

## An Analytical Procedure for the Assessment of Malfunctions in Thermomechanical Systems

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### Abstract

The behaviour of the thermal components is commonly described through a set of internal parameters: isentropic efficiencies, pressure ratios, effectiveness factors, pressure losses, temperature relations or differences, et cetera. In this paper a new type of internal parameters, the internal parameters  $\theta$  (Royo 1994, Royo and Valero 1995) are defined, that are specially adequate for the characterisation of thermal systems in an analytical way.

When a component of a thermal plant displays an internal deterioration (intrinsic malfunction) its performance gets worse. This fact is reflected in a variation of the internal parameters describing its behaviour. On the other hand, the remaining components of the plant may be affected (induced malfunction) because they are working under different operating conditions from the usual ones.

Knowledge of variations of the internal parameters in each component of a thermal plant does not suffice to determine their performance state under these new conditions. Additional information is needed. The dissipation temperature parameter can supply it (Royo 1994, Royo et al. 1997) as is explained in this paper.

Internal  $\theta$  parameters and the dissipation temperature parameters are appropriate tools for the study analysis and evaluation of malfunctions in thermomechanical systems. Three examples of this are shown, trying to quantify the influence of an intrinsic malfunction on the whole plant. This paper shows a strikingly new and simple vision of the analysis of thermal systems.

*Keywords: energy systems analysis, malfunctions, second law analysis, internal parameters.*

### 1. Introduction

Throughout the life of a thermal plant, several variations in the working conditions of their components frequently come about. These degradations are called malfunctions and may be classified in two types depending on their origins (Royo 1994, Royo et al. 1997):

- Intrinsic malfunction:

All machines deteriorate gradually with the passage of time, with the result that they cannot fulfil their purpose in the same way as when they were built. Thus, for example the blades of a turbine or a compressor frequently become eroded or dirty due to the continuous stream passage of gas or steam

to which they are exposed, which leads to a decrease in their isentropic performance. It is also common for solid deposits to accumulate in the tubes of heat exchangers, leading to increasing loss in the load and/or poorer heat transfer.

When the working of a component is impaired due to internal deterioration, it will be referred to as suffering from an intrinsic malfunction. The case of an improvement of the component, due to an appropriate internal modification, is similar to this (a negative intrinsic malfunction.)

- Induced malfunction:

When intrinsic malfunction arises in a component, as well as working under

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poorer conditions than the original ones, its energy output flows are modified in terms of quantity (m, W or Q) and/or quality (P, T, h, s,...)<sup>1,2</sup>. Since these flows are inputs of other components, such modifications lead to the rest of the components of the plant also working under conditions different from usual ones.

When a component in a plant has to work under off-design operating conditions caused by the appearance of an intrinsic malfunction in another component, it will be referred to as suffering from an *induced* malfunction. This type of malfunction is characterized by a change of the quantity and/or quality of its input flows.

Obviously, these changes in the behaviour of the components (due to intrinsic or induced malfunctions) imply a variation in the global behaviour of the plant. But how can this influence be quantified? This question is of utmost importance when it comes to making an energy and economic assessment of maintenance, repair or retrofitting decisions.

In order to answer the question, two different procedures may be used: a numeric procedure, by means of a simulator or an analytic procedure. The analytic procedure has the advantage of providing a deeper knowledge of the phenomena taking place in the plant. The aim of this paper is to show an analytic procedure for the thermodynamic study of malfunctions by using the internal  $\theta$  parameters and the dissipation temperature parameter.

## 2. Internal $\theta$ Parameters

When a mass flow goes through a component, it undergoes a process, throughout which its thermodynamic properties (P, T, h, s,...) are modified.

If the following assumptions are made,

- All the processes undergone by mass flows are considered to be quasi-static
- Mass flows are considered to be one-dimensional<sup>3</sup>
- Components work in a stationary state

then it is possible to represent the variation of mass flow properties through the process in an h-s diagram, obtaining as a result a specific tra-

jectory associated with each process which, in general, can be described by means of a mathematical function of the  $h = h(s)$  type.

It was proposed that such mathematical functions be called *process trajectory equations* (Royo 1994, Royo and Valero 1995) of the process undergone by the mass flow on going through a specific component (or part of the component, depending on the disaggregation level).

Obviously, the form of these process equations differ in terms of the process considered, and can be extremely complex, as it is the case for a turbine or a compressor (i.e. a simple ideal gas with constant calorific values, isobaric heating has the equation  $h=h_1+c_p T_1(\exp((s-s_1)/c_p)-1)$  as it is well known).

The classical internal parameters of the components do not usually describe these processes, but relate the properties of their output mass flows with those of their input mass flows.

In the same way, it is possible to describe the processes by means of a fictitious trajectory defined by a straight line which passes through both states. Assuming this linear characterisation, the substituted process equation would take the following form:

$$h = \theta s + \beta \quad (1)$$

When talking about process equations in this paper, reference will always be made to this linear characterisation. Note that it is possible to achieve an approach to the real process equation with as many linear equations as needed.

It should be noted that, since it is only necessary to know the input and output mass flows states of the component in order to define the process equation, there is no need for all the intermediate states of the process to be in equilibrium (quasi-static process) or for these flows to be one-dimensional at all times. These two conditions are only essential for the input and output mass flows states.

While in the intermediate states of the process this relation between h and s will only be complied with approximately, in the initial and final states it will be fulfilled exactly, since the straight line has been defined such that it passes through these two states. That is:

$$h_i = \theta_{ij} s_i + \beta_{ij} \quad h_j = \theta_{ij} s_j + \beta_{ij} \quad (2)$$

By eliminating the  $\beta$  parameter from both equations, one obtains:

$$\theta_{ij} = \frac{h_i - h_j}{s_i - s_j} \quad (3)$$

which is the slope of the process equation (see *Figure 1*) and can be interpreted as an internal parameter of the process (Royo 1994, Royo and Valero 1995). Although  $\theta$  has temperature di-

<sup>1</sup> In this paper we only treat thermomechanical flows. Potential and kinetic energy variations are neglected.

<sup>2</sup> The meaning of h and s is  $(h-h_{ref})$  and  $(s-s_{ref})$  respectively. That is, the values of the enthalpy and entropy of the flow with respect to some predetermined reference state.

<sup>3</sup> This means that:

- The flow is normal to the area through which matter flows.
- All intensive properties, including velocity, are uniform with position over each area through which matter flows.

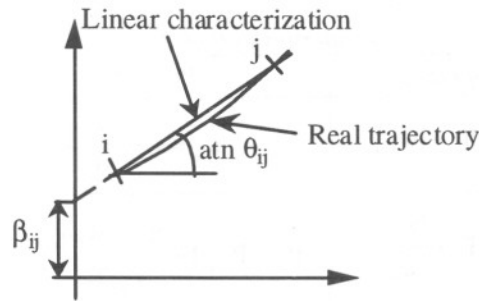


Figure 1. Characterisation of a real process using a linear trajectory in a  $n$   $h$ - $s$  diagram.

mensions, its sign can be positive or negative, since it does not respond to any measurable property but associated with the process undergone by the mass flow.

For the same reason, in the case of a component which processes more than one mass flow, it will be necessary to define one of these internal parameters for each flow processed.

Ishida and Chuang (1996) use the "energy quality degradation",  $d \equiv \Delta s / \Delta h$ , in order to describe the exergy change in processes. Such parameter  $d$  coincides conceptually with the internal  $\theta$  parameter,  $d = 1/\theta$ . Therefore, all ideas developed by Ishida and Chuang for "d" apply directly to "θ", thus enhancing the power and applications of "θ" or "d" parameters.

### 2.1. Internal $\theta$ parameters versus conventional ones

Internal  $\theta$  parameters and classical parameters have the following characteristics (Royo 1994):

- They are used to describe the behaviour of the component
- They depend solely on the component

However, it has been shown that internal  $\theta$  parameters always have the same formula, regardless of the component they describe, while conventional internal parameters do not. In other words, internal parameters  $\theta$  are normalised.

Despite this advantage, which may be very

interesting, the main property of internal  $\theta$  parameters is their far greater simplicity when it comes to describe the analytical behaviour of the components (Royo 1994, Valero et al. 1995, Royo and Valero 1995). In order to show this, an adiabatic compressor will be used (Figure 2).

Assuming that the input flows to the compressor ( $m$ ,  $h_1$ ,  $s_1$ ,  $P_1$ ,  $T_1$ ,  $W_C$ ) and the internal parameter are fully determined, the objective is to determine the output state 2 analytically, that is to find out two independent properties of this state.

The energy balance of the adiabatic compressor (once the material balance has been entered:  $m_1 = m_2 = m$ ) must always be fulfilled:

$$0 = |W_C| + m (h_1 - h_2)$$

or

$$0 = |w_C| + (h_1 - h_2) \quad (4)$$

The problem can be solved in two different ways:

- By using the internal  $\theta$  parameter.
- By using a conventional internal parameter which, for a compressor, is usually its isentropic efficiency,  $\eta_{s,c}$ .

It can be observed that, in general, the second case may not be analytically solved, unless the working fluid would be characterised by means of a sufficiently simple behaviour model. Only in this second case it is assumed that the working fluid is an ideal gas, with constant calorific values.

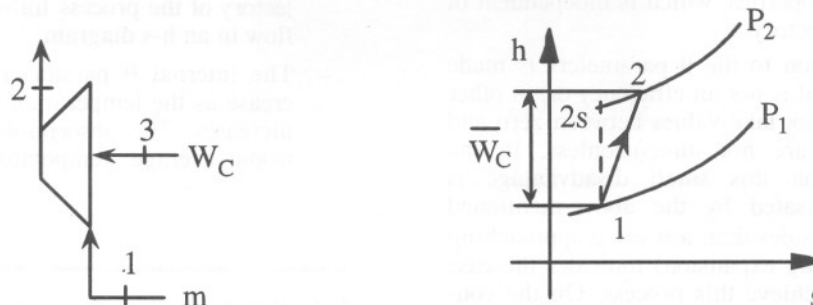


Figure 2. Adiabatic compressor.

TABLE I. Solution of adiabatic compressor using  $\theta$  or  $\eta_{s,c}$ .

Internal parameter	Properties of flow 2		Available data
	Property 1	Property 2	
$\theta_{12} = \frac{h_1 - h_2}{s_1 - s_2}$	$h_2 = h_1 +  w_c $	$s_2 = s_1 + \frac{ w_c }{\theta_{12}}$	$h_1, s_1,  w_c , \theta_{12}$
$\eta_{s,c} = \frac{h_{2s} - h_1}{h_2 - h_1}$	$T_2 = T_1 + \frac{ w_c }{c_p}$ (* )	$P_2 = P_1 \left( 1 + \frac{\eta_{s,c}  w_c }{c_p T_1} \right)^{\gamma/(\gamma-1)}$ (* )	$T_1, P_1,  w_c , \eta_{s,c}$ (* )

(\* ) Considering an ideal gas model with constant calorific values.

The results obtained in both cases are shown in TABLE I.

The following statements can be made in relation to the results obtained:

- In order to determine the state of flow 2, it was necessary in both cases to establish two properties which were independent of flow 1, the work per unit of mass processed and one internal parameter of the compressor. It is obvious that, the knowledge of the compression work could be substituted by any other internal parameter such as the pressure ratio.
- If one takes the internal  $\theta$  parameter, and if the aim is solely to characterise the state of flow 2 by means of  $h$  and  $s$  (or a combination of both properties, such as their exergy, for example) it is not necessary to use any behaviour model of the working fluid. This fact can be applied to several other plant components (Royo 1994).
- Taking the isentropic efficiency of the compressor as the internal parameter and even assuming that the working fluid can be modelled by means of a very simple model, the analytic expressions of the properties of the state of flow 2 are much more complex than those obtained when the  $\theta$  parameter is used. If the working fluid had been steam, it would have been practically impossible to obtain these analytic expressions.

Note also that, in both cases, the problem is solved as in a black box theory. This is because  $\eta_{s,c}$  as well  $\theta_{12}$  are defined as the quotient of the difference of properties, which is independent of the process trajectory.

An objection to the  $\theta$  parameters is made in relation that it is not an efficiency or, in other words, it does not take values between zero and one and they are not dimensionless. In the author's opinion this small disadvantage is largely compensated by the above-mentioned advantages. Besides that, a  $\theta$  value approaching to zero (throttling expansion) indicates the ease to practically achieve this process. On the contrary, the larger the  $\theta$  absolute values are the more practical difficulties appear during the

process design ( $\theta = \pm \infty$ , for isentropic processes).

## 2.2 Some real values of internal $\theta$ parameters

In order to get some values for the internal  $\theta$  parameters of actual components, a Rankine steam cycle plant of 350 MW has been chosen.

From the energy balance of this installation<sup>4</sup> it is possible to obtain TABLE II, which includes the properties of the mass input and output flows (only the main ones have been considered) of certain important components in the installation.

From the values shown in TABLE II, it is possible to obtain the internal  $\theta$  parameters of each component, the isentropic efficiencies of the turbines and the pump, and the pressure losses in the heaters, as shown in TABLE III.

The following observations can be made from the obtained values:

- The internal  $\theta$  parameter of turbines is always negative and its absolute value is high. This is logical, as  $\theta$  is the slope of the process equation which for the turbine presents a pronounced but negative slope.
- As it would be expected, the greater the isentropic efficiency of the turbines the greater the absolute value of the  $\theta$  parameter.
- The value of the internal  $\theta$  parameter of the water pump which feeds the boiler is also high, and positive. This fact can also be confirmed by the observation of the trajectory of the process followed by the mass flow in an  $h$ - $s$  diagram.
- The internal  $\theta$  parameters of heaters increase as the temperature of the mass flow increases. The absorption or delivery entropic average temperature,  $T_m$  (Alefeld

<sup>4</sup> Neither its  $h$ - $s$  plane nor a components diagram is presented due to the fact that it is a totally conventional plant.

TABLE II. Properties of flows in the installation.

Flow	No	P (bar)	T (K)	h (kJ/kg)	s (kJ/kg-K)
High pressure turbine inlet	2	166.2	812.0	3398.8	6.417
High pressure turbine outlet	5	44.7	626.1	3084.8	6.522
Intermediate pressure turbine – 1 inlet	8	40.4	810.4	3526.3	7.188
Intermediate pressure turbine – 1 outlet	9	21.8	719.7	3342.8	7.225
Intermediate pressure turbine – 2 inlet					
Intermediate pressure turbine – 2 outlet	11	8.4	593.9	3092.4	7.272
Low pressure turbine inlet					
Low pressure turbine outlet	12	0.07	311.0	2457.4	7.903
Heater – 1 feed water inlet	18	13.8	313.1	168.7	0.572
Heater – 1 feed water outlet	19	12.9	337.5	270.6	0.885
Heater – 2 feed water inlet					
Heater – 2 feed water outlet	20	12.0	362.6	375.8	1.186
Heater – 3 feed water inlet					
Heater – 3 feed water outlet	21	11.1	400.7	536.7	1.608
Pump inlet	23	8.0	441.8	713.1	2.028
Pump outlet					
Heater – 5 feed water inlet	24	196.6	444.0	743.7	2.049
Heater – 5 feed water outlet	26	192.6	486.8	934.6	2.457
Heater – 6 feed water inlet					
Heater – 6 feed water outlet	27	191.6	527.4	1125.8	2.832

1987) and the  $\theta$  parameters of these components are partly related (Royo 1994), which is in accordance with the result obtained here.

### 3. The Dissipation Temperature Parameter

The appearance of an *intrinsic malfunction* in a component is always the consequence of the deterioration of one of its internal elements, which leads to a decline in its behaviour. The internal  $\theta$  parameter describes the behaviour of the component and it is, therefore, logical that when an intrinsic malfunction comes about, this parameter is modified.

Thus, for example, if the blades of a compressor deteriorate, its isentropic efficiency will decrease, resulting, as it is well known, in a corresponding decrease in the trajectory slope followed by the mass flow on the h-s plane, that is, its internal  $\theta$  parameter. Nonetheless, even if the new internal  $\theta$  parameter of the component is known when the intrinsic malfunction occurs, the new process has not yet been fully determined. For this purpose it is necessary to impose an additional condition on the output mass flow state (Royo 1994, Royo et al. 1997).

On the other hand, the appearance of an *induced malfunction* in a component is due to the variation in the quantities and/or qualities of its input flows, as a result of the appearance of an intrinsic malfunction in another component

in the plant. This variation in the properties of the input flows will obviously affect the behaviour of the component in question, but as an initial approach, it is very reasonable to assume that, if the variation in the properties is differential, its internal  $\theta$  parameters will remain constant, which is what will be assumed in this paper<sup>5</sup>. Nonetheless, when an induced malfunction takes place, assuming that the input mass flow state,  $i'$ , is known, and that the internal  $\theta$  parameter will not be modified, the output flow state is not fully determined either. As in the case of intrinsic malfunctions, additional information is necessary to determine the state fully.

#### Definition of the dissipation temperature parameter

The additional information mentioned above could be the knowledge of the trajectory followed by the state of the outlet mass flow

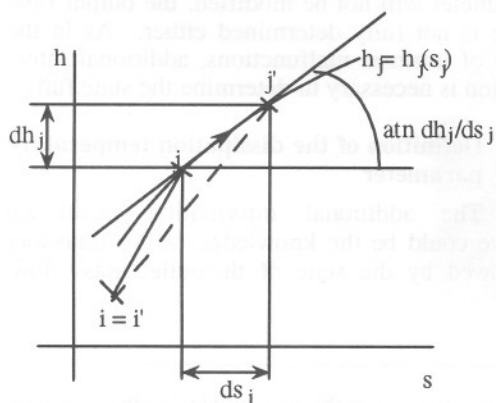
<sup>5</sup> Nevertheless, in the case in which either a certain unit cannot fulfil the hypothesis referred to, or where greater precision is required in the calculations, it must be known how  $\theta$  varies in terms of the quantity and the energy and entropy quality of the input flows. However, the methodology used here will be absolutely valid, although the expressions obtained, in the latter case may be a little more complex.

TABLE III. Internal parameters of the components.

Component	Parameter $\square$	Conventional parameter
High pressure turbine	$\theta_{\text{HPT}} = \frac{h_2 - h_5}{s_2 - s_5} = -2983.3 \text{ K}$	$\eta_{s,\text{HPT}} = 83.0 \%$
Intermediate pressure turbine – 1	$\theta_{\text{MPT1}} = \frac{h_8 - h_9}{s_8 - s_9} = -4934.8 \text{ K}$	$\eta_{s,\text{MPT1}} = 87.4 \%$
Intermediate pressure turbine – 2	$\theta_{\text{MPT2}} = \frac{h_9 - h_{11}}{s_9 - s_{11}} = -5532.1 \text{ K}$	$\eta_{s,\text{MPT2}} = 90.1 \%$
Low pressure turbine	$\theta_{\text{LPT}} = \frac{h_{11} - h_{12}}{s_{11} - s_{12}} = -1005.4 \text{ K}$	$\eta_{s,\text{LPT}} = 76.3 \%$
Heater – 1 feed water side	$\theta_{\text{H1}} = \frac{h_{18} - h_{19}}{s_{18} - s_{19}} = 325.3 \text{ K}$	$\Delta P_{\text{H1}} = 0.9 \text{ bar}$
Heater – 2 feed water side	$\theta_{\text{H2}} = \frac{h_{19} - h_{20}}{s_{19} - s_{20}} = 349.6 \text{ K}$	$\Delta P_{\text{H2}} = 0.9 \text{ bar}$
Heater – 3 feed water side	$\theta_{\text{H3}} = \frac{h_{20} - h_{21}}{s_{20} - s_{21}} = 380.9 \text{ K}$	$\Delta P_{\text{H3}} = 0.9 \text{ bar}$
Heater – 5 feed water side	$\theta_{\text{H5}} = \frac{h_{24} - h_{26}}{s_{24} - s_{26}} = 467.7 \text{ K}$	$\Delta P_{\text{H5}} = 4.0 \text{ bar}$
Heater – 6 feed water side	$\theta_{\text{H6}} = \frac{h_{26} - h_{27}}{s_{26} - s_{27}} = 510.5 \text{ K}$	$\Delta P_{\text{H6}} = 1.0 \text{ bar}$
Pump	$\theta_{\text{P}} = \frac{h_{23} - h_{24}}{s_{23} - s_{24}} = 1418.0 \text{ K}$	$\eta_{s,\text{P}} = 68.5 \%$

when a malfunction occurs in the component. This malfunction trajectory can be expressed, in the  $h$ - $s$  plane, using a general equation like  $h_j = h_j(s_j)$

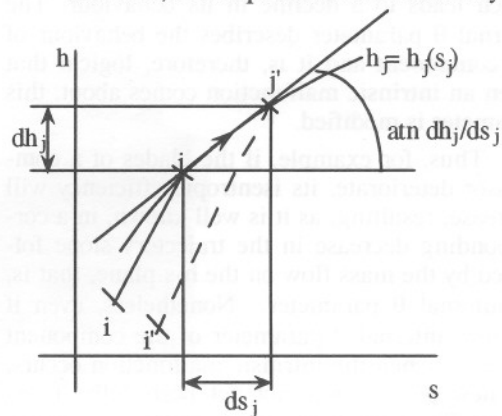
Thus, the new state  $j'$  is fully determined as



a)

the intersection of the malfunction trajectory and the straight line defined by the new internal  $\theta$  parameter (or by the new  $h_i$  and  $s_i$ , in the case of an induced malfunction.)

However, if it is pretended to work only



b)

Figure 3 (a,b). Trajectory followed by the state of the outlet mass flow when a malfunction occurs in the component.

with differential malfunctions (differential variations of the parameters  $\theta$  in the components which display an intrinsic malfunction), it will be enough to know the direction followed in the  $h$ - $s$  plane by the state of the mass flow (point  $j$ ), which is tangent to the above mentioned trajectory in the point  $j$ .

The slope of this tangent (see Figure 3), which has temperature dimensions, was introduced by Valero (Valero 1991, Valero and Lozano 1992) denoted as  $T_{xj}$ , and subsequently has been proposed to call it *dissipation temperature parameter* of the mass flow  $j$  and denoted as  $T_{dj}$  (Sciubba and Valero 1994, Royo 1994, Royo et al., 1997):

$$T_{dj} \equiv \left( \frac{\partial h_j}{\partial s_j} \right)_{\text{Malfunction}} \quad (5)$$

Some values of the dissipation temperature parameter.

The values of the dissipation temperature parameter depend on circumstances. Thus, for instance, a gas turbine that discharges into the atmosphere will always have a  $T_d$  equal to the discharge temperature, because

$$T_d \equiv (\partial h_j / \partial s_j)_{Pj} = T_j.$$

A similar situation arises in a low pressure steam turbine whose discharge pressure is the condenser vacuum pressure. Therefore, in that case  $T_d = T_{\text{vacuum}}$ . Another example is the case of a compressor whose objective is to keep an output target pressure regardless of the malfunctions occurring inside of the machine and subsequently the amount of work input required to achieve it. Also in this case  $T_d = T_j$ .

On the other and, a badly insulated combustor that losses heat through its walls has a 0 K dissipation temperature parameter because it does not increase the output gas enthalpy even at the cost of an additional fuel consumption.

The dissipation temperature parameter of an exiting mass flow should generally be calculated through equation (5) by means of the ob-

servation of the malfunction taking place in the component.

#### 4. Applications of $\theta$ and $T_d$

In this section, three examples of the use of  $\theta$  and  $T_d$  in relation with the analysis and evaluation of malfunctions in thermomechanical systems will be shown.

##### 4.1 Description of thermomechanical systems when a malfunction occurs

The knowledge of the internal  $\theta$  parameters and the dissipation temperature parameters of the mass flows of a thermal plant fully determine the new states of such flows when an intrinsic malfunction occurs in one of its components (characterised by a modification in its internal  $\theta$  parameter).

A gas turbine such as that shown in Figure 4 has been chosen as an example:

The internal  $\theta$  parameters of the components of this plant are as follows:

$$\theta_{12} = \frac{h_1 - h_2}{s_1 - s_2}, \quad \theta_{23} = \frac{h_2 - h_3}{s_2 - s_3}$$

$$\theta_{34} = \frac{h_3 - h_4}{s_3 - s_4} \quad (6)$$

It is possible to define the dissipation temperature parameter for each of the remaining mass flow states:

$$dh_2 = T_{d2} ds_2, \quad dh_3 = T_{d3} ds_3, \quad dh_4 = T_{d4} ds_4 \quad (7)$$

Supposing that the common operation of the plant keeps constant:

- The output mass flow pressure of the compressor, i.e.  $T_{d2} = T_2$ .
- The temperature (and, assuming a model of ideal gas, the enthalpy) of the output mass flow of the heat consumer component, i.e.  $T_{d3} = 0$ .

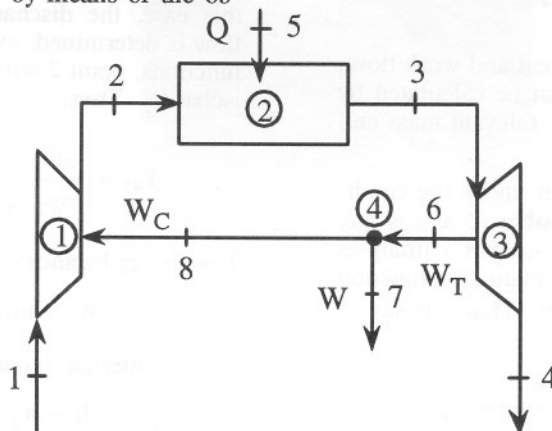


Figure 4. Components and interactions of a simple gas turbine.

TABLE IV. Variations of output enthalpies and entropies of the gas turbine flows of Figure 4 when an intrinsic malfunction takes place in different components

	Intrinsic malfunction in		
	Compressor	Heat consumer	Expander
dh <sub>2</sub>	$\frac{(s_1 - s_2)}{\left(\frac{\theta_{12}}{T_2} - 1\right)} d\theta_{12}$	0	0
ds <sub>2</sub>	$\frac{(s_1 - s_2)}{(\theta_{12} - T_2)} d\theta_{12}$	0	0
dh <sub>3</sub>	0	0	0
ds <sub>3</sub>	0	$\frac{(s_2 - s_3)}{\theta_{23}} d\theta_{23}$	0
dh <sub>4</sub>	0	$\frac{\theta_{34} (s_2 - s_3)}{\theta_{23} \left(\frac{\theta_{34}}{T_4} - 1\right)} d\theta_{23}$	$\frac{(s_3 - s_4)}{\left(\frac{\theta_{34}}{T_4} - 1\right)} d\theta_{34}$
ds <sub>4</sub>	0	$\frac{\theta_{34} (s_2 - s_3)}{\theta_{23} (\theta_{34} - T_4)} d\theta_{23}$	$\frac{(s_3 - s_4)}{(\theta_{34} - T_4)} d\theta_{34}$

- The output mass flow pressure of the turbine, i.e.  $T_{d4} = T_4$ .

Assuming that an intrinsic malfunction alternately takes place in the compressor (isentropic efficiency decrease) the heat consumer ( $\Delta P_{23}$  increase), and the expander (isentropic efficiency decrease).

From these hypotheses it is very easy (Royo et al. 1997) to obtain the values of the  $dh_j$  and  $ds_j$  of all the mass flow states of the plant in terms of  $\theta_{ij}$ , initial mass flow properties ( $s_i$  and  $T_i$ ), and the knowledge of the variation of the internal  $\theta$  parameter in the intrinsically malfunctioning component<sup>6</sup>.

The analytical results are shown in TABLE IV. Note that mass flow 1 is air from the environment, then its specific properties will remain constant:  $dh_1 = 0$  and  $ds_1 = 0$ .

The new values of the heat and work flows of the plant for each case can be calculated by means of TABLE IV and the relevant mass and energy balances.

Royo (1994) proved that under the conditions stated above, it is possible to assess any differential variation of the output enthalpies and entropies in any thermal plant as a function of the sets  $\{\theta\}$ ,  $\{T_d\}$ ,  $\{s\}$  and  $d\theta$ . That is to say

$$\{dh\} = f(\{\theta\}, \{T_d\}, \{s\}) d\theta_k \quad (8)$$

and

$$\{ds\} = f(\{\theta\}, \{T_d\}, \{s\}) d\theta_k$$

where  $d\theta_k$  is the given variation of the internal  $\theta_k$  parameter of the malfunctioning component,  $k$ . In other words, it is possible to simulate the behaviour of a plant from the knowledge of a plant's given state (plant's thermodynamic photograph) and the slopes in the ( $h$ ,  $s$ ) plane of the output flow (i.e.  $T_{dj}$ ) when different malfunctions occur. This could help to rethink simulators in a simpler and normalised way.

#### 4.2 Assessment of the importance of an intrinsic malfunction in component

Let us take the adiabatic compressor of Figure 4. The appearance of an intrinsic malfunction here will lead to a decrease in  $\theta_{12}$ . In this case, the discharge pressure of the mass flow is determined, even if the compressor malfunctions, point 2 will always be situated on the isobar  $P_2$ . Thus:

$$T_{d2} = \left( \frac{\partial h_2}{\partial s_2} \right)_{P_2 = \text{const.}} = T_2 \quad (9)$$

The energy balance of the compressor is:

$$0 = -W_C + m(h_1 - h_2) \quad (10)$$

And its internal  $\theta$  parameter:

$$\theta_{12} = \frac{h_1 - h_2}{s_1 - s_2} \quad (11)$$

<sup>6</sup> It has been assumed that the internal parameters  $\theta$  of the remaining components will not be modified.



By differentiating equation (11) under the imposed conditions:

$$d\theta_{12} = \left(1 - \frac{\theta_{12}}{T_2}\right) \frac{\theta_{12}}{h_2 - h_1} dh_2 \quad (12)$$

By differentiating equation (10), and assuming that:

- a) one wishes to maintain the mass flow processed by the compressor constant, and by introducing equation (12) into the result, one obtains:

$$\left(\frac{\partial \ln |W_C|}{\partial \ln \theta_{12}}\right)_{h_1, s_1, m} = \frac{1}{\left(1 - \frac{\theta_{12}}{T_2}\right)} \quad (13)$$

The importance of these results should be noted: since both the initial working conditions of the compressor ( $W_C$  and  $\theta_{12}$ ) and the dissipation temperature parameter of flow 2 are known, it is possible to find out the increase of work supplied to the compressor in order to keep the mass flow and the outlet pressure ( $T_{d2} = T_2$ ) constant when an intrinsic malfunction characterized by a variation in the internal parameter  $\theta_{12}$  occurs. In addition, it is possible to obtain the next information:

- As the internal  $\theta$  parameter value of the compressor increases, it will be necessary to introduce a smaller additional amount of driving work (expressed as a percentage) in order to continue to produce the same amount of product, since a specific variation takes place in the internal  $\theta$  parameter (expressed as a percentage).
  - As the dissipation temperature parameter value of the output mass flow of the compressor,  $T_{d2} = T_2$ , decreases, it will be necessary to introduce a smaller additional amount of driving work (expressed as a percentage) in order to continue to produce the same amount of product, since a specific variation takes place in the internal  $\theta$  parameter (expressed as a percentage).
- b) One wishes to maintain the work supplied to the compressor constant, and by introducing equation (12) into the result, one obtains:

$$\left(\frac{\partial \ln m}{\partial \ln \theta_{12}}\right)_{h_1, s_1, W_C} = \frac{-1}{\left(1 - \frac{\theta_{12}}{T_2}\right)} \quad (14)$$

From this equation it is possible to obtain information of a similar kind as was from equation (13).

That is, for a compressor, the greater the internal parameter  $\theta$  and the lower the dissipation temperature parameter of its output mass flow,  $T_{d2}$ , the smaller will the influence of a specific intrinsic malfunction be.

These results can be applied to other components (Royo 1994); that is, it is possible from the knowledge of only  $\theta$  and  $T_{dj}$ , to assess the importance of the appearance of a specific intrinsic malfunction (characterised by a variation in the internal  $\theta$  parameter, as a percentage) in the components (percentage of fuel consumption increase or product decrease, if the amount of product or fuel are each maintained constant).

On reporting the amount of additional resources which a component needs to consume in order to produce a given amount of product, or the decrease in the amount of product when the amount of fuel is maintained constant, we are reporting the real cost increase that arises when a malfunction takes place. Nonetheless, due to the close relation between the different components of a plant, the problem becomes substantially more complicated when it comes to analysing a whole plant.

With the aim of showing the usefulness of  $\theta$  and  $T_d$  to analyse plants, in the following subsection we present the example of a power plant in which malfunctions will take place in some of the main components.

#### 4.3 Assessment of the importance of an intrinsic malfunction in a complex system

For this example, a power plant based on a Rankine cycle with reheating, like that shown in Figure 5 is considered (Royo et al. 1997). The inlet and outlet mass flow states of the components are shown in TABLE V.

The question posed is how the appearance of a certain intrinsic malfunction in one of the components (mainly the turbine and the pump) affects the overall performance of the plant:

$$\eta = \frac{W}{Q_i} = \frac{h_1 - h_2 + h_3 - h_5 - h_7 + h_6}{h_1 - h_7 + h_3 - h_2} \quad (15)$$

An analytic procedure based on the concepts of  $\theta$  and  $T_d$  will be used. This method, as has been seen above, is only strictly valid to analyse differential malfunctions; for this reason only small malfunctions will be considered (isentropic performance variations of 1%). However, it should be noted that it is uncommon in reality to find malfunctions of higher value than these, and for this reason the methodology presented does not involve a significant limitation. Before any analysis of malfunctions, whether it be numeric or analytic, it is necessary to know some basic aspects of the operation system and the behaviour of the plant:

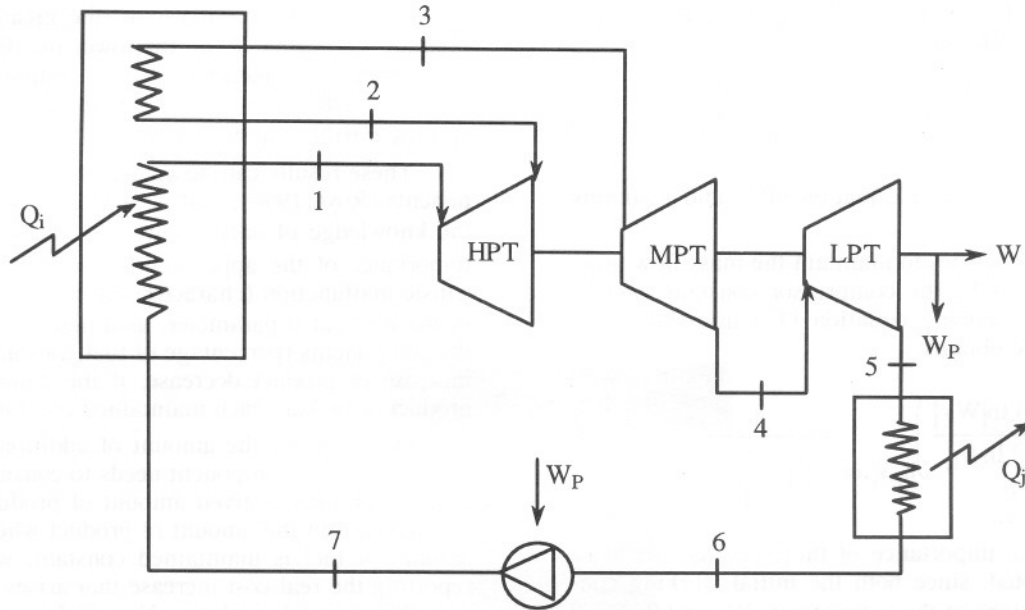


Figure 5. Power plant with reheating: components and interactions.

TABLE V. Power plant with reheating. Mass flows thermodynamic properties.

State	$P$ (bar)	$T$ (K)	$X$	$h$ (kJ/kg)	$s$ (kJ/kg K)	$m$ (kg/s)
1	166.00	812.5	----	3403.62	6.424	250.0
2	44.93	625.4	----	3087.71	6.525	250.0
3	40.44	813.8	----	3535.39	7.198	250.0
4	8.44	595.8	----	3102.79	7.288	250.0
5	0.069	311.9	0.939	2430.06	7.821	250.0
6	0.069	311.9	0	162.37	0.556	250.0
7	195.67	312.8	----	185.58	0.567	250.0

Let us suppose that even if an intrinsic malfunction takes place in one of the plant's components, the mass flow circulating through the plant remains constant (with the subsequent variation of the net power produced by the plant).

- The operation set point of the plant means that  $T_1$  and  $T_3$  remain constant.
- As regards  $P_1$ , it will be assumed that the plant is functioning in sliding pressure mode (high pressure turbine control valves widely open, i. e.  $A_1$  constant<sup>7</sup>), and that it is, therefore, variable.
- The vacuum pressure of the condenser ( $P_5 = P_6$ ) will be maintained at a fixed value,

even if the cooling system has to evacuate a greater amount of heat in some cases. Of course in state 6, there is always saturated liquid.

In order to simplify the problem and obtain expressions that are not too complex, it will be necessary to make a series of hypotheses on the behaviour of the plant:

- The internal  $\theta$  parameters of the remaining components will not be modified.
- Since  $m$  is constant, it will be assumed that  $P_3$  is only a function of  $P_2$ .
- Assuming that the behaviour of the three turbines satisfy Stodola's Ellipse Law (Cooke 1985), the following relations must be fulfilled<sup>8</sup>:

<sup>7</sup> For the sake of simplicity, it has not been considered malfunctions affecting the nozzle effective area of any of the turbines.

<sup>8</sup> It is assumed that the pressure relation in the low pressure turbine is greater than the critical one.

$$m = A_1 k_1 \frac{P_1}{\sqrt{T_1}} \sqrt{1 - \left(\frac{P_2}{P_1}\right)^2} \quad (16)$$

$$m = A_3 k_3 \frac{P_3}{\sqrt{T_3}} \sqrt{1 - \left(\frac{P_4}{P_3}\right)^2} \quad (17)$$

$$m = A_4 k_4 \frac{P_4}{\sqrt{T_4}} \quad (18)$$

The correctness of these hypotheses will be confirmed by comparing the analytic results with those obtained from a numeric simulation carried out using the GATE-Cycle (1995).

It is supposed first, that a modification in the low pressure turbine appears. This malfunction will be characterised by a variation in the internal parameter of the component,  $\theta_{\text{HPT}}$ .

According to the above, it is possible to state that:

- Since  $T_1$ ,  $T_3$  and  $m$  are constants and since they work in sliding pressure mode ( $A_1$  constant), it can easily be proven from expressions (16) to (18) that  $P_1$ ,  $P_2$  (and therefore  $P_3$ ),  $P_4$  and  $T_4$  will not vary.
- Since  $P_4$ ,  $T_4$  and  $\theta_{\text{LPT}}$  are constants,  $P_5$ ,  $P_6$  and  $x_5$  will also remain constant.

- Since  $P_1$ ,  $P_6$ ,  $\theta_P$  and  $m$  are constants,  $P_7$  and  $T_7$  will not be affected either.

- As  $P_2$  remains constant, according to equation (5),  $T_{d2} = T_2$ .

By differentiating equation (15) according to these hypotheses,

$$d\eta = \left(\frac{1}{Q} - \frac{1}{W}\right) \eta \, dh_2 \quad (19)$$

and

$$d\theta_{\text{HPT}} = \frac{\theta_{\text{HPT}}}{h_1 - h_2} \left(\frac{\theta_{\text{HPT}}}{T_2}\right) dh_2 \quad (20)$$

are obtained.

That is

$$\frac{d \ln \eta}{d \ln \theta_{\text{HPT}}} = \frac{\left(\frac{1}{Q} - \frac{1}{W}\right)(h_1 - h_2)}{\left(\frac{\theta_{\text{HPT}}}{T_2} - 1\right)} \quad (21)$$

This equation enables the variation of the performance of the plant to be quantified analytically when an intrinsic malfunction occurs in the high pressure turbine.

TABLE VI. Variation of the overall efficiency of plant in Figure 5 when a components malfunction takes place. Parameter ( $\partial \ln \eta / \partial \ln \theta$ )

Intrinsic malf.	Analytic procedure	GATE-Cycle	Discrepancy
HPT	$\frac{\left(\frac{1}{Q} - \frac{1}{W}\right)(h_1 - h_2)}{\left(\frac{\theta_{\text{HPT}}}{T_{d2}} - 1\right)} = 0.0232$	0.02316	0.02 %
IPT	$\frac{\frac{1}{W} \left(1 - \frac{\theta_{\text{LPT}}}{T_{d4}}\right)(h_3 - h_4)}{\left(1 - \frac{\theta_{\text{MPT}}}{T_{d5}}\right) \left(1 - \frac{\theta_{\text{LPT}}}{T_{d5}}\right)} = 0.0182$	0.0176	3.4 %
LPT	$\frac{\frac{1}{W} (h_4 - h_5)}{\left(1 - \frac{\theta_{\text{LPT}}}{T_{d5}}\right)} = 0.095$	0.0963	1.3 %
Pump	$\frac{\left(\frac{1}{Q} - \frac{1}{W}\right)(h_7 - h_6)}{\left(1 - \frac{\theta_B}{T_{d7}}\right)} = 0.00193$	0.00192	0.52 %

From the values shown in TABLE IV and equation (21), it can be obtained that, for the case being analysed:

$$\frac{d \ln \eta}{d \ln \theta_{\text{HPT}}} = 0.0232$$

In other words, an increase in  $\theta_{\text{HPT}}$  of 5.6% (1% decrease of the  $\eta_{\text{s,HPT}}$ ) affects the whole plant with a performance decrease of 0.13%.

The behaviour was simulated by means of a commercial simulator (GATE-Cycle 1995). The analysis of the steam turbines in this simulator is based on the research work (obtained from experimental data and field measures) developed by Spencer, Cotton and Cannon (1974). The result obtained is:

$$\frac{d \ln \eta}{d \ln \theta_{\text{HPT}}} = 0.02316$$

That is, there is a discrepancy between the values obtained by the two procedures (analytic and numeric) of only 0.02%. This fact indicates that the simplifying hypotheses are accurate. This methodology can be applied to other plant components (Royo et al. 1997), obtaining the results shown in TABLE V<sup>9</sup>.

It should be noted that, in order to perform this analysis it was only necessary to know the situation of the plant prior to the malfunction, and some basic details of its behaviour. It has been observed that the analytic results of this analysis coincide to a large extent with those obtained from simulation with GATE-Cycle. In all cases, the discrepancy between the two procedures is lower than 4%. However, the analytic procedure here shown would provide a better understanding of the phenomena that takes place in the plant.

## 5. Conclusions

From the observation of the trajectories followed in an h-s diagram by mass flows going through a specific component, the process trajectory equations have been formulated, which has enabled to define the new internal  $\theta$  parameters of the components. These internal parameters are capable of describing the thermodynamic processes which take place in these components in the same way as any of the conventional internal parameters which are normally used but having important additional advantages.

In relation with  $\theta$  and the malfunctions,

<sup>9</sup>  $T_{d4} = 662.2 \text{ K}$ ,  $T_{d5} = T_5$  and  $T_{d7} = T_7$  (Royo et al. 1997b)

another parameter, the dissipation temperature parameter of each mass flow of a thermal plant, has been defined and characterised. From this parameter, together with the internal  $\theta$  parameter, it is possible to make an analytic analysis of the influence of the appearance of an intrinsic malfunction on the overall performance of the plant of which it is a part.

It should be noted that, in order to perform this analysis it was only necessary in all cases to know *the state thermodynamic of the plant prior to the malfunction, and some basic details of its behaviour.*

Although, the variations of the internal  $\theta$  parameters of the components from the point of view of malfunctions have been dealt with here, the results obtained may also be used (with the necessary modifications) for analytical optimisation studies in the design of thermal plants, in which such internal  $\theta$  parameters are decision variables of the problem.

Besides these important applications of  $\theta$  and  $T_d$ , the dissipation temperature parameter is the starting point for the definition of a new thermodynamic function (the free relative energy), which is very important for the study of malfunctions. (Valero 1991, Valero and Lozano 1992, Valero and Royo 1992, Royo 1994).

## Nomenclature

A	Nozzle effective area
h	Specific enthalpy of a flow
k	Generic constant
m	Flow rate
P	Pressure
Q, $\bar{Q}$	Heat transfer rate, same per unit mass
$r_p$	Pressure ratio
s	Specific entropy of a flow
T	Temperature
$T_d$	Dissipation temperature parameter
W, w	Power, same per unit mass
x	Quality
$\gamma$	Specific heat ratio ( $c_p/c_v$ )
$\eta$	Efficiency
$\eta_s$	Isentropic efficiency
$\theta$	Internal parameter

## Subscripts

C	Concerning the compressor
H	Concerning the heater
HPT	Concerning the high pressure turbine
i	Concerning an entry flow of a compo-

	ment
j	Concerning an outlet flow of a component
LPT	Concerning the low pressure turbine
MPT	Concerning the intermediate pressure turbine
P	Concerning the pump
T	Concerning the turbine

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