Research regarding the design of a new transport pipe flow control system

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Abstract: - The object of this paper is the design of a new transport pipe flow control system through the adjustment of the speed of the centrifugal pump. The novelty of this type of flow control system is that the induction motor speed control system is a sensorless one, using as a flux and speed estimator a new type of observer, suggestively called “Extended Gopinath Observer”.

Key-Words: - Extended Gopinath Observer, Sensorless Control, Flow Control, Centrifugal Pumps.

1 Introduction
This paper presents a new flux and rotor speed observer [5] called an Extended Gopinath Observer (EGO). The design of the EGO observer is done based on an adaptive mechanism using the notion of Popov hyperstability [3].

Thus, this type of observer is included in the estimation methods based on an adaptation mechanism, along with the Extended Luenberger Observer (ELO) proposed by Kubota [2] and the Model Adaptive System (MRAS) observer proposed by Schauder [1].

This type of speed control system is used in the second part of the paper in the design of a pipe flow control system.

2 The Extended Gopinath Observer
The equations that define the rotor flux Gopinath observer are [5]:

\[
\begin{align*}
\frac{d}{dt} \hat{l}_s &= a_{11}^* \hat{l}_s + a_{12}^* \hat{l}_r + a_{13}^* \hat{\psi}_r + b_{11}^* u_s \\
\frac{d}{dt} \hat{\psi}_s &= a_{21}^* \hat{l}_s + a_{22}^* \hat{\psi}_r + g \left[ \frac{d}{dt} l_s - \frac{d}{dt} \hat{l}_s \right]
\end{align*}
\]

(1)

where:

\[
\begin{align*}
a_{11}^* &= a_4^* + j \cdot z_p \cdot \omega_k \\
a_{12}^* &= a_3^* - j \cdot (a_{14}^* \cdot z_p \cdot \omega_k) \\
\hat{a}_{13}^* &= \frac{L_m^*}{L_s^* \cdot T_r^* \cdot \sigma} \\
a_{14}^* &= \frac{L_m^*}{L_s^* \cdot T_r^* \cdot \sigma} \\
\hat{a}_{21}^* &= a_{31}^*
\end{align*}
\]

\[
a_{22}^* = a_{33}^* + j \cdot z_p \cdot \omega_k \\
\hat{a}_{11}^* = \frac{1}{T_r^*} \\
\hat{b}_{11}^* = \frac{1}{L_s^* \cdot \sigma} \\
T_r^* = \frac{L_s^*}{R_s}
\]

\[
\begin{align*}
a_{33}^* &= -\frac{1}{T_s^* \cdot \sigma} \\
a_{31}^* &= -\frac{1 - \sigma^*}{T_s^* \cdot \sigma} \\
a_{32}^* &= -\frac{1}{L_s^* \cdot \sigma} \\
a_{34}^* &= -\frac{1 - \sigma^*}{L_s^* \cdot \sigma}.
\end{align*}
\]

In the relations above, are marked with “*” the identified electrical sizes of the induction motor.

The block diagram of the Extended Gopinath Observer (EGO) is presented in figure 1.

![Block Diagram of the Extended Gopinath Observer](image)

Fig. 1. The Principle Schematic of the EGO

The essential element in the stability of the Gopinath flux observer is the g gain, which is a complex number in the following form:

\[
g = g_0 + j \cdot g_0
\]

(2)
In order to design this type of estimator we need to position the estimator’s poles in the left Nyquist plane so that the estimator’s stability is assured.

The expressions $g_a$ and $g_b$ after the pole positioning are [5]:

$$
g_a = -\frac{a_{31}^* a_{33}}{(a_{31}^*)^2 + (z_p \cdot \omega_r)^2}
$$

$$
g_b = -\frac{a_{31}^* Z_p \cdot \omega_r}{(a_{31}^*)^2 + (z_p \cdot \omega_r)^2}
$$

In these conditions the Gopinath rotor flux observer is completely determined.

Next, in order to determine the adaptation mechanism used in order to estimate the rotor speed, we will consider as a reference model the „stator currents - rotor fluxes” model of the induction motor and as an adjustable model, the model of the Gopinath rotor flux observer.

This problem is frequently treated in the literature of the non-linear systems, being exactly the configuration of the Lure problem, and of one of the problems treated by Popov [3].

The expression that defines the adaptation mechanism is:

$$
\dot{\omega}_r = k_1 \cdot \left( e_{33} \cdot \psi_{qr} - e_{yq} \cdot \tilde{\psi}_{dr} \right) dt
$$

where: $e_{33} = i_{ds} - \hat{i}_{ds}$ and $e_{yq} = i_{yq} - \hat{i}_{yq}$.

Sometimes, instead of the adaptation law (4) the following form could be used:

$$
\dot{\omega}_r = K_R \cdot e_a + K_i \cdot \int e_a \cdot d\tau
$$

where: $e_a = e_{33} \cdot \psi_{qr} - e_{yq} \cdot \tilde{\psi}_{dr}$; $K_i = \frac{K_R}{T_R}$.

From the above relation one may observe that a new proportional component appears from the desire to have 2 coefficients that control the speed estimation dynamics.

This fact isn’t always necessary because very good results by using only expression (4) could be obtained.

In these conditions the Extended Gopinath observer is completely determined.

### 3 The Mathematical Description of the Vector Control System

The block diagram of the control system of the mechanical angular speed $\omega_r$ of the induction engine with a discreet orientation after the rotor flux (DFOC) is presented in figure 2.

In figure 2 were marked with B1 the control block of the speed control system with direct orientation after the rotor flux (DFCO) and with B2 the extended Gopinath estimator block (EGO).

Some of the equations that define the vector control system are given by the elements which compose the field orientation block and consist of:

- stator tensions decoupling block (C1Us):

  $$
u_{ua} = \frac{1}{b_{11}} \hat{b}_{11} \cdot \left( \hat{b}_{11} \cdot \psi_{qr} - a_{33} \cdot \psi_{dr} \right) + a_{31} \cdot \omega_r \cdot \hat{a}_{31}
$$

- PI flux controller (PI_ψ) defined by the $K_\psi$ proportionality constant and the $T_\psi$ integration time:

  $$
u_{u{\psi}_q} = \frac{1}{b_{11}} \hat{b}_{11} \cdot \left( \hat{b}_{11} \cdot \psi_{q{\psi}_q} + a_{31} \cdot \omega_r \cdot \hat{a}_{31} \cdot \tilde{\alpha}_{q{\psi}_q} \right) + \frac{K_i}{T_\psi} \cdot \left( \psi_{q{\psi}_q} - \hat{\psi}_{q{\psi}_q} \right)
$$

- couple PI controller (PI_Mε) defined by the $K_M$ proportionality constant and the $T_M$ integration time:

  $$
u_{i{\psi}_q} = \frac{1}{T_M} \cdot \left( K_M \cdot \hat{\alpha}_{q{\psi}_q} \cdot \psi_{q{\psi}_q} + \hat{\alpha}_{q{\psi}_q} \cdot \tilde{\alpha}_{q{\psi}_q} \right) + \frac{K_i}{T_M} \cdot \left( \psi_{q{\psi}_q} - \hat{\psi}_{q{\psi}_q} \right)
$$

- Flux analyzer (AF):
\[
\begin{align*}
\sqrt{\psi_1} &= \sqrt{\psi_{dl}^2 + \psi_{dq}^2} \\
\sin \lambda_y &= \frac{\psi_{dq}}{\sqrt{\psi_1}}; \cos \lambda_y = \frac{\psi_{dl}}{\sqrt{\psi_1}}
\end{align*}
\] (9)

- Current PI controller (PI_I) defined by the \(K_i\) proportionality constant and the \(T_i\) integration time:

\[
\frac{dx}{dt} = i_{d\lambda_i} - i_{d\lambda_i} - \dot{i}_{d\lambda_i},
\]
\[
v_{d\lambda_i} = \frac{K_i}{T_i} \cdot x_{q_i} + K_i \cdot (i_{d\lambda_i} - \dot{i}_{d\lambda_i})
\] (10)

\[
\frac{dx_{10}}{dt} = i_{q\lambda_i} - \dot{i}_{q\lambda_i},
\]
\[
v_{q\lambda_i} = \frac{K_i}{T_i} \cdot x_{10} + K_i \cdot (i_{q\lambda_i} - \dot{i}_{q\lambda_i})
\] (11)

- The calculation of the couple block (\(C_iM_c\)):

\[
M_c = K_a \cdot i_{q\lambda_i}
\] (12)

where: \(K_a = \frac{3}{2} z_p \frac{L_{m}^*}{L_i} ; \ z_p \) is the pole pairs number.

In these conditions the vector control system is completely defined.

### 3.1 Pipe flow control system design

The flow control system based on the modification of the speed of the centrifugal pump is presented in figure number 3.

![Fig. 3 Conventional representation of a flow RSA](image)

The following notations were used in figure 3:

- TD - flow transducer
- SPC - centrifugal pump
- \(L\) - the length of the pipe, from the pump to the flow transducer
- \(D\) - the interior diameter of the pipe
- \(\Delta P\) - the pressure drop on the length \(L\) of the pipe
- \(F\) - inlet flow of the oil
- \(F^*\) - prescribed flow for the control system
- PI FLOW - integral proportional type flow regulator
- DFOC SPEED - speed control system presented in figure 2.

One of the main problems in the practical implementation of a speed control system for an induction motor is the controller tuning. In present, the controllers tuning of the induction motors speed control systems is made only through experimental methods, and the time allocated for this type of tests is a really long one. The paper deals with the analytical tuning controllers through the method of repartition of zeros - poles and the symmetry criteria and module Kessler instance. [5]

Therefore, for the regulators composing block B2 of the speed control system the following analytical adjustment formulas are used.

- Current controller:

\[
T_i = \frac{1}{a_{11}}; \ K_i = \frac{1}{b_{11} \cdot T_{dl}}
\] (13)

- Flux controller:

\[
T_{\psi} = T_{d1}^*; \ K_{\psi} = \frac{T_{d1}^*}{2 \cdot L_m \cdot T_{dl}}
\] (14)

- Couple controller:

\[
T_M = T_{d1}^*; \ K_M = \frac{T_{d1}^*}{K_a \cdot \psi_{rl}^* \cdot T_{d2}^*}
\] (15)

- Speed controller:

\[
K_\omega = \frac{T_{d2}^* \cdot (1 + \rho^2)}{2 \cdot K_4 \cdot T_{d2}^*} ; \ T_\omega = 4 \cdot \frac{T_{d2}^* \cdot (1 + \rho^2)}{(1 + \rho^3)} ; \ \rho = \frac{T_{d2}^*}{T_s}
\] (16)

where:

\[
K_4 = \frac{1}{F} \quad \text{and} \quad T_s = \frac{J}{F}
\]

In the above mentioned formulas, \(T_{d1}^*\) and \(T_{d2}^*\) are two time constancies imposed considering they need to respect the following conditions:

\[
T_{d1}^* < T_s; \ T_{d2}^* \ll T_s \quad \text{and} \quad T_{d2}^* > T_{d1}^*
\] (17)

Another important and difficult problem in the design of a pipe flow control system is the tuning of the flow controller.

For the tuning of this type of controller, in the beginning, the transfer matrix of the speed control system presented in figure 2 will be determined. Therefore, all the equations defining the control system
The input vector \( u \) of expression (18) is:

\[
\begin{bmatrix}
-u_1 \\
u_2 \\
u_3
\end{bmatrix}
\]

The expression of the vector function of the mathematical model (18) is:

\[
f = [f_1]^T
\]

The following notations have been used in the above mentioned expressions:

\[
g_i = \frac{b_i^1 v_{d1} - h_i}{b_1^1}; \quad g_2 = \frac{b_i^1 v_{q1} + h_i}{b_1^1} 
\]

\[
v_{d1} = \frac{K_r}{T_r} x_{1d} + K_v f_1; \quad v_{q1} = \frac{K_i}{T_i} x_{10} + K_i f_{10}
\]

\[
h_1 = a_{13}^* x_{13} + a_{31}^* \frac{x_{12}^2}{x_{13}} + z_p g_3 x_{12}
\]

\[
h_2 = a_{14}^* z_p g_3 x_{13} + a_{31}^* \frac{x_{11}^2}{x_{13}} + z_p g_3 x_{11}
\]

\[
\alpha_{11} = -\frac{1}{T_s \sigma}; \quad \alpha_{12} = \frac{1 - \sigma}{\sigma}; \quad \alpha_{13} = \frac{L_p}{L_1 T_s \sigma}; \quad \alpha_{14} = \frac{L_m}{L_2 \sigma}
\]

\[
\beta_{11} = \frac{L_m}{L_s T_s \sigma}; \quad \beta_{31} = \frac{L_m}{L_3 \sigma}; \quad \beta_{33} = \frac{1}{T_s \sigma}; \quad \beta_{34} = \frac{1}{\sigma}
\]

\[
\sigma = 1 - \frac{L_p^2}{L_m L_s T_s \sigma}; \quad K_{m1} = \frac{3}{2} \frac{Z_p}{J} L_m; \quad K_{m2} = \frac{F}{J}; \quad K_{m3} = \frac{1}{J}
\]

Because the speed control system is nonlinear, for the determination of the transfer matrix the system will be linearized (18) around the balance point. [6] For the determination of the balance point the following nonlinear equation system will be solved using Newton’s method, for an imposed input vector and invariant in time.

\[
f_i(x, u) = 0; \quad i = 1...14
\]

The obtained balance point for the input vector \( u_N \), formed from the nominal input values of the control system, will be marked with \( b = [b_1]^T \). Considering these conditions, the linearized system is:
\begin{align*}
\frac{d\Delta x}{dt} &= A_L \cdot \Delta x + B_L \cdot \Delta u \\
\Delta y &= C_L \cdot \Delta x 
\end{align*}
\quad (29)

where:
\[
A_L = \begin{bmatrix}
\frac{\partial f_i(b,u)}{\partial x_j} \\
\vdots \\
\frac{\partial f_i(b,u)}{\partial x_j}
\end{bmatrix} ; \\
B_L = \begin{bmatrix}
\frac{\partial f_i(b,u)}{\partial u_k} \\
\vdots \\
\frac{\partial f_i(b,u)}{\partial u_k}
\end{bmatrix} ; \\
C_L = [0 0 0 0 1 0 0 0 0 0 0 0 0 0 0].
\]

Going to Laplace transform in initial conditions null in expression (29) we may explain the transfer matrix of the control system in figure 2. [6]

\[
G(s) = C_L \cdot [s \cdot I_{14} - A_L]^{-1} \cdot B_L 
\quad (30)
\]

The transfer matrix (30) is composed of three transfer functions linking the output of the control system with the three inputs of the vector (20).

Within the design of the flow controller the used transfer function is the one that links the output of the system to the first element of the input vector. This transfer function will be noted as follows:

\[
G_1(s) = \frac{\Delta o_i(s)}{\Delta o_i^*(s)} 
\quad (31)
\]

The fixed part of the flow control system will be explained based on this transfer function. Therefore the transfer function of a small pipe \((L \approx D)\) will be presented. [4]

\[
G_2(s) = \frac{\frac{\Delta F(s)}{\Delta P(s)}}{\frac{F_0}{\Delta P_0}} = \frac{k_p}{T_p s + 1} 
\quad (32)
\]

where: \(k_p\) is the amplification factor \(k_p = 0.5\) and \(T_p\) is the delay constancy of the resistive tube:

\[
T_p = \frac{\alpha A L}{F_0}; \quad A = \pi \left( \frac{D}{2} \right)^2; \quad \alpha = \frac{D}{f \cdot L} 
\quad (33)
\]

In the last expression of relation (33), \(f\) is the coefficient of friction determined considering the Reynolds number.

Expression (32) is obtained following the linearization of the equation on a resistive tube based on Taylor’s theory around the balance point \((F_0; \Delta P_0)\).

\[
\begin{align*}
\Delta P(t) &= \Delta P_0 + \Delta(A\Delta P(t)) = \Delta P_0 + \Delta P(t) \\
F(t) &= F_0 + \Delta F(t)
\end{align*}
\quad (34)
\]

As the flow transducers dynamically act as first order aperiodic systems we may say that the transfer function of the flow transducer is:

\[
G_3(s) = \frac{F_0(s)}{F(s)} = \frac{k_T}{T_T s + 1} 
\quad (35)
\]

where: \(k_T\) is the amplification factor and \(T_T\) is the delay constancy of the transducer. In the design of the flow controller, this constancy \(T_T\) is neglected because it has a small value considering the time constancies dominated by the process.

Due to the fact that there is a difference between the flow and angular speed of the centrifugal pump actuating motor, there is a direct proportion, and we may say that the motor-pump ensemble is defined by the following transfer function:

\[
G_{PP}(s) = \frac{\frac{\Delta o(s)}{\Delta P(s)}}{F(s)} = K_{SP} \cdot G_1(s) 
\quad (36)
\]

where: \(K_{SP}\) is the slope characteristic to flow-speed of the centrifugal pump.

In these conditions, the transfer function of the fix slope of the system is:

\[
G_{FF}(s) = \frac{\frac{\Delta o(s)}{\Delta P(s)}}{F(s)} = 2 \cdot K_{SP} \cdot G_1(s) \cdot G_2(s) \cdot G_3(s) 
\quad (37)
\]

In the above mentioned relation the coefficient multiplying the slope of the flow-speed characteristic of the centrifugal pump appears due to the equation defining the pressure drop on a resistive tube:

\[
\Delta P = \frac{\rho \Delta F^2}{2 \alpha A^2} 
\quad (38)
\]

In these conditions the regulating system presented in figure 4 may have the following form:

Fig. 4. Flow regulating system

In these conditions, based on the transfer function (37) and on the pole-zero repartition method the flow controller may be easily tuned.

3.1.1 Application

In order to give examples for what we have presented in the paper we will design an oil flow control system on a main pipe.

Thus, a main pipe will be considered with an internal diameter \(D = 0.2\ m\) and the flow transducer will be placed at a distance \(L = 1\ m\). A 20°C oil operating temperature has been considered for the design.

The centrifugal pump used is a LQRY 150-125-270 type pump manufactured by Shanghai Pate Pump MFG.CO. [7] with a maximum capacity of \(400[\text{m}^3/\text{h}]\).
The actuating motor used has the following electrical and mechanical parameters:

\[ \begin{align*}
P_N &= 160\text{[kW]} ; 
U_N &= 400\text{[V]} ; 
n_N &= 1487 \, \text{[rot/min]} ; 
\end{align*} \]

\[ \begin{align*}
f_N &= 50\text{[Hz]} ; 
z_p &= 2 ; 
R_s &= 0.01379\text{[Ω]} ; 
\end{align*} \]

\[ \begin{align*}
R_t &= 0.007728\text{[Ω]} ; 
L_s &= 0.007842\text{[H]} ; 
\end{align*} \]

\[ \begin{align*}
L_t &= 0.007842\text{[H]} ; 
L_m &= 0.00769\text{[H]} ; 
\end{align*} \]

\[ J = 2.9\text{[Kg·m^2]} ; 
F = 0.05658 \, \text{[N·m·s]} \cdot 2 \, \text{[rad]} . \]

The length of the main pipe for the transport of petroleum product is equal to 1\text{[km]}.

For the design of the controllers from speed control systems the following values of the constancies \( T_{d1} \) and \( T_{d2} \) have been used:

\[ T_{d1}^* = 1\text{[msec]} ; 
T_{d2}^* = 7.5\text{[msec]} . \] (39)

Following the tuning of flow controller based on the described procedure in this paper the following values of the coefficients defining the controller have been obtained: \( K_d = 5 \) ; \( T_g = 0.0005\text{[sec]} \).

For the simulation of flow control system with the modification of the speed of the centrifugal pump, for the flow \( F \) different values have been imposed, respectively 50, 80, 150 and 200 \text{[m}^3\text{/h]}]. The simulation of the control system has used Matlab-Simulink software, and following the simulation the following graphs have been obtained:

Considering all the previously presented it is observed that the control system has a very good dynamics.

\section*{4 Conclusions}

The used concepts and ideas in the design of the control system presented in the paper may be used and developed for other types of flow control systems as well.

Because of the real advantages of the sensorless control of speed and good dynamic control performances of the new regulating system we may say that implementing and using the system is a real advantage.

\textbf{References:}


