

Velocity Feedback Control of a Mechatronics System

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Abstract— Increasing demands in performance and quality make drive systems fundamental parts in the progressive automation of industrial process. The analysis and design of Mechatronics systems are often based on linear or linearized models which may not accurately represent the servo system characteristics when the system is subject to inputs of large amplitude. The impact of the nonlinearities of the dynamic system and its stability needs to be clarified.

The objective of this paper is to present a nonlinear mathematical model which allows studying and analysis of the dynamic characteristic of an electro hydraulic position control servo. The angular displacement response of motor shaft due to large amplitude step input is obtained by applying velocity feedback control strategy. The simulation results are found to be in agreement with the experimental data that were generated under similar conditions.

Index Terms— Mechatronics System, Velocity Feedback, Servo Motor

I. Introduction

Hydraulic systems are commonly used in industries where high levels of power and accurate positioning are required to manipulate heavy objects or to exert force on environment. Examples include pick and place robots, positioning of aircraft control surfaces, flight simulators, and heavy-duty manipulators like excavators and feller punchers, hydraulic systems consists of components such as valves, actuators and pumps whose dynamic characteristics are complex, nonlinear and time varying, [1].

The nonlinearity arises from many sources including relationship between pressure and flow, flow deadband and saturation, change of fluid volume in different parts of the stroke, changes in the temperature-sensitive bulk modulus of the working fluid, and directional nonlinearity of the single rod actuator. Other factors that influence the performance of hydraulic functions

are friction between moving parts or changes of supply pressure and load, [2, 3, 4]

The modeling of the electro hydraulic components is to be the prime importance factor in the design of electro hydraulic system. In practice it is often difficult to formulate a sufficiently accurate model of an electro hydraulic system, [5].

The traditional approach for designing a controller for a given nonlinear systems is to first linearize the model equations, and then develop the control algorithm using well-established linear control design techniques. Although this method works well for some systems, there are other systems for which a linear model does not provide an adequate description of the actual system and therefore does not produce acceptable controller performance.

Nonlinear analysis techniques (such as Lyapunov method, [6]) do exist for verifying stability; however, these methods generally do not provide any indication of the system performance or how to improve the controller once a stable controller found. Hence, these methods are useful for verifying controllers, but are of little benefit in the design process.

J.M. Finney, etl, [7] implemented an adaptive pole placement controller for position regulation of a single rod hydraulic cylinder. However, since pole assignment schemes adjust only the position of the closed-loop poles, they cannot give good response characteristics for tracking cases.

Richard D. A., etl, [8] presented equations of motion for an electro hydraulic positioning system and experimental applications of the successive Galerkin approximation synthesis strategy to the system under a varying of operating conditions are compared with simulation results. However they used linearized model equations in their simulation.

The main objective of this paper is to present a nonlinear mathematical model which allows simulation and analysis of the dynamic characteristics of an electro hydraulic position control servo system. Also improving the system band width by adding the velocity feedback as a minor loop in the system is successfully

implemented. In the dynamic model, two major nonlinearities are considered: (1) pressure/flow characteristics associated with the spool valve, and (2) Coulomb-friction, which is already present or intentionally introduced in the valve motor load. The model includes valve dynamics as well as the effect of oil compressibility and actuator leakage. The dynamic response for the angular displacement of the servo system with position feedback as well as with velocity feedback is obtained.

The remainder of this paper is organized as follows: Section 2 gives the actuator mathematical model. Section 3 describes the used control strategy. Section 4 presents the results and discussions. Conclusion and future work are given in the final section.

II. Mathematical Modeling

The closed loop electro hydraulic position control system under consideration is shown in Fig. 1. A two-stage electro hydraulic servo valve is connected to a hydraulic rotary actuator by very short hoses. The closed-loop action is obtained by comparing the angular position of the motor shaft with the input signal by the interfaced circuit. A tachogenerator is used to measure the angular velocity, which can be used as a feedback signal to the input of the servovalve drive amplifier.

The electro hydraulic valve consists of a first stage nozzle-flapper valve, and a second-stage 4-way spool valve. The valve drive amplifier has a gain of 100 mA/V.

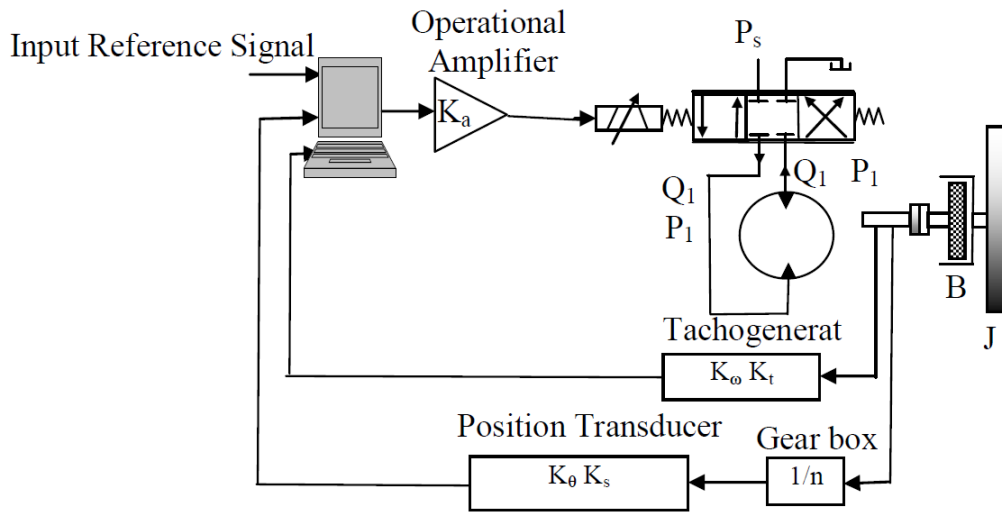


Fig. 1: Schematic Diagram of the Servosystem

The model is derived on the assumption that an inertially loaded rotary motor is controlled by the electro hydraulic servo valve. The steady-state valve model can be represented by the following relation, [9].

$$Q = K_x V_x \operatorname{sgn} \left[1 - \left(\operatorname{sgn} V_x \right) \frac{P_L}{P_s} \right] \sqrt{\left| 1 - \left(\operatorname{sgn} V_x \right) \frac{P_L}{P_s} \right|} \quad (1)$$

with

$$V_x = \begin{cases} V + B & \text{when } V < -\beta \\ 0 & \text{when } -\beta \leq V \leq +\beta \\ V - B & \text{when } V > +\beta \end{cases}$$

The dynamic performance of the servo valve is described by a first-order time lag and is given by:

$$\tau \frac{dQ}{dt} + Q = K_x V_x \quad (2)$$

Equations (1) and (2) are combined to yield a dynamic valve model as

$$\tau \frac{dQ}{dt} + Q = K_x V_x \operatorname{sgn} \left[1 - \left(\operatorname{sgn} V_x \right) \frac{P_L}{P_s} \right] \sqrt{\left| 1 - \left(\operatorname{sgn} V_x \right) \frac{P_L}{P_s} \right|} \quad (3)$$

The hydraulic motor is modeled by considering the rotary motor arrangement shown in Fig. 1, as well as by taking into account oil compressibility and leakage across the motor. Using the principal conservation of mass yields

$$Q = V_m \frac{d\theta}{dt} + \frac{V_c}{4K_h} \frac{dP_L}{dt} + L_e P_L \quad (4)$$

The equation of motion of the load can be given by

$$P_L V_m = J \frac{d^2 \theta}{dt^2} + B \frac{d\theta}{dt} + T_c \operatorname{sgn} \dot{\theta} \quad (5)$$

2.1 State Space Model

Definitions of the state variables and inputs of the system are given below:

States:

$$\begin{bmatrix} x_1 & x_2 & x_3 & x_4 \end{bmatrix} = \begin{bmatrix} \theta(t) & \dot{\theta}(t) & P_L(t) & \dot{P}_L(t) \end{bmatrix} \quad (6)$$

Inputs:

$$\begin{bmatrix} u_1 & u_2 \end{bmatrix} = \begin{bmatrix} V_i(t) & P_s \end{bmatrix} \quad (7)$$

Applying the states definition to the system of nonlinear (1-5), after manipulation, results in the state variable model as follows:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -\frac{B}{J}x_2 + \frac{V_m}{J}x_3 - \frac{T_c}{J}\text{sgn } x_2 \\ \dot{x}_3 &= x_4 \end{aligned}$$

$$\begin{aligned} \dot{x}_4 &= -x_1 \left[\frac{4K_h}{\tau V_c} \frac{K_x K_a k_\theta k_s}{n} \text{sgn} \left\{ 1 - (\text{sgn } V_x) \frac{x_3}{u_2} \right\} \sqrt{1 - (\text{sgn } V_x) \frac{x_3}{u_2}} \right] \\ &+ x_2 \left[\frac{4K_h}{\tau V_c} \left(\frac{\tau V_m B}{J} - V_m - K_x K_a k_t k_\omega \text{sgn} \left\{ 1 - (\text{sgn } V_x) \frac{x_3}{u_2} \right\} \sqrt{1 - (\text{sgn } V_x) \frac{x_3}{u_2}} \right) \right] \\ &- x_3 \left[\frac{4K_h V_m^2}{J V_c} + \frac{4K_h L_e}{\tau V_c} \right] - x_4 \left[\frac{1}{\tau} + \frac{4K_h L_e}{V_c} \right] + \frac{4K_h V_m}{J V_c} T_c \text{sgn } x_2 \\ &+ \left(\frac{4K_h}{\tau V_c} \right) K_x K_a u_1 \text{sgn} \left\{ 1 - (\text{sgn } V_x) \frac{x_3}{u_2} \right\} \sqrt{1 - (\text{sgn } V_x) \frac{x_3}{u_2}} \end{aligned} \quad (8)$$

The state variables model represented by (6-8) is of the nonlinear form

$$\dot{x}(t) = f[x(t), u(t)] \quad (9)$$

The initial conditions of the state variables are given by:

$$\begin{bmatrix} x_1(0) & x_2(0) & x_3(0) & x_4(0) \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \end{bmatrix} \quad (10)$$

The parameters of the system appearing in the state-variables model are given in Table 1. The experimental work was carried out at the Automatic Control Laboratory of Assiut University, Egypt.

Table 1: System physical Parameter

τ	valve time constant	s	2.3×10^{-3}
K_a	operational amplifier gain		-1
K_x	valve flow gain at $P_1=0$	$m^3/s/v$	-1.36×10^{-4}
V_c	volume of hoses	m^3	20.5×10^{-6}
V_m	motor displacement	m^3/rad	0.716×10^{-6}
L_e	leakage coefficient	m^3/Ns	2.8×10^{-11}
K_h	hydraulic bulk modulus	N/m^2	1.4×10^9
B_e	viscous coefficient	Nms/rad	2.95×10^{-3}
J	motor inertia	Nms^2/rad	3.4×10^{-3}
T_f	coulomb-friction	$N.M$	0.225
K_t	tachogenerator constant	$v/rad/S$	0.026
K_s	position transducer constant	v/rad	3.44
n	gear ratio		7.5

III. Structure of Control System

The control valve, which is a standard type 32- Moog servovalve, is connected to a A084 nine-axial piston Moog-Donzelli hydraulic motor. The feedback action is implemented by using a tachogenerator and a position transducer to measure the shaft speed and position.

A synchronal error channel was used to sense the position of the hydraulic motor shaft, compare it to the input signal and derive an error signal. This forms the major feedback loop which is described as:

$$e_1(t) = V_i(t) - \frac{K_s K_\theta \theta}{n}$$

The velocity feedback is generated by using a tachogenerator which derives a voltage signal and feeds it back to a differential amplifier, thus forming the minor loop. This action is given by:

$$e_2(t) = -K_\omega K_t \dot{\theta}$$

At first a single control loop is applied. A proportional controller is adopted for controlling the motor position with a step input, whereas the controller is defined by the following equation:

$$I = K_p K_a e_1(t)$$

In order to improve its dynamic response two loops are adopted, the minor feedback loop is formed by applying a tachogenerator to measure the motor speed and generate a feedback signal. On the other hand, a position transducer is adopted to measure the position and use the generated signal to form the major loop, which contributes significant damping effect to the system, whereas the controller is defined by the following equation:

$$I = K_a (K_p e_1(t) + K_d e_2(t))$$

IV. Results and Discussions

The first step in control system design is to obtain the mathematical model, which, describe the dynamics of the plant to be controlled. More accurate dynamic model of the plant led to better control system performance.

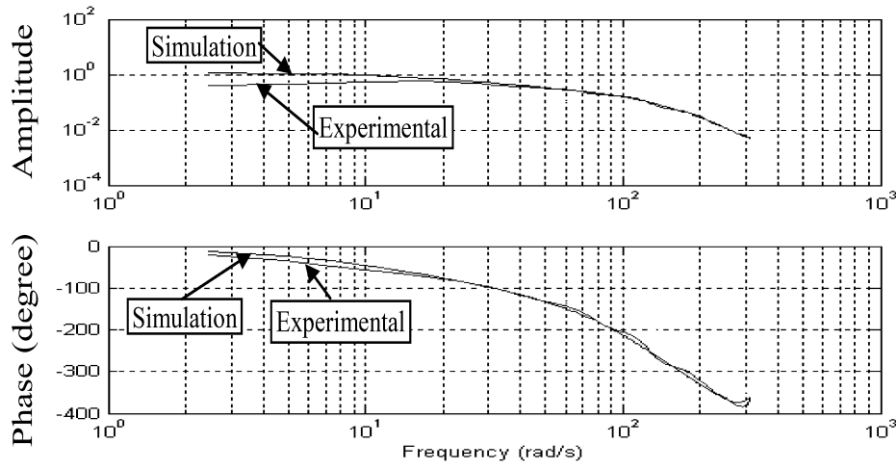


Fig. 2: System bode diagram for the Experimental and Simulation

To simplify the estimation of the model parameters, a closed-loop identification scheme is used. The simulated model bode diagram is presented in Fig.2, and it has good agreement with the experimental one.

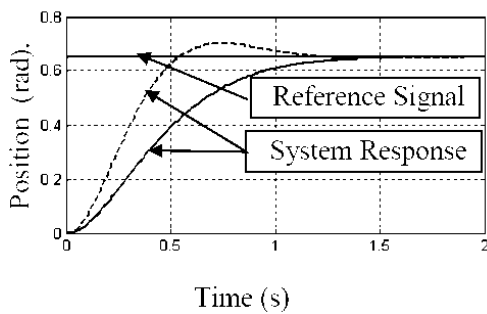


Fig. 3: System step response based on P-controller action

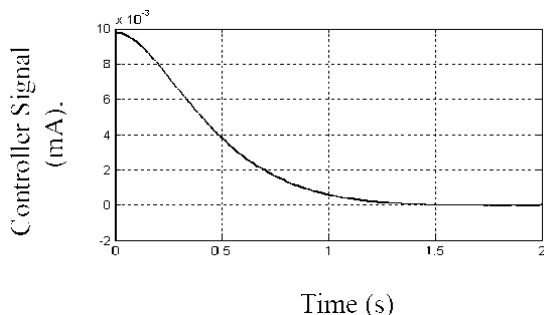


Fig. 4: P-controller action signal

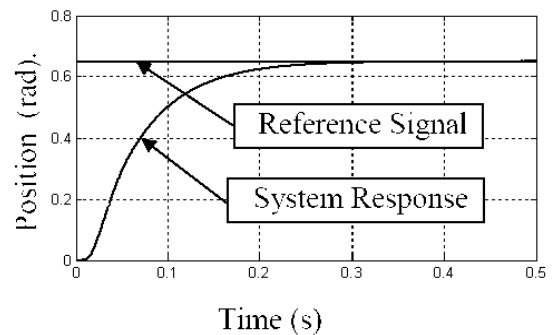


Fig. 5: System step response based on PD-controller action

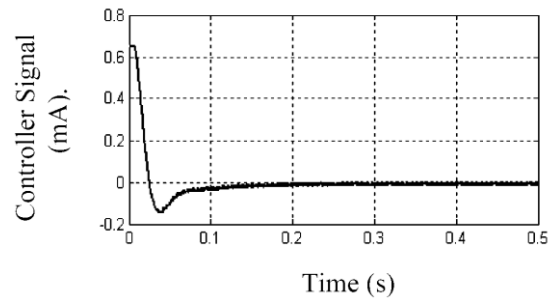


Fig. 6: PD-controller action signal

The desired response is designed without either overshoot or steady state error with smaller rise time as possible. The simulations were performed with a

constant supply pressure of 70 bar connected to a hydraulic rotary motor in two different closed loops. Fig. 3 obtained by comparing the input step signal with the synchronic error channel which perform single control loop (P-control), it is found that the rise time is about 1.5 sec. with no overshoot and steady state error is zero. However, if we improve the response rise time an overshoot appeared. Its corresponding control action is shown in Fig. 4.

The other applied with velocity feedback in addition to the position feedback, while the rise time is improved to be 0.3 sec., the overshoot kept to be zero as shown in Fig. 5, and the corresponding control signals is cleared at Fig. 6.

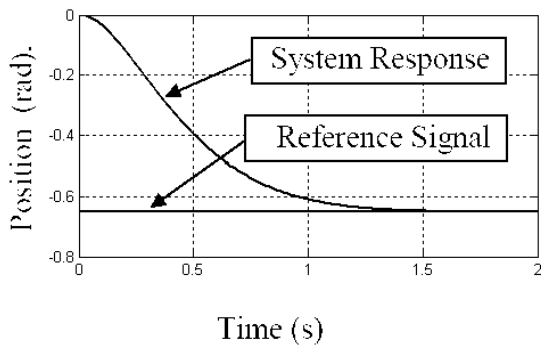


Fig. 7: System step decrease response based on signal P-controller action

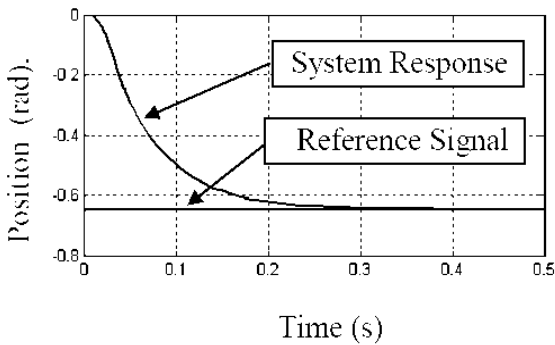


Fig. 8: System step decrease response based on signal PD-controller action

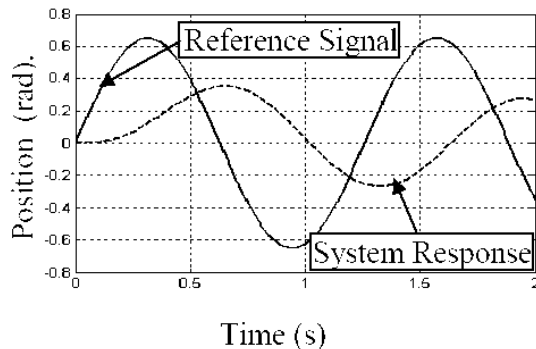


Fig. 9: System sin input response based on signal P-controller action

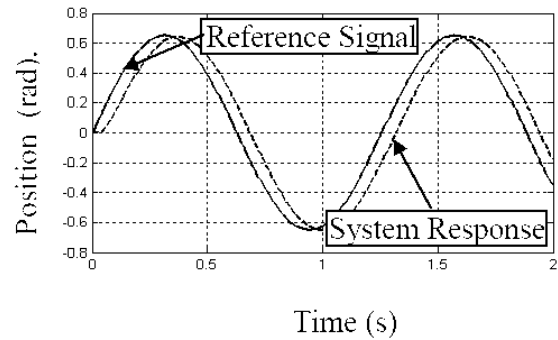


Fig. 10: System sin input response based on signal PD-controller action

Results presented so far in this paper are obtained with a step increase in the reference position. It is of interest to obtain the system response due to a step decrease in speed, in order to throw more light on the complicated role played by motor dry friction. The transient response of the system due to a step decrease in the reference position from 0.0 to -0.65 amplitude is displayed in Fig. 7 and 8 for the motor proportional control loop and for velocity feedback control loops together, respectively.

Another good application with sinusoidal input signal as a continuous motion can test the following ability with the control loops is shown in Figs. 9 and 10.

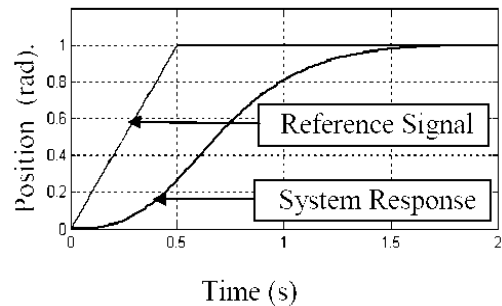


Fig. 11: System ramp input response based on signal P-controller action

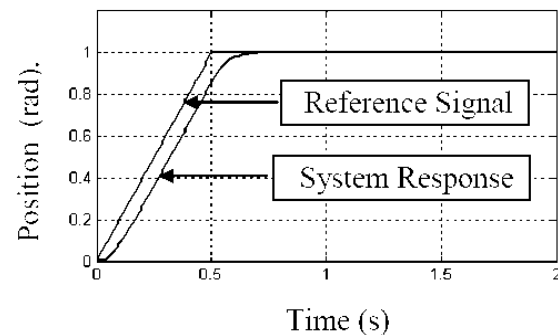


Fig. 12: System ramp input response based on signal PD-controller action

The ramp input is used in many applications, such as for tape drives of cutting tools. In order to follow such a command input, the controller must be able to deal

with both step and ramp commands (the step command corresponds to the constant speed). In Figs. 11 and 12 the response of system due to ramp input. It is completely clear that the system response has been improved by using velocity feedback control strategy.

V. Conclusions

The dynamic response of a Mechatronics speed control servo system was analyzed in order to throw more light on the complicated role played by the actuator nonlinearities. The following conclusions are derived from the experimental and simulated results:

- Good agreement in the system responses due to the random input which clear the precision of the simulated mathematical nonlinear model.
- Using different input signal prove that the velocity feedback loop in addition to the position feedback signal gave better dynamic response.
- Applying intelligent control system for positioning the electro hydraulic servo motor is under study as a promise target of our work.

Nomenclature

B	viscous damping coefficient, $N.m.s/rad$
B_c	Coulomb-friction coefficient, $N.m.s/rad$
B_e	viscous damping coefficient, $N.m.s/rad$
J	load inertia, $N.m.s^2/rad$
K_A	transfer function gain
K_a	operational amplifier gain
K_h	bulk modulus of fluid, N/m^2
K_p	valve pressure gain, $m^5/n.s$
K_x	valve flow gain at $P_l = 0$. $m^3/s/v$
K_s	position transducer constant, $V/rad/s$
K_t	tachogenerator constant, $V/rad/s$
K_θ	position feed back gain
K_ω	velocity feedback gain
L_e	equivalent leakage coefficient, $m^5/N.s$
n	reduction gear ratio
P_1, P_2	pressures at actuator ports, N/m^2
P_L	load pressure, N/m^2
Q_1, Q_2	inlet and outlet flow of the actuator, m^3/s
Q	mean flow rate, m^3/s
S	Laplace operator
t	time, s

T_c	coulomb -friction, $N.m$
V_c	volume of oil in motor and hoses, m^3
V_i	input voltage to the system, V
V_m	motor displacement, m^3/rad
V_x	valve drive voltage, V

Greek Symbols

τ	valve time constant, s
θ	shaft position, rad
ω	angular frequency, rad/s

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Authors' Profile



Dr. Ayman A. Aly, B.Sc. with excellent honor degree (top student), 1991 and M.Sc. in Sliding Mode Control from Mech., Eng., Dept., Assiut University, Egypt, 1996 and Ph.D. in Adaptive Fuzzy Control from Yamanashi University, Japan, 2003.

Nowadays, he is the head of Mechatronics Section at Taif University, Saudi Arabia since 2008. Prior to joining Taif University, He is also one of the team who established the "Mechatronics and Robotics Engineering" Educational Program in Assiut University in 2006. He was in the Managing and Implementation team of the Project "Development of Mechatronics Courses for Undergraduate Program" DMCUP Project-HEEPF Grant A-085-10 Ministry of Higher Education – Egypt, 2004-2006.

The international biographical center in Cambridge, England selected Ayman A. Aly as international educator of the year 2012. Also, Ayman A. Aly was selected for inclusion in Marquis Who's Who in the World, 30th Pearl Anniversary Edition, 2013.

In additions to 5 text books, Ayman A. Aly is the author of more than 60 scientific papers in Refereed Journals and International Conferences. He supervised some of MSc and PhD Degree Students and managed a number of funded research projects.

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Research interests: Robust and Intelligent Control of Mechatronics Systems, Automotive Control Systems, Thermofluid Systems Modeling and Simulation.

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