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Design of a fractional order *PI* controller for steam pressure in the steam drum of a bagasse fired boiler

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Abstract: This paper proposes a fractional-order *PI* controller for controlling the steam pressure in the steam drum of a bagasse fired boiler. The dynamics behavior of this process was experimentally identified. This identification procedure yielded an equivalent third order plus time delay model, and showed wide process static gain variations. We therefore propose a new methodology for the design of fractional-order robust controllers for this class of processes. It was shown that the attained controllers significantly outperformed the robustness achieved with current *PI* controllers.

1. INTRODUCTION

The sugar cane processing industry has played an important role in many developing countries where agricultural activeties provide the best potential for labor absorption in rural areas. Traditional sugar cane factories are characterized by their high energy consumption and pollution into the environment, largely due to the low-efficiency technology they have (Hugott, 1996). At these factories, bagasse fired boilers constitute the plants that represent the principal energy losses (Rein, 2007). That is why in the sugar cane processing industry (as a result of the increase in the cost of fuels, high competitiveness in energy markets and strong environmental demands) researches aimed at increasing energy efficiency as well as reducing the impact of the industrial activity on the environment have a high relevance and scientific-technical importance (De Prada Moraga et al., 2009).

Bagasse fired boilers constitute vital plants for the sugar cane processing industry because they guarantee energy production in a relatively economic way through the use of bagasse as fuel, which represents waste coming from the industry itself (Rivas-Perez, 2011b). These boilers are characterized by a complex dynamic behavior: time-varying parameters, as well as several interacting processes, that usually are controlled independently by *PID* controllers (Astrom and Bell, 2000; Dukelow, 1991; Rivas-Perez et al., 2014c, 2012; Rodriguez-Vazquez et al., 2008).

A high efficient operation of this class of boilers requires (Hugott, 1996): 1) guarantee that the amount of bagasse necessary for keeping steam pressure within required limits is burn, independently from the load variations; 2) keep a correct air/bagasse ratio, which enables complete combustion within the design limits of the boiler. These requirements cannot be guaranteed if there is not an effective steam pressure control in the steam drum of the boiler (Dukelow, 1991; Rivas-Perez, 2014c, 2011b).

In the sugar cane processing industry it is required that the bagasse fired boilers operate at constant pressure (Hugott, 1996). For this purpose several strategies for steam pressure control in the steam drum of these boilers have been proposed, the most popular and widespread of which are based in conventional *PID* controllers (analog or discrete) due to the simplicity of their structure, flexibility, simple tuning, as well as their general robustness properties (Dukelow, 1991; Hogg and El-Rabaie, 1990). However, some studies have shown that simple *PID* controllers do not perform well when the dynamic behavior of these boilers is characterized by time delay, time-varying parameters and unmeasured disturbances (Guin et al., 1989; Dong et al., 2005; Gerliga et al., 1992; Rivas-Perez et al., 1998). As a result, a large steam pressure settling time in the top drum is obtained, which leads to an excessive fuel consumption and consequently, to an inadequate combustion and a great environmental pollution. Consequently, any controller that is designed should be characterized by showing an adequate robustness in face of this class of dynamic behavior (Rivas-Perez et al., 2000).

Furthermore, during the last few years and as a result of a better understanding of the fractional order calculus, fractional order operators have been applied with satisfactory results to model and control processes with complex dynamical behaviors (see e.g. Calderon-Valdez et al., 2010; Castillo-Garcia et al., 2013a, 2011; Chen et al., 2004; Feliu-Batlle et al., 2011; Monje et al., 2010; Podlubny, 1999; Sanchez-Rodriguez et al., 2008; Vinagre et al., 2007).

The concept of extending classical integer order calculus to non-integer order cases is by no means new. For example, it was mentioned in (Podlubny, 1999) that the earliest systematic studies seem to have been made in the beginning and middle of the 19th century by Liouville, Riemann, and Holmgren. The significance of application of fractional order operators in the design of control systems is that it is a generalization of classical integer order control theory, which could lead to more adequate modelling of the processes and better performance of the control systems.

The fractional order basic operator is represented as ${}_aD_t^\alpha$ where a and t are the limits and $\alpha(\alpha \in \mathcal{R})$ is the order of the operation. When $\alpha > 0$ (positive values) this means a fractional derivative, and when $\alpha < 0$ (negative values) this means a fractional integral. In the Laplace domain, this operator corresponds with a fractional-order differentiator and/or integrator s^α (provided initial conditions are zero) and therefore, the frequency characteristic of this operator is $(j\omega)^\alpha$ (Monje et al., 2010).

Both qualitative behavior and the robustness of conventional PID controllers can be sensibly improved by means of their generalization to a $PI^\alpha D^\lambda$ fractional-order controllers involving an integrator of order α and a differentiator of order λ (Podlubny, 1999). Fractional order controllers ($PI^\alpha D^\lambda$) have consequently also been proposed and have received considerable attention (see e.g. Castillo-Garcia et al., 2013b; Feliu-Batlle et al., 2013; Monje et al., 2010; Podlubny, 1999; Rivas -Perez et al., 2014a; etc.). One interesting feature of fractional order controllers is that they exhibit certain advantages when designing robust control systems in the frequency domain for processes whose parameters vary in a large range. Consequently, the qualitative behavior, as well as the robustness of industrial PID controllers used in the control of the steam pressure in the steam drum of bagasse fired boilers, can be improved through the design of $PI^\alpha D^\lambda$ controllers.

The objectives of this paper are: (a) to derive a systematic and analytic design method for fractional order PI controllers (FPI), which will guarantee a minimum performance when the steam pressure dynamic alters as a result of calorific value variations; (b) to carry out a comparative study by means of computer simulations of the robustness of this FPI controller with equivalent conventional controllers (PI and PID).

The main contribution of this paper consist on the proposal of a fractional order controller (FPI) to control the steam pressure in the steam drum of a bagasse fired boiler with very satisfactory results, thus providing a practical solution to the complex problem of designing of effective controllers with uncertainties models of bagasse fired boiler dynamics and verifying that the fractional order controllers can improve the performance of conventional controllers (PI and PID) in this class of applications. In this paper, all the modelling and control methodologies have been carried out for a real industrial bagasse fired boiler, whose nominal dynamics and ranges of parameter variations have been experimentally determined.

This paper is organized as follows. The structure and characteristics of our bagasse fired boiler are introduced in Section 2. A new linear model of the steam pressure in the steam drum of this boiler is also proposed in that Section. A method to design FPI controllers for the process under study is offered in Section 3. Section 4 summarized the simulation results for the nominal process and when the model dynamic parameters change in the entire range of variation. Finally, some conclusions are drawn in Section 5.

2. SYSTEM DESCRIPTION AND DYNAMIC MODEL

2.1. System description

The basic objective of this class of systems consist in transforming energy contained in the bagasse, through combustion, into available thermal energy and transferring it to the water to generate steam at specific pressure and temperature, in agreement with the operation conditions of the boiler (Hugott, 1996). This steam, henceforth, is used to generate mechanical and electric energy, or as such, to feed other equipment and processes of the industry itself. The study presented in this paper is based on the bagasse fired boiler of the sugar cane processing industry Espartaco in province Cienfuegos, Cuba. This boiler is water tubes and can generate up to 80 T/h of steam with a maximum pressure of 30 Kg/cm^2. The steam drum is the upper drum of the boiler where the separation of water and steam occurs.

Bagasse is a fuel of varying composition, consistency, and calorific value. These characteristics depend on the climate, type of soil upon which the sugar cane is grown, variety of sugar cane, harvesting method, amount of sugar cane washing, and the efficiency of the milling plant (Rein, 2007). Bagasse, after being subjected to a process of drying and crushing, through a conveyor mat is routed to the rotary feeders (known as bagasse feeders) which are the ones that introduce the necessary bagasse into the furnace for its combustion. This class of feeders has a motor that turns two rollers, whose rotation speed is proportional to the bagasse mass flow that is inputted into the furnace. The steam pressure required in the steam drum of the boiler determines the bagasse mass flow that is inputted into the furnace and consequently, the speed of the motors of rotary feeders. The speed of motors in these devices is regulated through frequency variators and can reach from 7 rpm at 25 Hz up to 17 rpm at 60 Hz. The nominal operation steam pressure in the steam drum of the boiler, ascending to 23 kgf/cm^2, is reached with the combustion of a nominal bagasse flow of 8,000 Kg/h.

2.2. Dynamic model

The steam pressure in the steam drum of a bagasse boiler is one of the main variables to control in this class of plants because it is an indicator of energy balance between the generated and demanded steam. This pressure is proportional to the generated steam and constitutes the output energy, while the bagasse flow represents the input energy (Rein, 2007). Therefore, the mathematical model to obtain will have as the output variable, the steam pressure variation in the steam drum of the boiler ($\Delta y(t)$) and as input variable, the % of the speed variation of bagasse feeder motors into the furnace ($\Delta u(t)$)). The % of moisture content in the bagasse which enters the furnace constitutes the fundamental disturbance ($D(t)$)) that affects this process (Rivas-Perez et al., 1994).

In order to obtain the mathematical model of the steam pressure in the steam drum of the bagasse boiler under study, a system identification procedure was used (Pedregal et al., 2009; Rodriguez-Vazquez et al., 2006). The identification experiment was developed so as to, initially lower the steam pressure in the steam drum of the boiler up to a value that does not affect the turbine operation (19.3 kgf/cm^2) and then through a step signal carrying this variable to its nominal operating value (23 kgf/cm^2). Thus, the mathematical model to obtain of the steam pressure will represent the nominal dynamic behavior of such variable (nominal plant).

The speed of the bagasse feeder motor received an increment $\Delta u(t)$) of 10%. The data of the steam pressure variation $\Delta y(t)$), along with those of the increment of the speed of the bagasse feeder motors $\Delta u(t)$) were registered and stored in a computer. The experimental response of the nominal process to a step command is shown in Fig. 1.

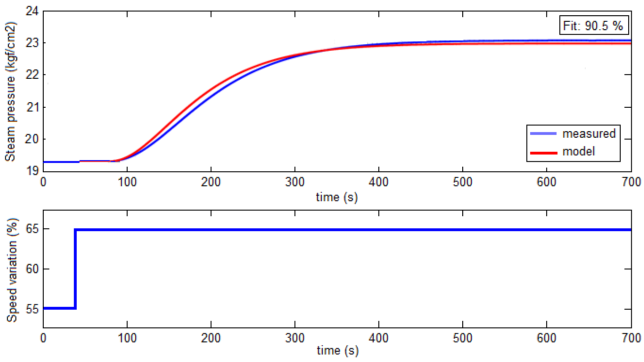

Fig. 1. Experimental response to step command of the nominal plant.

The obtained response can be described by a third-order transfer function with a time delay given by:

$$G(s) = \frac{\Delta y(s)}{\Delta u(s)} = \frac{K}{(T_1 s + 1)(T_2 s + 1)(T_3 s + 1)} e^{-\tau s}, \quad (1)$$

where K is the static gain, T_1, T_2, and T_3 are the time constants, and τ is the time delay. When the steam pressure in the steam drum of the boiler corresponds with the nominal operating regime, the nominal values of the parameters of model (1) are obtained. The nominal model is denoted as $G_0(s)$, and its parameters as K_0, T_{10}, T_{20}, T_{30} and τ_0. The estimated values of these parameters are K_0= 0.37 kgf/cm^2%, T_{10} = 80.4 s, T_{20}= 63.8 s, T_{30}= 35.2 s and τ_0= 50 s.

Validation results of linear model (1) with the parameters' estimated nominal values (nominal model) are also shown in Fig. 1. This figure shows a good agreement between the results obtained from the step test and the predictions provided by the nominal linear model (1).

The calorific value is the most important property of the fuels and is defined as the amount of heat $Q(t)$ produced by the complete combustion of a fuel measured in units of energy per amount of material (Rein, 2007). In the case of bagasse, the calorific value depends mainly of the % of moisture content and of cane fiber (cellulose) content in the bagasse (Hugot, 1996). The calorific value of bagasse is 19,250 kJ/kg at 0% moisture and 9,950 kJ/kg at 48% moisture. Hence, the % of moisture content of bagasse is the most significant parameter that determines its calorific value. Thus, the higher the moisture content, the lower the bagasse calorific value. A good milling process will result in low bagasse moisture content. On the other hand, the increase in the fiber content in the sugar cane raises the fiber content in the bagasse and therefore this implies an increase in the calorific value. During a sugar harvest and as a result of complex agro-industrial process of the sugar cane transformation, both the % of moisture content as well as the fiber content in the bagasse show wide variations, which results in variations in the calorific value $Q(t)$ of the bagasse in the operation range $[Q_{min}, Q_{max}]$. These variations affect the dynamic behavior of the steam pressure in the steam drum of the boiler.

Developing more real-time experiments in the steam pressure of our bagasse fired boiler, and using a robust system identification procedure (Rivas-Perez et al., 2014b, 2011a, 2008), it was shown that variations of the calorific value in the operation range $[Q_{min}, Q_{max}]$ originate variations in the static gain of the mathematical model (1) in the range $[K_{min}, K_{max}]$:

$$0.18 \le K \le 1.11. \quad (2)$$

For this reason, any controller that is designed to control the steam pressure in the steam drum of the boiler under study must guarantee a priori a specific level of minimum performance in the whole range of variation of the dynamic parameter (model uncertainties) of the mathematical model (1).

3. DESIGN OF THE FPI CONTROLLER

As it was shown in the previous section, when the calorific value of the bagasse varies in the operation range $[Q_{min}, Q_{max}]$ the static gain of the steam pressure process in the steam drum of the boiler under study vary in a wide range (2). This leads to the performance of *PI* controllers, designed for nominal operating conditions, degrade rapidly. The approach developed in this paper is to design a new controller that exhibits the same performance as a conventional *PI* controller against nominal dynamic behavior of the process under study, that is designed to the same specifications, but that is the least sensitive to variations in the dynamic parameters of the process, hence more robust. Therefore, our objective is to design a controller that behaving like a *PI* controller under nominal operating regime, provides a response that deteriorates less when the calorific value of bagasse presents variations in the operating range $[Q_{min}, Q_{max}]$. Our novel proposal is to design a *FPI* controller, where the additional degree of freedom that allows increasing the robustness and achieving behaviors in the predetermined frequency domain is given by the fractional order α of the controller.

3.1. Control objectives

Taking into account the previous considerations, we propose as control objectives: 1) a small overshoot in the nominal process response, 2) the double settling time of the open loop one, and 3) a guaranteed minimum level of performance in all the range of variation of static gain. The controller design is developed based on the obtained nominal model of the process under study:

$$G_0(s) = \frac{0.37}{(80.4s+1)(63.8s+1)(35.2s+1)} e^{-50s}, \quad (3)$$

using the following specifications in the frequency domain: a) a phase margin (ϕ_{m0}), which guarantees the desired damping with the nominal process. The maximum overshoot should be $M_p \approx 5\%$; b) a crossover frequency (ω_{c0}), which provides the required settling time, while guaranteeing closed loop stability in all the range of static gain variation ($K \in [0.18, 1.11]$). In accordance with the operating conditions, the closed loop control system requires a settling time $t_s \approx 874$ s; c) zero steady state error to a step command, which implies that the controller must include an integral term. Figure 2 shows the block diagram of the control system of the steam pressure in the steam drum of bagasse fired boiler that is proposed. This diagram considered the disturbance $D(s)$ (% of moisture content in the bagasse), which is modeled by a step signal passing through a first order filter with a time constant $T_4 = 17.5$ s.

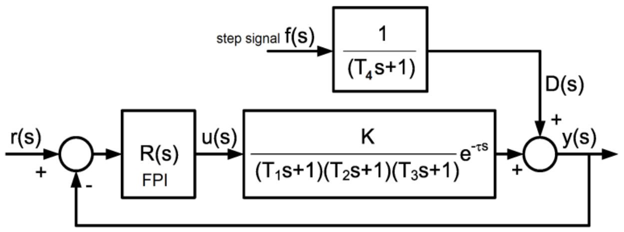

Fig. 2. Block diagram of the proposed control system.

The expressions that define the phase margin and gain crossover frequency are (Monje et al., 2010):

$$|R(j\omega_c)G_0(j\omega_c)|_{dB} = 0dB, \quad (4)$$

$$arg\{R(j\omega_c)G_0(j\omega_c)\} = -\pi + \phi_m. \quad (5)$$

3.2. FPI controller design

The fractional order $PI^\alpha D^\lambda$ controller, which has an integrator of order α and a differentiator of order λ, in the time domain can be represented by the expression (Podlubny, 1999):

$$u(t) = \hat{K}_p\left(e(t) + \frac{1}{\hat{T}_i}D_t^{-\alpha}e(t) + \hat{T}_d D_t^{\lambda}e(t)\right), 0 \le \beta, \lambda \le 1, \quad (6)$$

This controller can improve the performance of the control systems of the steam pressure in the steam drum of the boiler because it has 2 more tuning parameters than the conventional PID. Particularizing controller (6) for $\lambda = 0$, we obtain:

$$u(t) = K_{p1}e(t) + K_{i1}D_t^{-\alpha}e(t), 0 < \alpha \le 2, \quad (7)$$

where $K_{p1} = \hat{K}_p$ and $K_{i1} = \frac{\hat{K}_p}{\hat{T}_i}$.

Controller (7) is denoted as FPI and its transfer function can be represented as:

$$R_{FPI}(s) = K_{p1} + \frac{K_{i1}}{s^\alpha} = \frac{K_{i1} + K_{p1}s^\alpha}{s^\alpha}. \quad (8)$$

This class of controllers require the tuning of three parameters K_{p1}, K_{i1} and α, that is one more parameter than in the case of the conventional PI controller. The fractional order parameter α is used to fulfill additional specifications of robustness performance of the controller (Monje et al., 2010). The fractional order differentiator s^α in the frequency domain can be denoted as:

$$s^\alpha = (j\omega)^\alpha = \omega^\alpha\left(\cos\left(\frac{\pi}{2}\alpha\right) + j\sin\left(\frac{\pi}{2}\alpha\right)\right) = \omega^\alpha e^{j\frac{\pi}{2}\alpha}. \quad (9)$$

Therefore, considering (9), the transfer function of the FPI controller (8) is represented by:

$$R_{FPI}(j\omega_c) = \frac{K_{i1} + K_{p1}\omega^\alpha e^{j\frac{\pi}{2}\alpha}}{\omega^\alpha e^{j\frac{\pi}{2}\alpha}}. \quad (10)$$

The design specifications a) and b) given by a certain phase margin (ϕ_{m0}) and a gain crossover frequency (ω_{c0}) can be expressed compactly by the complex tuning equation of controllers (Monje et al., 2010):

$$R_{FPI}(j\omega_c)G_0(j\omega_c) = e^{-(\pi-\phi_m)j} = -e^{j\phi_m}, \quad (11)$$

where $G_0(j\omega_c)$ represents the dynamic behavior of the nominal plant (3). From (11), considering (10) yields:

$$R_{FPI}(j\omega_c) = \frac{K_{i1} + K_{p1}\omega^\alpha e^{j\frac{\pi}{2}\alpha}}{\omega^\alpha e^{j\frac{\pi}{2}\alpha}} = \frac{-e^{j\phi m}}{G_0(j\omega_c)}. \quad (12)$$

The tuning parameters of the FPI controller are obtained from the expression (12):

$$K_{i1}(\alpha) = \frac{1}{\sin\left(\frac{\pi}{2}\alpha\right)}\Im\left\{\frac{-\cos(\phi_m) - j\sin(\phi_m)}{\omega_c^\alpha G_0(j\omega_c)}\right\}. \quad (13)$$

$$K_{p1}(\alpha) = \Re\left\{\frac{-\cos(\phi_m) - j\sin(\phi_m)}{\omega_c^\alpha G_0(j\omega_c)}\right\} - \frac{1}{\tan\left(\frac{\pi}{2}\alpha\right)}\Im\left\{\frac{-\cos(\phi_m) - j\sin(\phi_m)}{\omega_c^\alpha G_0(j\omega_c)}\right\} \quad (14)$$

where $\Re\{\}$ and $\Im\{\}$ represent real and imaginary components of a complex number, respectively.

The parameter that has the highest influence on the stability of the closed loop control system is the static gain. The results presented in the previous section show that this parameter can have variations of up to 2.5 times the nominal value ($\Delta K \approx 2.5K_0$). Therefore, the fractional order parameter α of FPI controller will be used to improve the robustness of the control system in closed loop, in the sense of stability against static gain variations, i.e. to ensure that the gain margin (M_g) of the control system reaches its maximum value. Figure 8 shows the obtained gain margin as function of the fractional integral action, α.

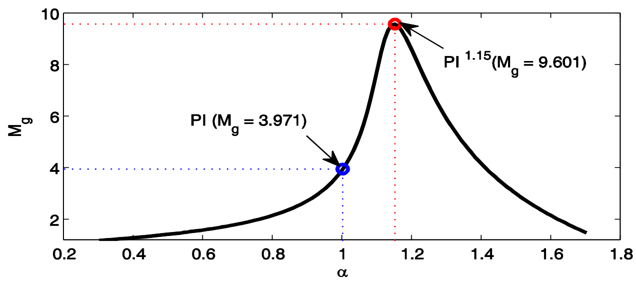

Fig. 3. Gain margin as function of α : PI vs PI^α.

From this figure it is observed that the maximum value of gain margin $M_g = 9.601$ dB is achieved for $\alpha = 1.15$. Note that the gain margin obtained for $\alpha = 1$ (standard PI controller) is $M_g = 3.971$ dB, so that has been achieved an improvement of approximately 241%. As a result of the operating requirement that the closed loop control system has the same settling time that the open loop system, the gain crossover frequency was chosen as $\omega_c = 0.00128$ rad/s. A phase margin $\phi_m = 61.49°$ was taken, which approximately corresponds to $M_p \approx 5\%$ and it is within the standard value range for this specification. From the expressions (13)-(14) and considering the two specification ω_c y ϕ_m and $\alpha = 1.15$, the values of the tuning parameters of the FPI controller are obtained:

$$R_{FPI}(s) = \frac{0.5374 + 0.0022s^{1.15}}{s^{1.15}}. \quad (15)$$

The values of the tuning parameters of the PI controller for the nominal plant, obtained under the same specifications in the frequency domain are the followings:

$$R_{PI}(s) = \frac{-0.1257+0.055s}{s}. \quad (16)$$

4. SIMULATIONS

4.1. Simulation setup

Simulations have been carried out using *MATLAB*. The model continuous transfer function has been converted to its discrete state space model. The fractional-order action of the controllers has been implemented by means of its Grünwald-Letnikov approximation (without any series truncation) in order to obtain accurate results (Podlubny, 1999). The Grünwald-Letnikov coefficients have been determined before to run the simulation loop. On every loop iteration control signal is determined by means of the aforementioned coefficients and introduced to the discrete state space model in order to obtain the system output. All simulations have been executed with a sample time of 1 s.

4.2. Simulation results

In order to verify the advantages of the designed FPI controller to control the steam pressure in the steam drum of the boiler under study in comparison with the current PI controller installed in said boiler, as well as its robustness against variations of plant dynamic parameters, different simulations were performed with both controllers under the same operating conditions. Fig. 4 shows the comparative simulation results of time responses of the control system with the nominal plant and controllers FPI/PI respectively.

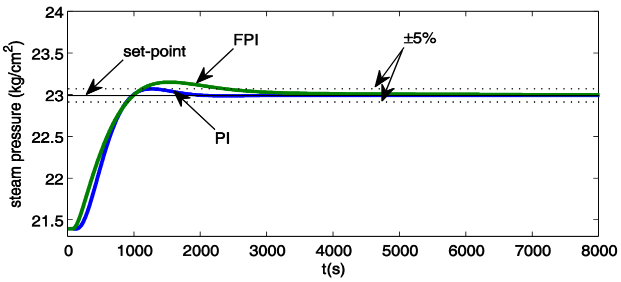

Fig. 4. Time responses of the control system with nominal plant and FPI/PI controllers.

From Fig. 4 it is observed that when there are no variations in the static gain of the plant the time responses of the control system obtained with both controllers are very similar.

Fig. 5 and Fig. 6 shows the simulation results of the time responses of the control system with PI and FPI controllers, when originating variations in the static gain of the plant in the operating range $[K_{min}, K_{max}]$. From these figures it can be observed that when the static gain of the plant increases, and the PI controller (16) is used, the time response of the control system notoriously deteriorates and for $K = K_{max}$ the system is near to be unstable. Since, if the FPI controller (15) is used the time response suffers much less deterioration and remains stable in the whole range of variation of the static gain.

5. CONCLUSIONS

The design of a FPI controller for effective control of the steam pressure in the steam drum of bagasse fired boiler in operation, which behaves more robust than other equivalent PI controller (in the sense of exhibiting both the same closed-loop dynamic behavior for the nominal plant specifications) against variations of the plant static gain was developed. These results present a special relevance in bagasse fired boilers whose dynamic parameters exhibit a wide variation when the calorific value varies in the operating range $[Q_{min}, Q_{max}]$.

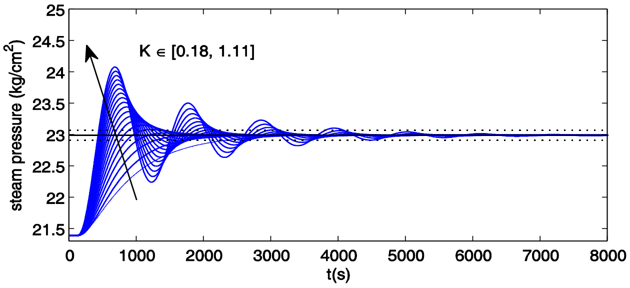

Fig. 5. Time response of the control system with PI controller and variations in the plant static gain in the operating range.

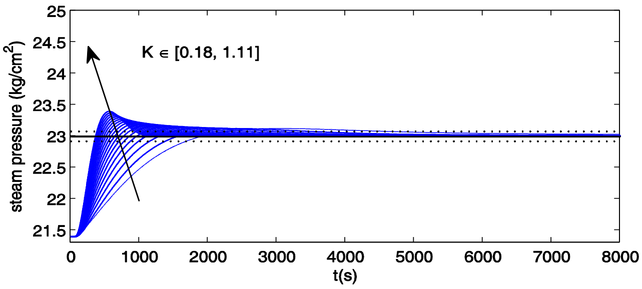

Fig. 6. Time response of the control system with FPI controller and variations in the plant static gain in the operating range.

The following objectives of our research consists precisely in the practical implementation of the designed FPI controller based on a *PLC SIMATIC S7-300* in the bagasse fired boilers of the sugar cane processing industry Espartaco. The benefits obtained with increasing of robustness and effectiveness in the control of steam pressure in the steam drum of the boiler immediately are reverted into an increase in energy efficiency as well as in greater environmental protection by reducing the load of gases and pollutants particles.

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