Preprints of the 19th World Congress
The International Federation of Automatic Control
Cape Town, South Africa. August 24-29, 2014

A Novel PID Controller Design Methodology for Specified Performance Using Ultimate Plant Parameters

Š. Bucz, V. Veselý, A. Kozáková, Š. Kozák

Faculty of Electrical Engineering and Information Technology,
Slovak University of Technology in Bratislava, Ilkovičova 3, SK-812 19 Bratislava, Slovak Republic
(e-mail: stefan.bucz@stuba.sk, vojtech.vesely@stuba.sk, alena.kozakova@stuba.sk, stefan.kozak@stuba.sk)

Abstract: The paper deals with a new PID controller design method based on integrating requirements on transient performance into the popular frequency-domain Ziegler-Nichols design approach. The developed method provides support to the designer by converting identified ultimate plant parameters into PID controller parameters using variable weights that depend on expected maximum overshoot η_{max} and settling time t_s of the closed-loop step response. The weights $(\alpha_1,\alpha_2,\alpha_3)$ of ultimate plant parameters T_c and K_c occurring in tuning rules $\Theta_{PID}=(P,T_i,T_d)=(\alpha_1K_c,\alpha_2T_c,\alpha_3T_c)$ differ from standard recommendations of the Ziegler-Nichols method $(\alpha_{1Z\text{-}N},\alpha_{2Z\text{-}N},\alpha_{3Z\text{-}N})=(0.6,0.5,0.125)$. Developed PID controller tuning rules are presented in the modified Ziegler-Nichols table.

Keywords: performance evaluation, overshoot, settling time, crossover frequency, phase margin

1. INTRODUCTION

No need for mathematical model of the plant, quick computation of controller parameters and simple algorithmisation are main attributes due to which the frequency response Ziegler-Nichols method (Ziegler and Nichols, 1942) is widely used for tuning PID controllers implemented in industrial control loops (Kristiansson and Lennartson, 2002). However, it is a closed design method not allowing the designer to modify the performance with respect to the specific technological process (O'Dwyer, 2000), (Osuský et al., 2010).

Since it was first published in 1942, many studies on extension of the Ziegler-Nichols method have appeared (McAvoy and Johnson, 1967), (Atkinson and Davey, 1968), (Tinham, 1989), (Blickley, 1990). (Pettit and Carr, 1987) propose three settings $(\alpha_1,\alpha_2,\alpha_3)=(1,0.5,0.125)$, $(0.5,1,0.167)$, $(0.67,1,0.167)$, the first two leading to underdamped and aperiodic responses of the output variable, respectively, and the third one to a response on the aperiodicity border. Rather than fixed values (Karaboga and Kalinli, 1996) propose intervals $\alpha_1\in\langle 0.32,0.6\rangle$, $\alpha_2\in\langle 0.213,1.406\rangle$, $\alpha_3\in\langle 0.133,0.469\rangle$, however, without any recommendations with respect to expected performance. According to its authors, the methods according to (Chau, 2002) guarantee „just a small overshoot“ if using the weights (0.33,0.5,0.333), and an „overshoot-free response“ for (0.2,0.5,0.333). However, assessment of expected performance achieved by PID controllers tuned according to these methods is very approximate and only representative.

To remove this drawback, the proposed modified frequency response Ziegler-Nichols method allows to achieve specified maximum overshoot $\eta_{max}\in\langle 0\%,50\%\rangle$ and settling time $t_s\in\langle 7/\omega_c,22/\omega_c\rangle$ of the closed-loop response to the setpoint step change, where ω_c is the plant critical frequency.

The paper is organized as follows: Section 2 describes the classical frequency response Ziegler-Nichols method and demonstrates its modification with respect to transient performance requirements. Achieved performance is assessed and modified Ziegler-Nichols tuning rules for various values of maximum overshoot and settling time are provided. The proposed method was verified via simulation on benchmark examples and on a real plant - a DC motor; the results are in Sections 3. Evaluation of achieved results is summarized in Section 4.

2. PROBLEM FORMULATION AND SOLUTION

2.1 *Ziegler-Nichols frequency response method: principle and analysis*

The frequency domain Ziegler-Nichols method (Ziegler and Nichols, 1942) is a direct PID controller tuning method with fast rejection of the disturbance d(t) being most frequently cited in technical literature. To design a controller, only two characteristic parameters of the unknown plant are to be identified.

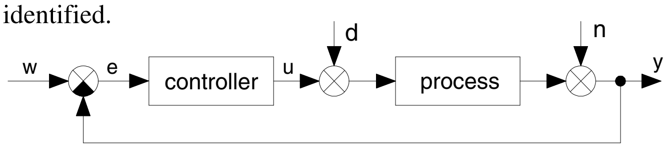

Fig. 1. Feedback control loop

Consider the feedback loop in Fig. 1; put the PID controller in proportional mode and increase the gain K of the controller $G_R(s)=K$ until the output y(t) exhibits persistent oscillations; from them, the critical period $\boldsymbol{T_c}$ and the related critical gain $\boldsymbol{K_c}$ are read. If considering the standard interacting form of the PID controller

$$G_R(s)=K\left(1+\frac{1}{T_i s}+T_d s\right) \quad (1)$$

coefficients of P, PI and PID controllers are calculated according to the Table 1.

Table 1. PID tuning rules according to the Ziegler-Nichols frequency response method

Controller	***K***	T_i	T_d	T_p
P	$0,5K_c$	-	-	T_c
PI	$0,45K_c$	$0,8T_c$	-	$1,4T_c$
PID	$0,6K_c$	$0,5T_c$	$0,125T_c$	$0,85T_c$

Relations in the last column of Table 1 can be used to estimate the dominant closed-loop dynamics T_p (Åström and Hägglund, 1995). According to the Ziegler-Nichols frequency response method, if the open-loop transfer function with the proportional controller (Fig. 1)

$$L(j\omega)=G(j\omega)G_R(j\omega)=KG(j\omega) \quad (2)$$

is at the limit of instability, it can be expressed in polar form according to the Nyquist condition

$$L(j\omega_c)=-1=1e^{-j180^\circ} \quad (3)$$

where $\omega_c=2\pi/T_c$ is the critical frequency of the plant. From comparison of (2) and (3) at $\omega=\omega_c$ and for $K=K_c$ results the complex equation

$$G(j\omega_c)=[1/K_c]e^{-j180^\circ} \quad (4)$$

which expresses position of the plant critical point $C=G(j\omega_c)$ with coordinates $\{\omega_c, 1/K_c, -\pi\}$ on the negative half-axis of the complex plane. This point is crossed by the frequency characteristics of the unknown plant. If we substitute the Ziegler-Nichols PID controller tuning rules from Table 1 into the frequency response transfer function of the PID controller

$$G_R(j\omega)=K\left[1+j\left(T_d\omega-\frac{1}{T_i\omega}\right)\right] \quad (5)$$

and consider critical frequency, we obtain the complex number

$$G_R(j\omega_c)=0,6K_c\left[1+j\left(\frac{T_c}{8}\frac{2\pi}{T_c}-\frac{T_c}{0,5T_c.2\pi}\right)\right]=0,66K_c e^{j25,01^\circ}$$

with a magnitude depending solely on the critical gain of the plant, and a constant argument. Hence, the PID controller designed by the Ziegler-Nichols method moves the critical point C of the plant with coordinates (4) into the fixed position in the complex plane $L(j\omega_c)=G(j\omega_c)G_R(j\omega_c)$

$$L(j\omega_c)=\left[\frac{1}{K_c}e^{-j180^\circ}\right]\left[0,66K_c e^{j25^\circ}\right]=0,46e^{-j155^\circ}=-0,6-j0,28$$

which will be one point of the open-loop Nyquist plot $L(j\omega)$ under the designed PID controller. The PID controller tuned according to the Ziegler-Nichols rules in Table 1 moves the frequency characteristic of the unknown plant $G(j\omega)$ in the critical frequency ω_c into the target point $L_{ZN}=L(j\omega_c)=[-0,6;-j0,28]$, i.e. into a due distance from the critical point ***(-1+j0)*** (see Fig. 2a).

One can question about how to generalize the Ziegler-Nichols method to be able to shift the identified point of the unknown plant not into the fixed L_{ZN} but rather to a free point L with general coordinates (x+jy) specified by the designer in terms of the performance measures η_{max} and t_s (Veselý, 2003)?

2.2 Principle of the modified frequency response Ziegler-Nichols method for specified performance

The presented modified version of the Ziegler-Nichols method integrates performance requirements into its classical version (Bucz and Kozáková, 2012). The PID controller is tuned using the derived modification of the Ziegler-Nichols table which includes separate rules for adjusting controller coefficients for:

- maximum overshoot $\eta_{max}\in\{0\%,10\%,20\%,30\%,40\%,50\%\}$,
- settling time $t_s\in\{7/\omega_c,10/\omega_c,13/\omega_c,16/\omega_c,19/\omega_c,22/\omega_c\}$.

Principle of the proposed modification consists in moving the identified critical point of the plant $C=G(j\omega_c)=[-1/K_c,j0]$ using PID controller into the complex plane point $L(j\omega_c)=x+jy$ which will be a point of the Nyquist plot $L(j\omega)$ of the designed open-loop (see Fig. 2b). This compensation is carried out at critical frequency ω_c of the plant. Coordinates x and y specifying the future position of the critical point C at ω_c will depend on the expected performance specified by the designer in terms of η_{max} and t_s.

Mathematically, this compensation can be described by the open-loop transfer function at ω_c:

$$L(j\omega_c)=G(j\omega_c)G_R(j\omega_c)=x+jy \quad (6)$$

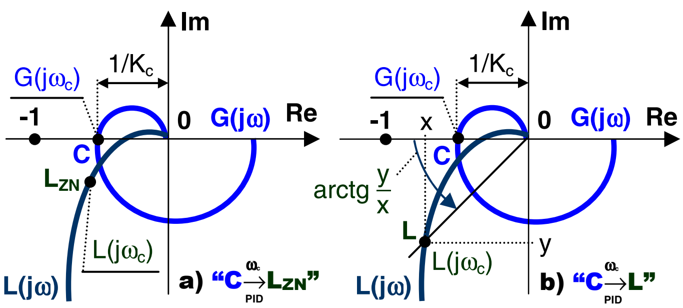

Fig. 2. Illustration of moving the critical point C into a) $L_{ZN}=[-0,6-j0,28]$ (by Z-N method); b) L=[x+jy] (by the proposed method)

After substituting coordinates of the critical point C into (6), the controller transfer function $G_R(j\omega_c)$ turns into a complex number

$$G_R(j\omega_c)=\frac{L(j\omega_c)}{G(j\omega_c)}=-K_c(x+jy) \quad (7)$$

If equating (7) and the PID controller frequency transfer function (5), controller coefficients can be obtained from the complex equation at $\omega=\omega_c$

$$-K_c(x+jy)=K\left(1+j\left(T_d\omega_c-\frac{1}{T_i\omega_c}\right)\right) \qquad (8)$$

To calculate PID controller coefficients, following relations resulting from (8) are used

$$K=-K_c x;\ KT_d\omega_c-\frac{K}{T_i\omega_c}=-K_c y \qquad (9)$$

Two of the unknown parameters occur in (9b): T_i and T_d. To obtain unambiguous solution, introduce a new variable – ratio of the integral and derivative constants $\beta=T_i/T_d$ and substitute for $T_i=\beta T_d$ in (9b). After simple manipulations, the derivative constant can be obtained by solving a quadratic equation in T_d

$$K\beta T_d^2\omega_c^2+yK_c\beta T_d\omega_c-K=0 \qquad (10)$$

Considering (9a), solution of (10) provides following relations to calculate PID controller coefficients

$$T_i=\beta T_d;\ T_d=\frac{y}{2x\omega_c}+\frac{1}{x\omega_c}\sqrt{\frac{y^2}{4}+\frac{x^2}{\beta}} \qquad (11)$$

After substituting $\omega_c=2\pi/T_c$ into (9a) and (11b) and choosing $\beta=4$ we obtain

$$K=-xK_c;\ T_i=T_c\left[\frac{y}{\pi x}+\frac{1}{\pi}\sqrt{\left(\frac{y}{x}\right)^2+1}\right];\ T_d=\frac{T_i}{4} \qquad (12)$$

After small modifications, final relations for calculating PID controller coefficients are obtained (Table 2).

Table 2. Relations for calculating PID controller coefficients by classical and modified Ziegler-Nichols methods

Method	***K***	T_i	T_d
Classical Z-N	$0{,}6K_c$	$0{,}5T_c$	$0{,}125T_c$
Modified Z-N	$-xK_c$	$\left[\frac{1}{\pi}\frac{y}{x}+\frac{1}{\pi}\sqrt{\left(\frac{y}{x}\right)^2+1}\right]T_c$	$\left[\frac{1}{4\pi}\frac{y}{x}+\frac{1}{4\pi}\sqrt{\left(\frac{y}{x}\right)^2+1}\right]T_c$

Note that the critical point of the plant can be identified using the well-known Rotach (relay) experiment (Rotach, 1984).

While according to Ziegler-Nichols, the PID controller parameters are computed using the formula $\Theta_{PID}=(P,T_i,T_d)=(\alpha_1K_c,\alpha_2T_c,\alpha_3T_c)$ with fixed weights $(\alpha_1,\alpha_2,\alpha_3)=(0{,}6;0{,}5;0{,}125)$ on critical parameters K_c and T_c, the proposed new method provides assistance in converting the identified critical parameters of the plant into PID controller coefficients using variable weights $(\alpha_1,\alpha_2,\alpha_3)$ given as

$$\alpha_1=-x;\ \alpha_2=\frac{1}{\pi}\left[\frac{y}{x}+\sqrt{\left(\frac{y}{x}\right)^2+1}\right];\ \alpha_3=0{,}25\alpha_2 \qquad (13)$$

which depend on position x (proportional gain) and on the ratio y/x (constants T_i and T_d). Tuning formulas of the modified Ziegler-Nichols method are in Table 2. Another question arises: What is the relation between the coordinates of the point $L(j\omega_c)=x+jy$ and the expected maximum overshoot η_{max} and settling time t_s?

2.3 *Performance evaluation and creation of the modified Ziegler-Nichols tuning rules*

It is well-known from the control theory that a satisfactory performance is achieved if in the middle frequency range the slope of the open-loop magnitude Bode plot is -20 dB/decade and that of the phase Bode plot is -66°/decade. If we depict the position of the point $L(j\omega_c)=x+jy$ in open-loop Bode plot coordinates, we obtain two points: $L_A=[20\log|L(j\omega_c)|,\log\omega_c]$, and $L_F=[\arg L(\omega_c),\log\omega_c]$. According to Fig. 3 their coordinates are

$$L_A: 20\log|L(j\omega_c)|=20\log\sqrt{x^2+y^2} \qquad (14)$$

$$L_F: \arg L(j\omega_c)=-\pi+arctg(y/x) \qquad (15)$$

It is supposed that both the magnitude and phase Bode plots of $L(j\omega)$ are known only in the point L, yet they conform to the above performance requirement. Then a straight line can be drawn with the slope s_A=-20 dB/decade passing through L_A with coordinates (14) expressed by the equation

$$20\log|L(j\omega)|-20\log\sqrt{x^2+y^2}=-20(\log\omega-\log\omega_c) \qquad (16)$$

From (16) the open-loop magnitude crossover frequency ω_a^* can be estimated (Fig. 3). For $\omega=\omega_a^*$ the magnitude is $20\log|L(j\omega_a^*)|=0$; after substituting into (16) we obtain

$$-20\log\sqrt{x^2+y^2}=-20(\log\omega_a^*-\log\omega_c) \qquad (17)$$

Using (17), the magnitude crossover ω_a^* can be expressed as

$$\omega_a^*=\omega_c\sqrt{x^2+y^2} \qquad (18)$$

The straight line with a slope s_F=-66°/decade passing through L_F with coordinates (15) expressed by the equation

$$\arg L(\omega)+\pi-arctg(y/x)=-66(\log\omega-\log\omega_c) \qquad (19)$$

enables to estimate the open-loop phase margin ϕ_M (Fig. 3)

$$\phi_M=\pi-\arg L(\omega_a^*) \qquad (20)$$

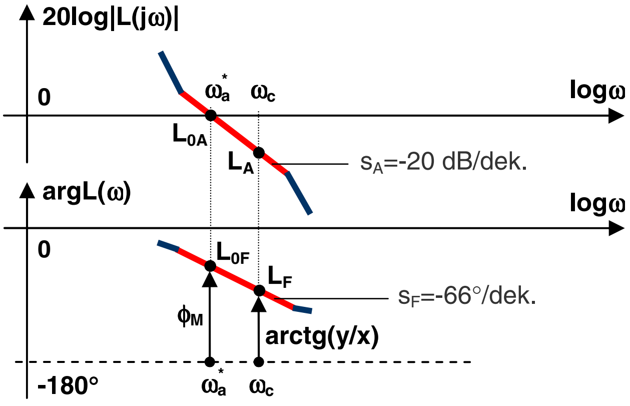

Fig. 3. Identification of ϕ_M and ω_a^* from the approximation of the middle frequency range of $L(j\omega)$

After substituting for $\omega=\omega_a^*$ and (19) into (20) we obtain

$$\phi_M = -66\left(\log \omega_a^* - \log \omega_c\right) + arctg(y/x) \tag{21}$$

The settling time can be estimated according to the relation

$$t_s = \frac{\gamma}{\omega_a^*} \tag{22}$$

where the curve factor $\gamma \approx 3$ for aperiodic closed-loop responses (Grabbre et al., 1959-61) and $\gamma \in \langle \pi, 4\pi \rangle$ for oscillatory output variable response (Reinisch, 1974), (Hudzovič, 1982). Analysis of PID controller designs for benchmark examples (Åström and Hägglund, 2000) has revealed that the closed-loop performance is satisfactory if

$$\omega_a^* = \sigma \omega_c \tag{23}$$

where $\sigma \in \langle 0.5, 0.95 \rangle$ (Bucz et al., 2011). Left-hand-side of the equation

$$t_s \omega_c = \gamma / \sigma \tag{24}$$

obtained by substituting (23) into (22) defines a new performance measure, the so-called relative settling time

$$\tau_s = t_s \omega_c \tag{25}$$

which expresses the real settling time weighted by the critical frequency ω_c of the controlled plant. It is a dimensionless quantity which enables to express expected closed-loop dynamics for plants with various dynamics.

Squaring (18) can be manipulated to obtain the equation of a circle

$$x^2 + y^2 = \left(\frac{\omega_a^*}{\omega_c}\right)^2 \Rightarrow x^2 + y^2 = \left(\frac{\gamma}{\tau_s}\right)^2 \tag{27}$$

with the radius $\sigma=\omega_a^*/\omega_c=\gamma/\tau_s$ specifying the relative settling time (25) for the constant curve factor γ. Parameters $\sigma=\omega_a^*/\omega_c$=0.5, 0.65, 0.8 and 0.95 substituted into (27) define a set of concentric circles $\boldsymbol{\Omega}$ of settling times centred in the origin of the complex plane.

If the complementary sensitivity plot $|T(j\omega)|$ is a non-monotonic function of angular frequency ω, i.e. it has a magnitude peak (Ingimundarson et al., 2004)

$$M_t = \sup_{\omega} |T(j\omega)| = \sup \left| \frac{L(j\omega)}{1 + L(j\omega)} \right| \tag{28}$$

the maximum overshoot η_{max} of the closed-loop step response can be estimated according to

$$\eta_{\max} \le 100 \frac{1{,}18 M_t - |T(0)|}{|T(0)|} \; [\%] \tag{29}$$

If the controller includes the integrator channel of the control error then $|T(0)|=1$. There exist some $L(j\omega_c)=x+jy$ such that the following equality holds

$$M_t^2 = \frac{x^2 + y^2}{[1 + x]^2 + y^2} \tag{30}$$

After some manipulations of (30), the equation describing the Hall circles $\boldsymbol{M_T}$ is obtained

$$\left[x + \frac{M_t^2}{M_t^2 - 1}\right]^2 + y^2 = \left[\frac{M_t}{\left|1 - M_t^2\right|}\right]^2 \tag{31}$$

centred in $C_T=-M_t^2/[M_t^2-1]$ with radii $R_T=M_t/|1-M_t^2|$ obtained for various M_t, and hence for various maximum overshoots η_{max} according to (29).

For specified maximum overshoot and settling time yielding the set of M_T, and a set of $\sigma \in \langle 0.5, 0.95 \rangle$, the set of Hall and Ω circles are drawn in the complex plane. Then we are looking the points where the circles ***(M_T,Ω)*** touch (contact points CP); each of them defines the maximum value of overshoot η_{max} and an approximate value of the settling time t_s. Selection of one of these CPs is upon the designer. Finally, the coordinates (x+jy) are obtained of the searched target point L of the open loop frequency response $L(j\omega_c)$.

The sets of M_T circles for maximum overshoots η_{max}=0 % to 50 %, and Ω circles for settling times for $\gamma/\tau_s=\omega_a^*/\omega_c$=0.5, 0.65, 0.8 and 0.95 are depicted in Fig. 4.

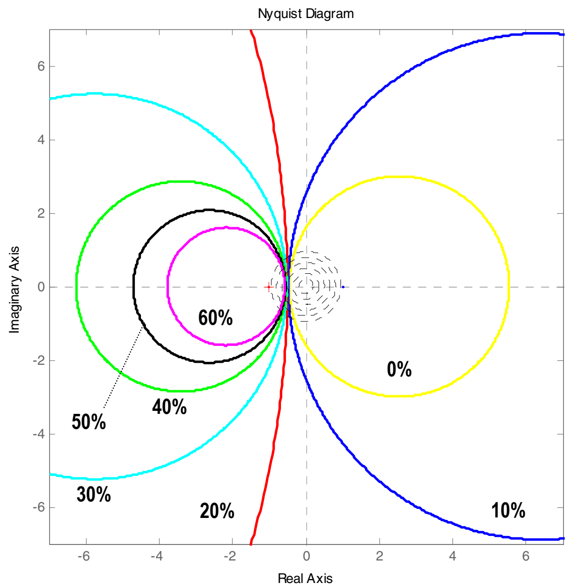

Fig. 4. Contact points of the M_T and Ω circles

If in (29) the equality holds, the Nyquist plot of $L(j\omega)$ exactly touches the Hall circle M_T, in case of inequality it avoids the area delineated by the Hall circle M_T.

Intersections and contact points of the M_T and Ω circles can be calculated by solving the set of equations (27) and (31) with respect to x,y:

$$x = \frac{1}{2}\left[\frac{118^2 - (\eta_{\max} + 100)^2}{2(\eta_{\max} + 100)^2}\left(\frac{\omega_a^*}{\omega_c}\right)^2 + 1\right] \tag{32}$$

$$y = -\sqrt{\left|\left(\frac{\omega_a^*}{\omega_c}\right)^2 - x^2\right|} \tag{33}$$

The coordinates (32) and (33) specify the point L in the complex plane to which is moved the identified critical point C of the plant at the critical frequency ω_c. After substituting (32) and (33) into the weights (13) we obtain the modified Ziegler-Nichols tuning table (Table 3).

Table 3. Modified Ziegler-Nichols table for required maximum overshoot and (relative) settling time

η_{max}	0 %	10 %	20 %	30 %	40 %	50 %
K	$0.1304K_c$	$0.2714K_c$	$0.5433K_c$	$0.6159K_c$	$0.6692K_c$	$0.7093K_c$
T_i	$1.0535T_c$	$0.8705T_c$	$0.7948T_c$	$0.6827T_c$	$0.6183T_c$	$0.5777T_c$
T_d	$0.2634T_c$	$0.2176T_c$	$0.1987T_c$	$0.1707T_c$	$0.1546T_c$	$0.1444T_c$
τ_s	7	10	13	16	19	22
K	$0.2309K_c$	$0.2773K_c$	$0.2816K_c$	$0.2844K_c$	$0.2654K_c$	$0.2549K_c$
T_i	$1.0083T_c$	$1.4364T_c$	$1.7102T_c$	$1.9885T_c$	$2.1480T_c$	$2.2468T_c$
T_d	$0.2521T_c$	$0.3591T_c$	$0.4275T_c$	$0.4971T_c$	$0.5370T_c$	$0.5617T_c$

3. VERIFICATION OF THE PROPOSED METHOD

Let us design PID controllers (1) for the benchmark example

$$G_A(s) = \frac{1}{(0{,}01s+1)^3}$$

using the modified frequency response Ziegler-Nichols method; the control objective is to achieve required performance specified in terms of maximum overshoots η_{max}=0%, 10%, 20% and 30% and required relative settling times τ_s=7, 10, 13 and 16 of the closed-loop step response. All data needed for the design of PID controllers for the plant $G_A(s)$ along with achieved performance measure values (marked with "*" in the lost column) are summarized in Table 4.

Table 4. Summary of given and achieved performance measure values and corresponding PID controller coefficients

System	η_{max}/τ_s	ω_c [rad/s]	K	T_i	T_d	η_{max}^*/τ_s^*
$G_A(s)$	0 %	173.22	1.0433	0.0382	0.0096	0 %
$G_A(s)$	10 %	173.22	2.1715	0.0316	0.0079	9.7 %
$G_A(s)$	20 %	173.22	4.3470	0.0288	0.0072	18.6 %
$G_A(s)$	30 %	173.22	4.9279	0.0248	0.0062	27.4 %
$G_A(s)$	7	173.22	1.8475	0.0366	0.0091	6.85
$G_A(s)$	10	173.22	2.2187	0.0521	0.0130	9.36
$G_A(s)$	13	173.22	2.2531	0.0620	0.0155	12.84
$G_A(s)$	16	173.22	2.2755	0.0721	0.0180	15.65

Fig.6 and Fig.7 show closed-loop step response shaping for different values of η_{max} and τ_s. The modified frequency response Ziegler-Nichols method was applied to control a physical model of a DC permanent magnet motor; controlled variable was the speed and plant input was armature voltage generated using the Matlab-Realtime Workshop control system. A speed-voltage generator was used to sense the output variable y(t). Block diagram of the control loop configuration with the DC motor is in Fig. 8.

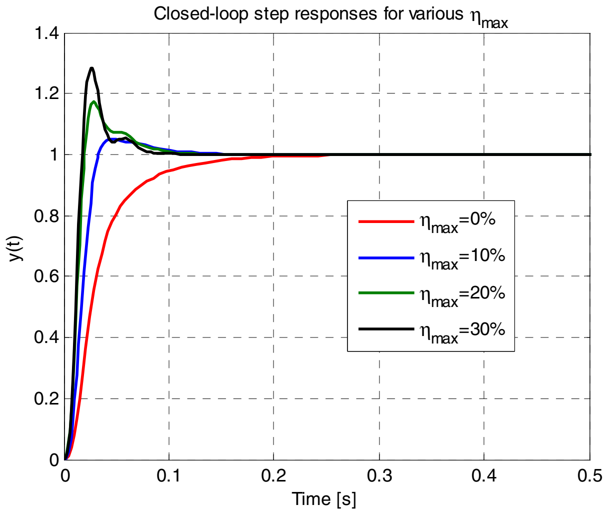

Fig. 6. Closed-loop step responses for $G_A(s)$ and various values of η_{max}

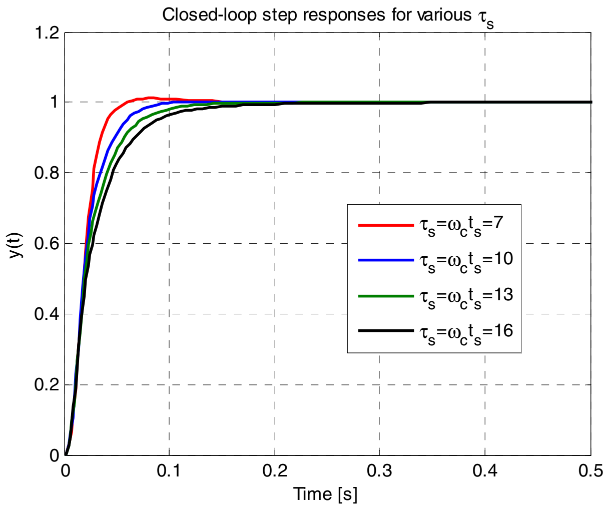

Fig. 7. Closed-loop step responses for $G_A(s)$ and various values of τ_s

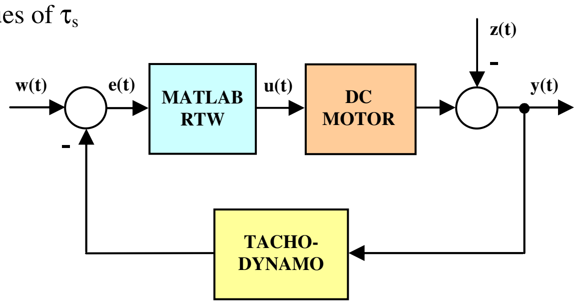

Fig. 8. Control loop with the DC motor

The control objective was to guarantee maximum overshoot η_{max}=0% and 20%. Resulting closed-loop step responses are depicted in Fig. 9. Figures 6, 7 and 9 prove that PID controllers designed by the modified frequency response Ziegler-Nichols method were able to guarantee the required performance measure values.

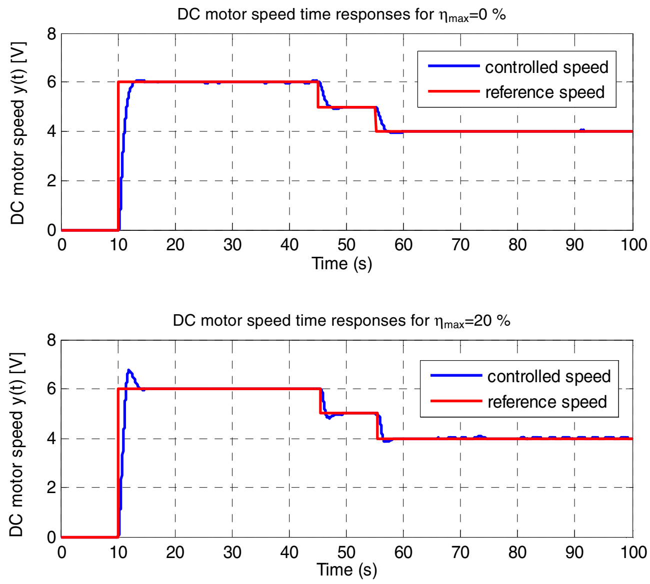

Fig. 9. Time responses of the DC motor speed for various values of maximum overshoot η_{max} (red – setpoint w(t), blue – output y(t))

4. CONCLUSIONS

The proposed PID controller design method provides a design tool that enables the designer to systematically shape the closed-loop response. We recommend it to control systems with monotonic step response. The method was successfully verified on a vast set of benchmark examples (Åström and Hägglund, 2000).

The developed approach preserves simplicity of the original Ziegler-Nichols method (only measured critical parameters of the plant are needed to design a PID controller design), and is easy-to-implement in autotuners of industrial controllers. Using the derived Ziegler-Nichols table modification will contribute to improving cost-effectiveness of industrial processes operation, and also the unfavourable portion of properly designed controllers out of all installed PID controller types.

ACKNOWLEDGMENT

This research work has been supported by the Scientific Grant Agency of the Ministry of Education of the Slovak Republic under grants No. 1/1241/12, 1/2256/12, 1/0937/14 and the Slovak Research and Development Agency under grant APVV-0772-12.

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