A NOVEL TOOL FOR MULTI-MODEL PID CONTROLLER DESIGN

Flávio Faccin and Jorge O. Trierweiler

Group of Integration, Modeling, Simulation, Control and Optimization of Processes (GIMSCOP)
Department of Chemical Engineering, Federal University of Rio Grande do Sul (UFRGS)
Rua Duque de Caxias, 1303 apt. 603. CEP: 90010-283 - Porto Alegre - RS - BRAZIL
Fax: +55 51 3316 3277, Phone: +55 51 3316 4072
E-MAIL: {faccin, jorge}@enq.ufrgs.br

Abstract: In this paper, it is shown how a robust performance PID controller can be synthesized for a set of possible linear models. The method is carried out by solving a min-max optimization problem formulated in frequency domain. As the synthesis of a low-order controller for a high-order plant is in general nonconvex, the original optimization problem is divided into two convex optimization subproblems, which are solved iteratively until to achieve the complete convergence. These subproblems are constructed using a general two degree-of-freedom control configuration, in which all commercial industrial PID controllers can be converted.

Keywords: PID controllers, frequency response methods, multi-objective optimization, nonlinear control systems, uncertain linear systems.

1. INTRODUCTION

There is no doubt that the PID algorithm is the most widely used controller in the industrial process control systems of the whole world. It is a robust easily understood algorithm that can provide good control performance despite the varied dynamic characteristics of process plants. Since the creation of PID controllers, roughly sixty years ago, several PID tuning methods were and are being proposed. These methods can be classified in empirical (e.g., Ziegler-Nichols), analytical (e.g., direct synthesis), or based on some kind of optimization (e.g., ITAE optimization criterion). But all of them have at least one gap or a deficiency. Usually, they work for only some special conditions producing an unsatisfactory result for some classes of plant behavior.

In order to fill these gaps, this paper presents a novel tool, which intends to help the PID design. It works with a multi-model system (different linear models representing a nonlinear system, or a set of parameter uncertainties derived from a fixed linear model), and designs the parameters (for any kind of PID) based on an optimization problem to bring all the closed loop systems as close as possible to the same attainable closed loop performance.

2. MYTHS AND FACTS ABOUT PID CONTROLLERS

According Dorf and Bishop (1998), "the popularity of PID controllers can be attributed partly to their robust performance in a wide range of operating conditions and partly to their functional simplicity, which allows engineers to operate them in a simple, straightforward manner". Skogestad (2003), however, believes that "although the proportional-integral-derivative (PID) controller has only three parameters, it is not easy, without a systematic procedure, to find good values (settings) for them. In fact, a visit to a process plant will usually show that a large number of the PID controllers are poorly tuned".

In Åström and Hägglund (1995), an interesting and amazing study was reported, which concludes, "a typical mill has more than 2000 control loops and that 97% use PI control. Only 20% of the control loops were found to work well and decrease process variability. Reasons for poor performance were poor tuning (30%) and valve problems (30%). The remaining 20% of the controllers functioned poorly for a variety of reasons such as: sensor problems, bad choice of sampling rates, and anti-aliasing filters".

In another study also reported by the same authors it was claimed that "30% of installed process controllers operate in manual, that 20% of the loops use 'factory tuning', i.e., default parameters set by the controller manufacturer, and that 30% of the loops function poorly because of equipment problems in valves and sensors".

Due to the enormous amount of PID controllers do not working in a suitable and satisfactory manner, some control systems have packages for automatic controller performance assessment. This study area is relatively young, but very important, since even the most good tunings, with the time, present a deteriorated performance, that occurs because natural and continuously variations in the process dynamics, as discussed by Kempf (2003) and Harris et al. (1999). Although this powerful technique can warn when a loop shows a poor performance, it must be combined to a design method, which is used to retune the control loop.

After this review, one question needs to be answered: how this so simple, so studied, so old and the most used controller in the world is *not* working satisfactorily in the majority of the cases? Regardless of technical trouble (as sensor and valves), which is an instrumentation matter, the most important factor is how to tune the controller in a suitable manner, despite of the enormous amount of design methods, proposed throughout the last six decades. All these methods are compiled in special books about PID controllers, such as O'Dwyer (2003), Åström and Hägglund (1995) or in any textbooks about process control.

Actually, the PID tuning is only so easy for simple dynamics, as sluggish systems, when just a PI controller is already enough. To more complicated systems, the derivative action is needed to improve the closed loop performance, and the process of tuning becomes much more difficult. According Piazzi and Visioli (2002), "for many processes the derivative term of the controller is not useful and it is often difficult to tune, so that practitioners prefer to avoid its use". Luyben (2001) and Ingimundarson and Hägglund (2002) also agree with this idea. Isaksson and Graebe (2000) have including attracted attention to the fact that "there is an industrial myth that derivative action does not work".

In general, a control engineer working in a process plant is very busy and he does not want to have more work to do, so at first he tries to tune a provisional PI with simple memorized rules. If the control loop has a fairly good performance, he does not spend more time with this loop and the provisional PI becomes a permanent controller. This is one reason for 20% of the controllers use 'factory tuning', and the majority of them are PI. In fact, the derivative action tuning is normally much more complicated and implicates in the use of additionally filters, when the system noise is significant.

In addition, due to the existence of many different types of parameterization (specially in commercial control systems) when a design method is used, the results obtained may not be compatible with the industrial controller. Goodwin et al. (2001) have already pointed out "caution must be exercised when applying PID tuning rules, as there are a number of possible parameterizations". If a practitioner tries to use a design method, which is not compatible with the existing in the control system, the result may not be satisfactory, and then he gives up using the derivative action and even thinks that the method is not good. It helps to keep the 'myth'.

Another important factor is that in spite of the advanced techniques in PID tuning, the nearly whole amount of these methods were done for *ideal parameterizations*, without derivative action filter and passing the derivative action through the error signal. In practice, the industrial use of these parameterizations is not conceivable, due to the strong control action at high frequency produced by measurement noise and suddenly setpoint changes. Some of them consider the derivative filter action, but pretty few use the derivative action passing just at the process variable signal, even the most modern optimization techniques. This means that the results of these methods are not completely satisfactory for industrial uses. One exception is presented in Carotenuto et al. (2002), which uses a PID controller with derivative action on the filtered output and a global optimization method, but it is not able to work with multi-model systems, and it claims that the method convergence for a PID is *more difficult* to attain, due to the complexity of the method.

The problems above quoted are only related to the PID algorithm. Up to now the process worries were not taken into account, but of course they need to. A common tuning method is normally model-based, or in some cases, based in some measured characteristics of the system (such as: ultimate gain, Ku, and period, Pu), but in all of them only one operating region is considered. It is well known that the industrial processes are nonlinear at some extension. In spite of the intrinsic process nonlinearities, usually the dynamic behavior can be satisfactory approximated by a linear model at each operating point. Therefore, if the process works in several operating points, a set of linear models can be constructed to represent the system behavior. Even if the process is linear, during the identification procedure, usually more then one linear model is identified, or an uncertain bound is given for the identified parameters, what again produce a set of linear models. Another source for multi-model representation is time variant systems, what often occurs in the process industry, since during the process operation the dynamic behavior can change due to the equipment fouling, impurities level in the row materials, etc. All these situations can be well and easily described by a multi-model approach (i.e., a set of possible linear models).

3. MULTI-MODEL APPROACH (MMA) PID CONTROLLER DESIGN

In this section, it is shown how a robust performance PID controller can be synthesized for a set of possible linear models. The method is carried out by solving a min-max optimization problem formulated in frequency domain. As the synthesis of a low-order controller for a high-order plant is in general nonconvex (Safonov et al., 1994), the original optimization problem is divided into two convex optimization subproblems, which are solved iteratively until to achieve the complete convergence.

3.1 Control Configuration

The optimization subproblems are constructed using a general two degree-of-freedom (2DOF) feedback control configuration shown in Fig. 1, where y and y_{SET} are respectively, the control variable and the setpoint, G is the plant model, and the C_{SP}, C_{PI} and C_{PV} blocks constitute together the PID controller. The C_{PI} block is a PI controller whose structure is always fixed and given by (1), whilst C_{SP} and C_{PV} are dependent on the PID controller parameterization.

$$C_{PI} = K_C\left(1+\frac{1}{T_I s}\right) \tag{1}$$

The advantages to use the 2DOF control configuration are threefold: (a) It divides a typical nonconvex optimization problem (when the standard configuration is used) into two convex problems, which can be solved with the Sequential Iterative Optimization Method (SIOM) proposed by Faccin and Trierweiler (2004). (b) It consists in a common base, in which all possible industrial PID parameterization can be converted. In Faccin (2004) it is shown this conversion for several industrial PID parameterizations. (c) The controller order can be easily increased and implemented in modern DCS. For example, process filters for noise averting can be synthezed and incorporated into C_{PV}.

The controller used in this article is the ISA standard form (Åström and Hägglund, 1995) with $b = 1$, $c = 0$, $N = 10$ and has three adjustable parameters: K_C, T_I and T_D, given by (2). When it is put into the general configuration of Fig. 1, the controller blocks C_{SP} and C_{PV} are given by (3) and (4).

$$C = K_C\left[\left(1+\frac{1}{T_I\cdot s}\right)Y_{SET}-\left(1+\frac{1}{T_I\cdot s}+\frac{T_D\cdot s}{1+0.1T_D\cdot s}\right)Y\right] \tag{2}$$

$$C_{SP} = 1 \tag{3}$$

$$C_{PV} = \frac{1.1\cdot T_I\cdot T_D\cdot s^2+\left(T_I+0.1\cdot T_D\right)\cdot s+1}{0.1\cdot T_I\cdot T_D\cdot s^2+\left(T_I+0.1\cdot T_D\right)\cdot s+1} \tag{4}$$

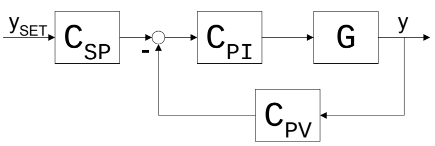

Fig. 1. The 2DOF feedback control configuration.

3.2 Optimization Problem

To design and convert the PID parameters, it is formulated a multiobjective optimization problem in frequency domain:

$$\begin{aligned}&\min_{\gamma, x\in R^+}\ \gamma\\ &subject\ \ to:\ F_n(x)-w_n\cdot\gamma\le 0;\quad n=1..N\end{aligned} \tag{5}$$

Where N is the total number of plant models, w_n is the weight given to the model n (as default, all weights are equal to 1), γ is an auxiliary optimization variable, x is the vector of the decision variables (the controller parameters which are optimized), and $F_n(x)$ is given by:

$$F_n(x)=\sum_{s=j\omega_0}^{j\omega_1}\left(\left[T_n(s,x)-T_0(s)\right]\frac{1}{s}\right)^2 \tag{6}$$

It consists of minimizing the difference between the closed loop transfer function of each model (T_n) and the desired attainable performance (T_O) on the frequency domain ($s = j\omega$) for a logspaced frequency range $\omega = [\omega_0\ \omega_1]$. The term ($1/s$) is used to emphasize that both response should be as close as possible for stepwise setpoint changing. Considering the control configuration of Fig. 1, the closed loop transfer function is given by:

$$T_n(s,x)=\frac{G_n(s)\cdot C_{PI}(s,x)\cdot C_{SP}(s,x)}{1+G_n(s)\cdot C_{PI}(s,x)\cdot C_{PV}(s,x)} \tag{7}$$

In (7), G_n is the model n and the C_{PI}, C_{SP} and C_{PV} are given respectively by (1), (3) and (4). When the parameters of these blocks are simultaneously optimized, the optimization problem is nonconvex. To overcome the nonconvexity, a sequential iterative procedure is applied, where the C_{PI} block is separately optimized to the other two blocks in two steps. At the first iteration, the following approximation is used to calculate the start point for the PI parameters (i.e., K_C and T_i):

$$F_n(x)=\sum_{s=j\omega_0}^{j\omega_1}\left\{\begin{bmatrix}G_n(s)\cdot C_{PI}(s,x)\cdot\left(1-T_0(s)\right)^2\\ -T_0(s)\cdot\left(1-T_0(s)\right)\end{bmatrix}\frac{1}{s}\right\}^2 \tag{8}$$

With these parameters, the C_{PI} block is fixed and a new optimization using (6) is done to calculate C_{PV} and C_{SP}. With the converged C_{PV} and C_{SP} a new C_{PI} is determined with a fixed C_{PI} obtained in the last iteration in the denominator of (7) and then with the new C_{PI}, the blocks C_{PV} and C_{SP} can be updated, and thus for ahead until all three blocks converge. The default stop criterion is 1% of relative error (from two sequential iterations) for all the parameters.

Particularly, the selected controller parameterization used in this paper, the ISA standard form given by (2), has a fixed C_{SP} block (no optimization parameter available). In spite of the MMA method is able to work with 2DOF controllers, to make the comparison with another tuning method, it was just designed this 1DOF controller.

In the optimization problem (5), another additional constraint can be considered (such as: maximal sensitivity or control action boundaries), but in the manner as the optimization problem was built, it is not normally necessary, since all these characteristics can be considered selecting an appropriated attainable performance.

3.3 Attainable Performance

The attainable performance is a very important factor in the MMA procedure. It must be a function with all desired characteristics (e.g., stable, unit gain and fast convergence to the setpoint, without a significant overshoot and good robustness characteristics). It could be a second order time delay model, with specified overshoot and rise time. However, to obtain better results, the order of this function should be compatible with the corresponding closed loop system order. In Dorf and Bishop (1998), transfer functions for several orders with the optimal coefficients considering an ITAE criterion for step response setpoint changes are presented. These functions are properly to be used as attainable performance, since they have unit gain, small overshoot, the smallest ITAE criteria and only one adjustable parameter (ω_n). The coefficients of these functions' denominator are shown in Table 1.

When a model with dead-time is used, the attainable performance must consider this constraint as a limitation. It could be done in two ways. In the first one, a higher order model without dead time approximates the dead time model. Large poles (that do not influence in the system dynamics, and which sum is exactly equivalent to the dead time) are added in the model. In this case, a higher order attainable performance is required to ensure a compatibility with the models. In the second way, the higher dead time needs to be included in the attainable performance. In this case, the solution becomes more difficult. Therefore, the first way was used in this article.

3.4 User-Friend Interface

An interface with the whole procedure was made in Matlab® platform, which turns easier the understanding and the obtaining of satisfactory results. This interface lets the practitioner to choose a suitable attainable performance by varying its order and the parameter ω_n. A graph with a step response of this function and all models together helps this chosen. Based on this function, an appropriated frequency vector is calculated automatically.

Table 1 Optimum coefficients of $T_O(s)$ based on the ITAE Criterion for step response

$s+\omega_n$
$s^2+1.4\omega_n\cdot s+\omega_n^2$
$s^3+1.75\omega_n\cdot s^2+2.15\omega_n^2\cdot s+\omega_n^3$
$s^4+2.1\omega_n\cdot s^3+3.4\omega_n^2\cdot s^2+2.7\omega_n^3\cdot s+\omega_n^4$
$s^5+2.8\omega_n\cdot s^4+5.0\omega_n^2\cdot s^3+5.5\omega_n^3\cdot s^2+3.4\omega_n^4\cdot s+\omega_n^5$

Several types of PID parameterizations are available and can be easily selected. All results are shown in a table where F_n values for each model and each iteration step are shown. Therefore, it is easy to visualize the convergence of the optimization method. The converged controller is then simulated for load disturbance and setpoint changes. In addition, robustness and performance criteria are calculated for each model.

4. CASE STUDY: SPHERICAL TANK LEVEL CONTROL

A nonlinear spherical tank system (Fig. 2) was studied aiming to show the benefits of the proposed procedure. The objective is to control the fluid level (*h*) through varying the inlet flow (*Fin*). This system is related to storage tanks, however it is not so common in the process industries. Cylindrical tanks are normally used but the spherical ones have a more nonlinear behavior, and because that they are more difficult to control.

4.1 Process Model and Transfer Function

Through a material balance for the system shown in Fig. 2, where the outlet flow (*Fout*) is dependent of the fluid level considering turbulent flow, equation (9) is obtained.

$$\frac{dh}{dt}=\frac{Fin-K\cdot\sqrt{h}}{\pi\cdot h\cdot\left(D-h\right)} \tag{9}$$

In (9), D is the tank diameter and K is the outlet flow capacity coefficient. For each inlet flow, a transfer function can be obtained, which is given by:

$$G=\frac{\Delta h\left(s\right)}{\Delta Fin\left(s\right)}=\frac{K_P}{\tau\cdot s+1}\cdot e^{-\theta\cdot s} \tag{10}$$

$$K_P=\frac{2\cdot Fin}{K^2} \tag{11}$$

$$\tau=\frac{2\cdot\pi\cdot Fin^3\cdot\left(D\cdot K^2-Fin^2\right)}{K^6} \tag{12}$$

A small pure time delay θ was included to consider the delays produced by valve and sensor dynamics. The stationary tank heigth is $h=(Fin/K)^2$. Here, an operating inlet flow range of 0 to 10 L/min, D = 25 cm, and $K = 2\ \mathrm{L.min^{-1}.cm^{0.5}}$ are used as parameters of the model. Fig. 3 shows the static gain, the time constant and the stationary tank height as a function of the inlet flowrate.

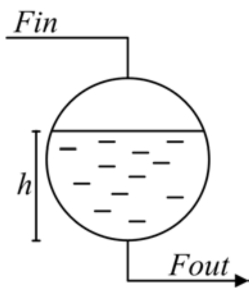

Fig. 2. Representation of the spherical tank system.

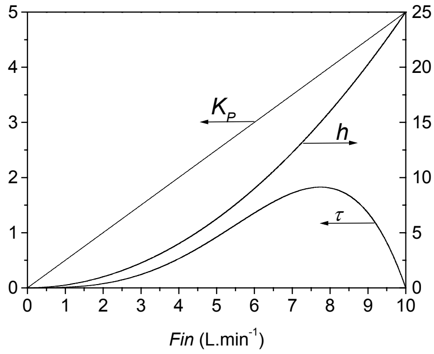

Fig. 3. Static gain (K_P, cm.min.L^{-1}), time constant model (τ, min) and stationary height (h, cm), for the whole inlet flowrate (*Fin*) range.

There is a significant nonlinearity in the model parameters, mainly in the time constant τ, which should be intuitively higher at the middle than at the top or bottom, however the maximum point is not at the middle, but a little higher up, at exactly 60% of height (or 15 cm). This asymmetry is a consequence of the nonlinear relation between *Fout* and *h*.

Assuming the nominal operating point as h = 12.5 cm and the normal operating range between 2 and 23 cm, three models representing these operating points where selected to describe the plant behavior. The transfer function G_1 is based on the minimum operating point (*Fin* = 2.83 L.min^{-1}), G_2 is the nominal one (*Fin* = 7.07 L.min^{-1}) and G_3 is based on the maximum operating point (*Fin* = 9.59 L.min^{-1}). A pure time delay of 0.1 min is used in all cases producing the following final expressions:

$$G_1 = \frac{1.41 \cdot e^{-0.1 \cdot s}}{0.204 \cdot s + 1}\ ;\quad G_2 = \frac{3.54 \cdot e^{-0.1 \cdot s}}{1.736 \cdot s + 1}\ ;\quad G_3 = \frac{4.80 \cdot e^{-0.1 \cdot s}}{0.693 \cdot s + 1}$$

4.2 Controller Design

A third order attainable performance with $\omega_n = 8$ was chosen because it is feasible to be achievable for all models. Fig. 4 shows the step responses for all models normalized by the respective gain and the desired attainable performance T_0. For the MMA procedure, the frequency range was divided in 200 points and the stop criterion was 0.1% of relative error. As the G_2 is the slowest model, the nominal one, and it represents the region where the system will operate more frequently, its weight (w_2) is 0.5, while the other both 1.

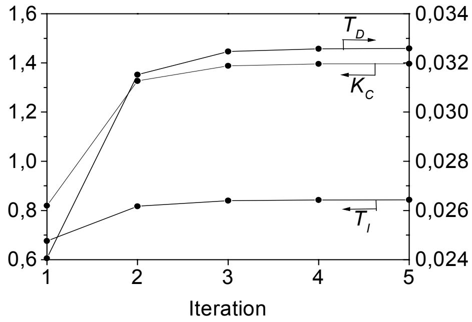

Fig. 5. Parameters convergence (K_C, T_I and T_D), for the iterations of the MMA optimization.

With these settings the optimization process converged in 5 iterations and the solution obtained was: K_C = 1.396; T_I = 0.843; T_D = 0.033. Fig. 5 shows the convergence of these parameters throughout the iterations. As it can be seen, the initial estimates are very similar to the final solution, and the procedure converges quickly to the optimal values. The corresponding step response to setpoint change for all linear models is shown in Fig. 6.

Normally, when a multi-model system needs to be controlled by only one set of PID parameters, the controller is designed for the limiting model. If the controller can control this model, it will also control the other ones, of course with a loss of performance. Nevertheless, determine the worst-case is not an easy task, mainly when there are differences in the gain and in the dynamic. When there are differences in the dead time or in the model order, it becomes even more difficult and can be strongly dependent on the desired closed loop response.

Therefore, to apply a simple tuning rule for this system, first it is necessary to find the worst-case. With an analysis on the bode diagram (not shown) one will conclude that it is the G_3 model. In this way, a Simple IMC (SIMC) tuning rule proposed by Skogestad (2003) was also employed to compare the performances. The SIMC tuning rules have for first order time-delay models only PI setting, which applied to the G_3 gives K_C = 0.722 and T_I = 0.693. Through a comparison between these parameters and the ones produced by the proposed multi-model approach (MMA), it can be seen that the MMA has a small derivative action, which is fundamental to the system's robustness. Fig. 7 shows the closed loop responses for the SIMC controller.

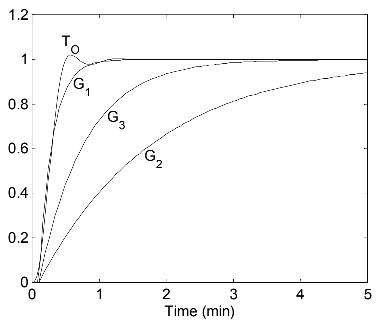

Fig. 4. Step responses for T_O and for G_1, G_2 and G_3 models normalized by the static gain.

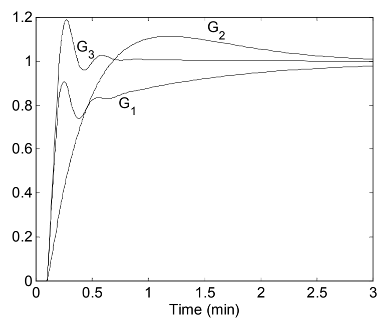

Fig. 6. Step response to setpoint change for G_1, G_2 and G_3 models with MMA controller.

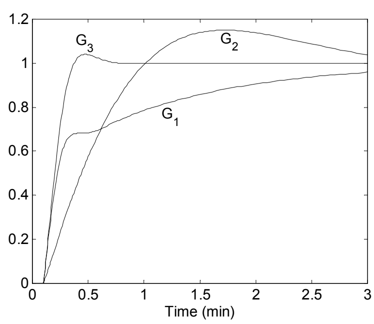

Fig. 7. Step response to setpoint change for G_1, G_2 and G_3 models with SIMC controller.

Table 2 Comparison criteria between MMA and SIMC tuning for all closed loop models

	G_1		G_2		G_3	
	MMA	SIMC	MMA	SIMC	MMA	SIMC
MS	1.27	1.07	1.04	1.02	1.21	1.06
GM	1.95	3.65	6.12	10.24	1.83	3.14
PM	79.2	107.2	67.2	58.7	53.3	61.3
IAE	0.404	0.631	0.455	0.678	0.185	0.217
ITAE	0.287	0.516	0.277	0.538	0.031	0.029
ST	3.06	3.88	2.62	3.25	0.64	0.61
OV	-	-	11.4	15.0	18.9	4.1
MD	0.293	0.376	0.238	0.355	0.379	0.549

Now, the performances of these two controllers are compared at Table 2 using maximal sensitivity (MS), gain margin (GM) and phase margin (PM) as robustness criteria and integral of the absolute value of the error (IAE), integral of product of time and the absolute value of the error (ITAE), 2% settling time (ST, in min), maximal overshoot (OV, in % of the setpoint), and the maximal deviation to the setpoint, when a load disturbance of 0.5 $L.min^{-1}$ on the inlet flowrate is done (MD, in cm) as performance criteria.

Several remarks can be taken from Table 2. The SIMC tuning produced more robust response for all models, of course, because it was designed for the worst-case, but it has a significant performance loss for models G_1 and G_2 (higher ST values). The MMA tuning has shown responses with better performance (smaller values for IAE and ITAE), a little aggressive for G_3 model (high OV), but with suitable robustness criteria. For load disturbance it seems clear that MMA tuning is much better (the MD values for SIMC tuning is 40% higher in average), since with a more aggressive tuning, it can reject more easily the disturbances.

5. CONCLUSIONS

A new powerful multi-model approach (MMA) for PID controller tuning was presented in details. The MMA produces a robust controller with a trade-off among the performance of all considered models and it is flexible to be used for any parameterization type. The desired attainable performance is the main tuning parameter, which can be selected to produce better performance or robustness. The paper showed how the desired attainable performance could be easily set. The proposed tool considerably extends the set of process where a PID controller will produce good results, since a low order controller with robust performance can be easily and fastly synthesize.

The proposed method can work with models of different orders and structures, i.e., it is not based on a fixed model type (such as first order plus time delay, used by several methods). This characteristic together with the multi-model approach are responsible for the high flexibility and suitability of the method for industrial applications, where classical tuning methods do not always give satisfactory results.

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REFERENCES

Åström, K.J. and T. Hägglund (1995). *PID Controllers: Theory, Design, and Tuning.* Instrument Society of America, Research Triangle Park.

Carotenuto, L., P. Pugliese and Y.D. Sergeyev (2002). Maximizing Performance and Robustness of PI and PID Controllers by Global Optimization. *15th IFAC World Congress*, Barcelona.

Dorf, R.C. and Bishop R.H. (1998). *Modern Control Systems*. Addison-Wesley, Menlo Park.

Faccin, F. (2004). *Development and Application of a Novel Tool for PID Controller Design* (In Portuguese). Master Thesis. Federal University of Rio Grande do Sul, Chemical Eng. Department, Brazil.

Faccin, F. and J.O. Trierweiler (2004). Performance Limitations of Some Industrial PID Controllers. Submitted to *14th ESCAPE Symposium*, Lisboa.

Goodwin, G.C., S.F. Graebe and M.E. Salgado (2001). *Control System Design.* Prentice-Hall, New Jersey.

Harris, T.J., C.T. Seppale and L.D. Desborough (1999). A review of performance monitoring and assessment techniques for univariate and multivariate control systems. *Journal of Process control*, **9**, 1-17.

Ingimundarson, A. and T. Hägglund (2002). Performance comparison between PID and dead-time compensating controllers. *Journal of Process Control*, **12**, 887-895.

Isaksson, A.J. and S.F. Graebe (2000). The derivative filter is an integral part of PID design. Presented at *Control Systems 2000*. Victoria, British Columbia.

Kempf, A.O. (2003). *Control Loops Performance Assessment* (In Portuguese). Master Thesis. Federal University of Rio Grande do Sul, Chemical Engineering Department, Brazil.

Luyben, W.L. (2001). Effect of Derivative Algorithm and Tuning Selection on the PID Control of Dead-Time Process. *Ind. Eng. Chem. Res.*, **40**, 3605-3611.

O'Dwyer, A. (2003). *Handbook of PI and PID Controller Tuning Rules.* World Scientific Pub Co, London.

Piazzi, A. and A. Visioli (2002). Improved PI Control via Dynamic Inversion. *15th IFAC World Congress*, Barcelona.

Safonov, M.G., K.C. Goh and J.H. Ly (1994). Control System Synthesis via Bilinear Matrix Inequalities. *Proc. of American Control Conference*, pp. 45-49, Baltimore.

Skogestad, S. (2003). Simple analytic rules for model reduction and PID controller tuning. *Journal of Process Control*, **13**, 291-309.