the above equations when applicable.

2.5. Steam flows to seals

The mass flow going through the steam turbines should be corrected by taking the quantities into account that are used for the valve and turbine seals. In general, the steam flow through the seals is calculated by using Martin's equation [31]. If the pressure through the seal drops from the initial pressure of p_{1p} to a final pressure of p_{2p}, then the mass flow through the seal will be:

$$\dot{m}_p = 4.6459 \cdot 10^{-6} k_p A_p \sqrt{\frac{1 - \left(\frac{p_{2p}}{p_{1p}}\right)^2}{N_p - \ln\left(\frac{p_{2p}}{p_{1p}}\right)}} \sqrt{\frac{p_{1p}}{v_{1p}}} \qquad (23)$$

where, A_p is the sectional area in the clearance between the dynamic and the static surfaces in *mm*2, N_p the number of throttlings through the labyrinth, v_{1p} the upstream specific volume and k_p a coefficient. This last variable is tabulated as a function of the packing geometry [30].

In the power plant under study, the valve seals will be calculated using the equations provided in the *Thermal Kit*. Thus a 0.1% of the mass flow supplied to the steam turbines is used for the valve seals, corrected with the actual upstream pressure. Using the notation in Fig. (1):

$$\dot{m}_1 = 0.001 \dot{m}_{1;d} \frac{p_1 - p_{HTR11}}{p_{1;d}} \qquad (24)$$

$$\dot{m}_2 = 0.001 \dot{m}_{1;d} \frac{p_{HTR11}}{p_{1;d}} \qquad (25)$$

2.6. Steam seal regulator

The following flows go to the steam seal regulator: the steam leaked from the turbine valves, $\dot{m}_2$, and the steam leaked from the HP turbine seals, $\dot{m}_4$ and $\dot{m}_5$. These flows are used for the LP turbine seals. In Fig. (1), it is represented as a flow that goes to the main condenser and a flow that goes to the SPE. In Fig. (5) these flows are properly represented.

According the steam flows represented in Fig. (5), the mass flow and enthalpy of the steam that goes to the feedwater heater can be solved by a simple balance of mass and energy, taking into account that the mass flows $\dot{m}_6$ and $\dot{m}_7$ can be calculated by using Eq. (23).

2.7. Mechanical losses in the steam turbine and generator

Mechanical losses in the steam turbine are often included in the Thermal Kit as a fix value. In the literature, however, diagrams for their calculation as a function of the apparent power supplied by the generator can be found. The diagram for the calculation of these losses in [10] is used for the generation of the model.

In Fig. (6) the generator losses of the power plant under study are plotted as a function of the apparent power.

3. Results and discussion

The SMGNPP has been fully modeled using the equations and procedures described above. Firstly, using the on-design parameters provided in the Thermal-Kit, the thermodynamic properties and mass flows of the steam in each state and represented in Fig. (1) were simulated, as well as the gross electric power that is generated.

The following parameters should be introduced to simulate the power plant: pressure and quality of the steam that goes to the steam turbines (p_0 and x_0), mass flow of the steam that goes to the SJAE ($\dot{m}_6$), the generator power factor, the mass flow of feedwater to CRD and the mass flow of feedwater returning to the reactor (after HTR12A/B), the mass flow of make-up water to the cycle and the condenser pressure (p_{15}).

The calculated gross electric power for the on-design parameters was 467.2 *MWe*, in all 7 *MWe* higher than the expected value. According to the thermal balance included in the Thermal-Kit, the HP turbine efficiency should be 83.79% and the LP turbine should be 82.45%. However, these efficiencies in the model were 84.36% and 84.32%, respectively. The difference could be due to a higher base efficiency used in the traditional model, which is applied in this case to a steam turbine built in the nineteen-sixties.

The model was used to calculate the base efficiency that achieves the efficiencies indicated by the designer for the steam turbines in the SMGNPP. The new gross electrical power of 459.5 *MWe* is in good agreement with the figure specified by the designer.

In Tab. (1) the values of mass flow ($\dot{m}_i$), pressure (p_i), enthalpy before (h'_i) and after (h_i) the extraction points and the thermal power ($\dot{W}_i$) of each relevant thermodynamic state *i* are tabulated according to Fig. (1) from data supplied by the designer in the Thermal-Kit and calculated by using the model, as well as the deviation between them. The deviation of the thermal power is 0.02 % (the deviation between the gross power that is calculated and that estimated by the designer is 0.04 %).

If the thermal power in each section is analyzed, the deviation in the thermodynamic states corresponding to the LP turbine sections is highly significant. However, the global result is very close between both calculations.

The mass flow looks similar, although the mass flows into the seals are in fact significantly different, when individually considered. The rest of parameters are in good agreement.

Thus, it is very curious that the values of mass flow and thermal power are so close in the global result, although they are significantly different in each turbine section.

The model was applied to different performance tests performed in the nineteen-nineties. The objective is to compare the values estimated by using the present model with the readings taken from instruments during those tests.

The values to be introduced in the model are tabulated in Tab. (2) and the deviations between the estimated main parameters of the cycle and the measured values are tabulated in Tab. (3).

In view of the gross electrical power, the results obtained can be considered acceptable. However, some of the parameters show high deviations, especially the pressures and the drain temperatures of the feedwater heaters. The first are a consequence of the use of the Stodola equation (with the exception of the pump discharge pressures, calculated by using the characteristic curves of each one), and the second are the consequence of the hypothesis that the TTD and DCA of the feedwater heaters are constant (the feedwater temperature presents a low deviation, but the drain temperature depends on the steam condensed and subcooled in the heater).

However, in the case of the deviations in the heater pressures, the measured values have been proven to be wrong in cases where these deviations are more significant, because the resulting TTD would be negative. The high deviation in the HP turbine discharge pressure is of interest, especially taking into account the low deviation in the HTR12 pressure.

4. Conclusion

In the present paper, the conventional approach to the thermal balance of wet-steam turbines in nuclear power plants has been applied to a particular case study: the Santa María de Garoña Nuclear Power Plant.

The on-design thermal balance has been compared with the thermal balance developed by the designer, and included in the Thermal-Kit documentation. In general terms, both calculations were quite similar, but the mass flow and thermal power of each particular turbine section yielded different results. The mass flow for turbine seals were also different from those expected in the original calculations. Furthermore, the base efficiency has been corrected, to take into account that the power plant under study was built prior to the development of the procedures that were applied in this study.

The model has been applied to compare the parameters simulated with those measured during several performance tests. Some of the feedwater heater pressure values obtained were highly different from those measured. However, it can be proven that the measured values are wrong: the value of the saturated temperature at this pressure and the outlet feedwater temperature implied a negative TTD value in these cases. Thus the results of the Stodola equation are not demonstrably unacceptable.

Besides, the deviations in the feedwater heater drain temperatures imply that the simplification in the use of constant TTD and DCA values could be improved. Once this correction is implemented, a proper analysis of the application of the Stodola equation could be performed.

In any case, the development of a rigorous model using the conventional methodology for the simulation of nuclear power plants could be considered acceptable for diagnostic purposes, bearing in mind that simplifications in the process are, in the majority of cases, the cause of unacceptable deviations.

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Figure 1: CNMSG schematic diagram

Figure 2: HP turbine discharge enthalpy calculation

Figure 3: LP turbine discharge enthalpy calculation

Figure 4: Drain and extraction to a feedwater heater

Figure 5: Steam flows in the steam seal regulator

Figure 6: Generator losses

Table 1: Mass flow and thermodynamic properties of the steam according to Fig. (1)
[T.K. = Thermal-Kit; Calc. = Calculated; Dev. = Deviation].

	$\dot{m}_i$			p_i			h'_i			h_i		
	T.K.	Calc.	Dev.	T.K.	Calc.	Dev.	T.K.	Calc.	Dev.	T.K.	Calc.	Dev.
	kg/s		%	*MPa*		%	*kJ/kg*		%	*kJ/kg*		%
0	684.7	684.7	0.00	6.65	6.65		2,770.28	2,772.84	-0.09	2,770.28	2,772.84	-0.09
1	684.1	684.1	0.00	6.65	6.65		2,770.28	2,772.84	-0.09	2,770.28	2,772.84	-0.09
3	683.4	683.4	0.00	6.09	6.09	0.00	2,741.21	2,772.84	-1.14	2,741.21	2,772.84	-1.14
4	678.9	680.0	-0.16	1.19	1.20	-1.55	2,527.68	2,532.10	-0.17	2,527.68	2,532.10	-0.17
5	616.4	617.1	-0.11	1.19	1.20	-1.55	2,527.68	2,532.10	-0.17	2,527.68	2,532.10	-0.17
7	548.8	550.6	-0.32	1.12	1.12	-0.18	2,740.51	2,738.65	0.07	2,740.51	2,738.65	0.07
8	526.6	526.2	0.08	0.47	0.47	-0.09	2,603.97	2,609.09	-0.20	2,603.97	2,609.09	-0.20
9	500.4	500.1	0.07	0.19	0.19	-0.32	2,482.55	2,487.90	-0.22	2,482.55	2,487.90	-0.22
10	494.7	494.5	0.06	0.12	0.12	-0.22	2,439.29	2,430.92	0.34	2,425.33	2,444.64	-0.79
11	468.5	468.4	0.01	0.08	0.08	-0.25	2,411.61	2,392.77	0.79	2,389.28	2,413.94	-1.02
12	460.3	460.1	0.03	0.06	0.06	-0.28	2,395.56	2,373.93	0.91	2,369.98	2,403.01	-1.37
13	427.9	428.7	-0.20	0.03	0.03	-0.73	2,363.46	2,326.25	1.60	2,321.13	2,372.53	-2.17
14	414.8	415.9	-0.26	0.02	0.02	-0.53	2,354.86	2,307.17	2.07	2,300.66	2,366.25	-2.77
15	414.8	415.9	-0.26	0.01	0.01			2,239.49		2,279.49	2,288.80	-0.41

Table 2: Parameters introduced in the model, instrumented during several performance tests

Description	oct-92	feb-94	jun-94	jun-05
Rx to HP turbine pressure (MPa)	6.64	6.62	6.64	6.62
Condenser pressure (mmHg)	39.48	31.95	42.47	34.42
Feedwater mass flow (kg/s)	689.17	676.80	685.40	692.55
SJAE mass flow (kg/s)	0.66	0.66	0.66	0.66
CRD mass flow (kg/s)	1.47	1.39	1.50	1.52
factor of power	0.9879	0.9835	0.9971	0.9934

Table 3: Deviation between the estimated and the instrumented values during several performance tests *[pressure in MPa, temperature in °C and electrical power in MWe]*

	oct-92	feb-94	jun-94	jun-05
First stage pressure (HP)	5.47	13.83	3.76	3.13
HP discharge pressure	3.54	3.25	4.32	4.35
LP bowl pressure	1.46	1.88	0.38	0.67
Condensate pumps discharge pressure	5.38	4.85	8.46	9.00
Feedwater pumps discharge pressure	0.91	1.03	1.20	1.95
HTR12 pressure	0.73	0.27	1.14	0.86
HTR11 pressure	1.34	1.26	0.96	3.12
HTR9 pressure	2.18	5.14	8.29	2.53
HTR8 pressure	19.05	18.93	26.28	13.64
HTR7 pressure	4.93	63.72	35.35	6.57
Condensate to SJAE temperature	1.02	1.81	0.44	1.65
Condensate to SPE temperature	1.03	0.64	0.91	1.70
Condensate to HTR7 temperature	0.58	0.51	0.49	1.31
Condensate to HTR8 temperature	1.07	2.60	0.70	2.42
Condensate to HTR9 temperature	1.11	1.56	0.63	1.00
Condensate to FW pump temperature	0.70	0.13	0.49	0.13
Condensate to HTR11 temperature	0.83	0.23	0.27	0.22
Condensate to HTR12 temperature	0.18	0.53	0.80	0.77
Condensate to reactor temperature	0.22	0.69	0.16	0.49
HTR12 drain temperature	1.07	0.60	0.27	0.21
HTR11 drain temperature	2.33	1.06	1.09	0.92
HTR9 drain temperature	2.02	1.44	0.83	2.74
HTR8 drain temperature	3.20	4.20	2.48	2.17
HTR7 drain temperature	2.70	3.78	2.52	3.60
Gross electrical power	0.50	0.52	0.24	0.17