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Computer Aided Design for Dynamic Modeling and Control of Closed Loop Mechanisms

Z. AFFI, L. ROMDHANE
Laboratoire de Génie Mécanique, Département de Génie Mécanique
Ecole Nationale d'Ingénieurs de Monastir
Avenue Ibn Eljazzar, 5019 Monastir
TUNISIA
[zouhaier.affi, lotfi.romdhane]@enim.rnu.tn
http://www.enim.rnu.tn

Abstract

In this work, computer aided method for dynamics modeling and control of closed loop mechanisms is presented. A crank slider mechanism is used to illustrate our approach. The proposed method allows us to analyze the dynamic behavior of the mechanism in a simple and rapid way. This tool allowed us to show that for a mechanism with general inertia characteristics, the PD controller does not yield an acceptable behavior, i.e., a non zero steady state error, and it is necessary to have a PID controller to eliminate this error. Then the crank slider mechanism is redesigned by applying a negative mass distribution approach to obtain a statically balanced mechanism, which exhibits a simpler dynamic behavior. We showed that in this case the PD controller is sufficient to eliminate the steady error. Hence, the mechanical design of parallel robots can be improved in order to simplify the control strategy. This goal presents the main advantage of the proposed approach.

A sensitivity analysis, using the Taguchi method, is then presented to analyze the effect of some physical parameters on the dynamic behavior of the system. We show, for instance, that the steady state error is very sensitive to the location of the center of gravity of the crank.

Keywords: *Dynamic behavior, Crank slider mechanism, PD controller, PID controller, Mechatronic systems, sensitivity analysis.*

1. Introduction

Closed loop mechanisms are present in virtually all mechanical systems. These mechanisms are usually synthesized to follow a pre-specified trajectory. Synthesis of a mechanism controller and the simulation of the behavior of the whole mechatronic system are based on the dynamic model of the mechanical system.

Comparing with the open chain mechanism, the dynamic equations of the closed-chain mechanism include more system parameters and are far more complex for the same degrees of freedom. The complicated dynamic model generates a grate difficulty for controller design. Parallel mechanisms can be considered as mechanical systems with one or more closed loop mechanisms.

The main characteristics of parallel mechanisms are high accuracy, high load capacity, high rigidity and quickness. These characteristics require a controller with excellent performance. However, the performances of the controller are greatly influenced by the dynamic model of the mechanical system.

The dynamics of closed loop mechanisms is usually highly non linear and complex. Several methods reported in the literature are proposed to derive these dynamic models [2, 6, 8]. All these works are based on deriving analytical models of the mechanical system. However, the complexity of these models makes them inadequate for any real time applications.

To overcome this difficulty, some authors [1, 4, 7] suggested the use of the Design for Control (DFC) concept which consists in designing an appropriate structure so that it can result in a "simple" dynamic model. This model was derived analytically for simple cases using Lagrangian formalism.

This paper suggests the use of a general mechatronic design approach, i.e., the design for control (DFC) approach, to handle this problem. The software ADAMS, was used to generate automatically the dynamic model of a closed loop mechanism. This model was then fed under MATLAB/Simulink environment to be analyzed. The model is used in the control scheme to predict the behavior of the system using a PID controller. The slider crank mechanism is used to illustrate this method.

The ADAMS model is parameterized to facilitate the modification of the model. Any changes in the model under ADAMS are automatically taken into account in the dynamic model under Simulink.

ADAMS and MATLAB/Simulink were successfully interfaced to yield a powerful tool to model and analyze the dynamic behavior of mechatronic systems. Also, the

CAD model can improve the model accuracy by taking advantage of the automatic calculation of the inertia properties of all the parts of the mechanism [5, 6].
This tool allowed us to easily modify several designs and investigate their effect on the dynamic behavior of the system. In particular, it is shown that a balanced closed loop mechanism needs only a simple PD controller to yield a zero static error. Whereas, a more complicated controller, i.e., PID controller, is necessary to have the same behavior for non balanced mechanism. Moreover, the balanced mechanism requires less energy to execute the same motion, than the non balanced mechanism [4].
This paper is organized as follows. Section II contains the description of the ADAMS model of the slider crank mechanism. Section III contains the Simulink control scheme of the mechatronic system. Section IV contains some simulation results. Section V presents the sensitivity analyses of the mechatronic system based on the *design of experiments*. Finally, section VI contains some concluding remarks.

2. ADAMS Model

Many techniques used to derive the dynamic model of the mechanism are based on the Lagrange equations. In the case of closed loop mechanisms, this model is difficult to obtain and is very complex. In some cases, when the mechanism contains several loops, the dynamic model can not be obtained analytically. The software ADAMS can present an interesting alternative to derive numerically the dynamic model for complex closed loop mechanism. Moreover, the ADAMS model can be parameterized to investigate in a simple way the dynamic behavior of different designs.

2.1. Description of the crank slider mechanism

Table 1 and Table 2 show the geometric and dynamic characteristics of the different links of the crank slider mechanism. Fig.1 shows the crank slider model.

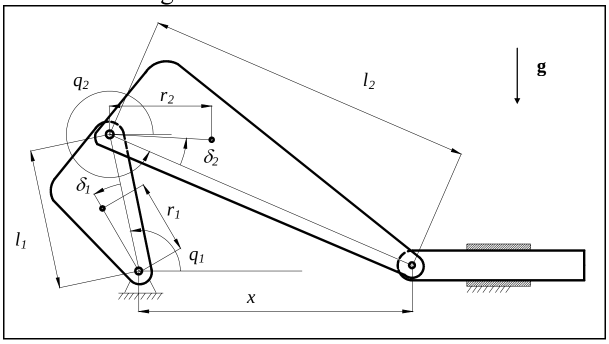

Fig. 1: schematic of the crank slider mechanism

This parameterized model is convenient for testing different designs by simply modifying the values of the different parameters of the system (r_i, δ_I, I_{zzi}; i=1,2) . Two designs are modeled under ADAMS (Fig. 2) and their dynamic models were determined in order to investigate their dynamic behavior. The geometric parameters of these mechanisms are the same and they are given in Table 1. This dimensions are arbitrary.
The input of the model is the torque applied at the joint between the crank and the ground. The output motion is the displacement of the slider. For a given displacement of the slider, we calculate the necessary torque to be applied on the crank.

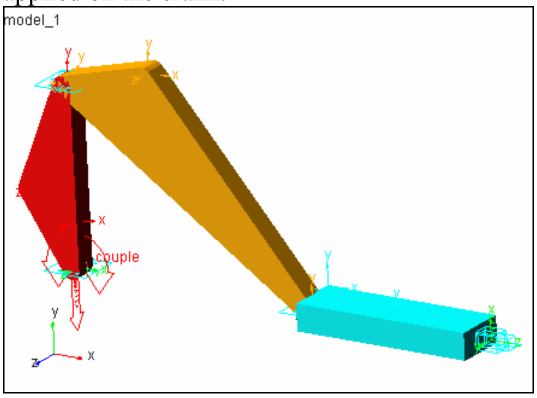

Fig. 2: ADAMS model of the crank slider mechanism

Table 1: Geometric parameters of the Crank slider mechanism

Parameter	**Crank**	**Connect. Rod**
Length (m)	$l_1 = 0.1$	$l_2 = 0.4$

The first design to be considered is the general case where the links have no particular geometry or mass distribution. The inertia parameters of each link are given in Table 2.

Table 2: Inertia parameters of the non balanced mechanism

Parameter	**Crank**	**Connect. Rod**	**Slider**
r (m)	$r_1 = 0.05$	$r_2 = 0.2$	
δ	$\delta_1 = 0$	$\delta_2 = 0$	
Mass (Kg)	$m_1 = 1$	$m_2 = 3$	$m_3 = 5$
Moment of inertia (10^{-3} Kg.m²)	$I_{zz} = 0.04$	$I_{zz} = 0.1$	

2.2. Statically balanced crank slider mechanism

The second design is a statically balanced crank slider mechanism. A statically balanced mechanism is one that has a stationary potential energy. It can be shown that this condition is equivalent to having the center of gravity of the whole mechanism fixed during the motion of the different links. In our case, the potential energy of the crank slider mechanism is given by:

$$U = m_1 r_1 g \sin(q_1+\delta_1) + m_2 g\left(l_1 \sin(q_1) + r_2 \sin(q_2+\delta_2)\right) \quad (1)$$

where the parameters are defined in Fig. 1.
Since this mechanism has only one degree of freedom (q_1), we can substitute the value of the variable (q_2) using the first solution of the following relation:

$$q_2 = \left\{ \arcsin\left(-\frac{l_1 \sin(q_1)}{l_2}\right), \pi - \arcsin\left(-\frac{l_1 \sin(q_1)}{l_2}\right) \right\} \quad (2)$$

The potential energy can then be expressed solely as a function of (q_1) as:

$$U = m_1 r_1 g \sin(q_1+\delta_1) + m_2 g\left(l_1 \sin(q_1) + r_2 \sin\left(\arcsin\left(-\frac{l_1 \sin(q_1)}{l_2}\right)+\delta_2\right)\right) \quad (3)$$

The balancing condition can be expressed as:

$$\frac{dU}{dq_1}=0 \tag{4}$$

where:

$$\frac{dU}{dq_1}=m_1 r_1 g\cos(q_1+\delta_1)$$

$$+m_2 g l_1\cos(q_1)\left(1-\frac{r_2\cos\left(\arcsin\left(-\frac{l_1\sin(q_1)}{l_2}\right)+\delta_2\right)}{l_2\sqrt{1-\left(\frac{l_1\sin(q_1)}{l_2}\right)^2}}\right)$$

This condition has to be satisfied for every value of the variable (q_1). One simple solution of the above relation can be given by:

$$\delta_1=\pi,\ r_1=\frac{m_2 l_1}{m_1} \tag{5}$$

$$r_2=0$$

The first two relations concern the location of the center of mass of the crank. The third condition means that the center of mass of the connecting rod has to be on the center of the revolute joint linking the crank to the connecting rod.
Table 3 presents the values of the parameters used in the simulation of the balanced crank slider mechanism.

Table 3: Inertia parameters of the balanced Crank slider mechanism

Parameter	**Crank**	**Connect. rod**	**Slider**
r (m)	$r_1 = 0.1$	$r_2 = 0$	
δ	$\delta_1 = 180$	$\delta_2 = -$	
Mass (Kg)	$m_1 = 3$	$m_2 = 3$	$m_3 = 5$
Moment of inertia (10^{-3} Kg.m²)	$I_{zz} = 0.07$	$I_{zz} = 0.125$	

We note that δ_2 is unspecified for the completely balanced mechanism.
Once the model is finalized, the dynamic model is generated by ADAMS and exported under the MATLAB/Simulink environment.

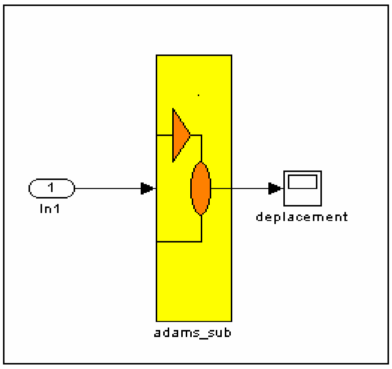

Fig. 3: the encapsulated dynamic model generated by ADAMS and shown under Simulink

The block adams_sub (Fig. 3) is the file exported from ADAMS containing the dynamic model with all the simulation parameters. Fig. 4 shows the inside of the block adams_sub.

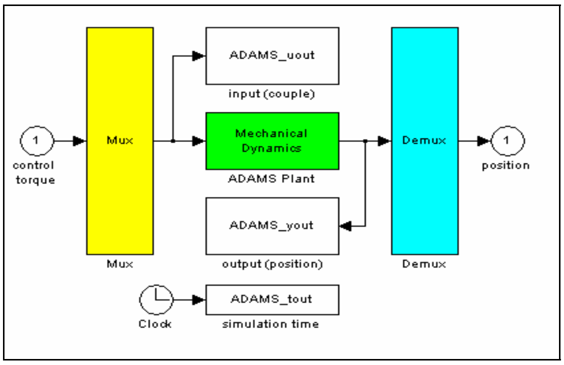

Fig. 4: architecture of the adams_sub block

Adams_uout, Adams_yout, Adams_tout are the input and output variables and the simulation time, respectively.
The block ADAMS Plant is the encapsulated dynamic model of the mechanism.

3. The Control Scheme

The Simulink library contains several control schemes, e.g., linear, non linear and fuzzy logic. Compared with all these control strategies, the advantage of the PD or PID controller is the fact that they do not require a deep knowledge of the dynamic model of the robotic system. Indeed, for closed loop parallel robots, the dynamic models are complex, highly non linear and need much effort to be derived. Therefore, we will be interested, in this paper, in the PD and the PID controller.
The PID control algorithm is given by:

$$\tau(t)=K_P e(t)+K_I\int e(t)dt+K_D\dot{e}(t) \tag{6}$$

where $\tau(t)$ is the driving torque generated by the controller, K_P, K_I, and K_D are the proportional, integral, and derivative gains, respectively. $e(t)$ is the error on the slider position given by:

$$e(t)=x_d-x(t) \tag{7}$$

where x_d is the desired position of the slider (constant), and $x(t)$ is its actual position.
$\dot{e}(t)=\dot{x}(t)$ is the slider velocity.
The control scheme used in this paper is shown on Fig.5. The gains of the PID controller are set by the user. To have a simple PD controller we just set the integrator gain to zero.

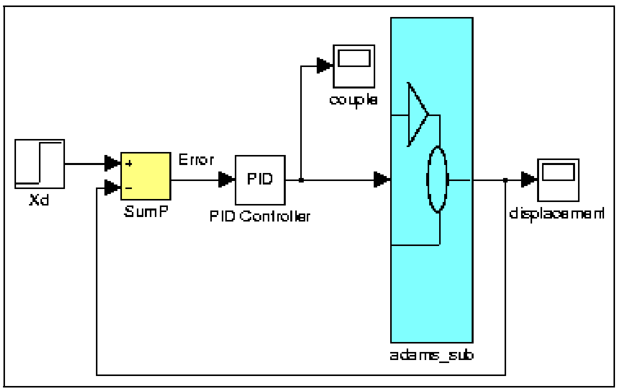

Fig. 5: the control scheme of the mechanism

4. Simulation and results

To investigate the effectiveness of our approach, simulation studies were carried out for the crank slider mechanism in two different cases. The first mechanism is a general one and has the parameters given in Table 1 and Table 2. The second one is a statically balanced crank slider mechanism whose parameters are given on Table 1 and Table 3.
In the simulation, the input crank was required to rotate to bring the slider from its original position (x = 780 mm) to the new position at x = 850 mm.
Fig. 6 shows the behavior of the mechanism given by Table 1 and 2, when the simple PD controller is used. It can be noticed that the static error is important (around 16 mm).

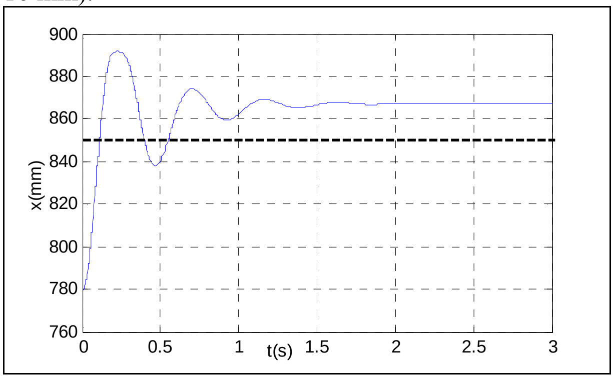

Fig. 6: Dynamic response of the crank-slider mechanism for (K_P=90, K_D=20)

This error is mainly due to the gravity. One way of eliminating this error is by using a PID controller or more sophisticated control strategies [3]. Fig. 7 shows this behavior using a PID controller.

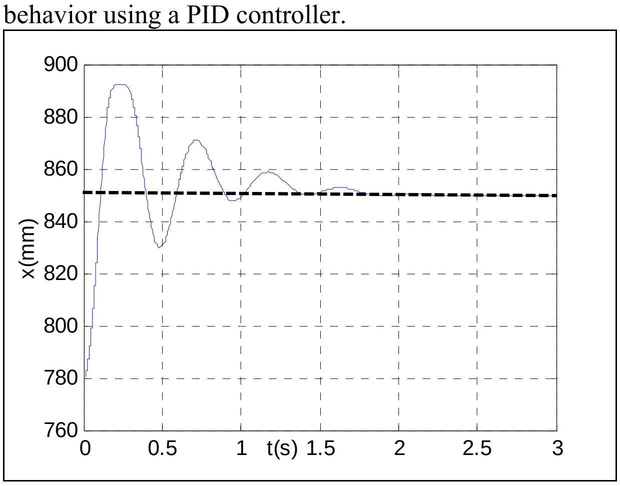

Fig. 7: Dynamic response of the crank-slider mechanism for (K_P=90,K_D=20, K_I=150)

However, in order to eliminate the steady state error, without having an integrator term in the controller, we should statically balance the mechanism. Hence, the gravitational term will disappear from the dynamic behavior of the mechatronic system. Consequently, a simple PD controller can yield satisfactory results without having the need to add the integrator. Fig. 8 shows such dynamic response.

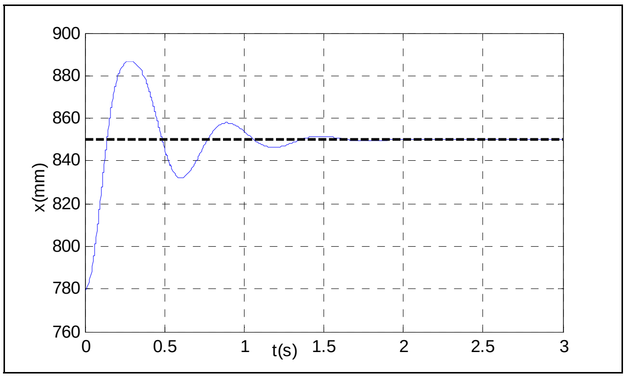

Fig. 8 : Dynamic response of the balanced crank-slider mechanism for (K_P=90,K_D=20)

One can notice that, with the same gains, the simple PD controller eliminated completely the steady state error and the slider reached the desired position. Moreover, Fig. 8 shows that in this case the overshoot is slightly reduced compared to the non balanced mechanism. Also, the stabilization time for the mechanism, to reach the final position, is smaller than in the previous case.
Using the developed interface, one can investigate the dynamic behavior of different designs and can modify the original design until acceptable dynamic behavior is obtained.

5. Sensitivity analysis of the mechatronic system

The goal of this study is to predict the steady state error, witch occurs when we use a PD controller, caused by the balancing parameters uncertainties. The problem is that we do not have an explicit relation that relates the steady state error to the fluctuation of the dynamic parameters of the system. To solve this problem, we will use the Taguchi method [9], which consists in developing a model to test the robustness of the system response. We will choose the following intervals of parameters uncertainties (fig 1):

$$r_1 = 100 \pm 5\,mm$$

$$r_2 = 0 \pm 5\,mm$$

$$\delta_1 = 180 \pm 9 \text{ deg}$$

$$\delta_2 = 0 \pm 9 \text{deg}$$

As we have the variation of 4 factors with 2 levels, we must carry out 16 tests. These different tests were done via the coupling ADAMS/Simulink interface (fig 5) and we have calculated the steady state error at each case.
If we analyze figure 9, we can see that the angle δ_1 is the most influent one on the precision of the mechatronic system. We can also notice that δ_2 has no significant effect on the steady state error. The degree of influence of the different parameters on the steady state error is presented in the Pareto chart (fig 10) at a level of 95% of confidence. It shows also that the only interaction that affects the response is the one between r_1 and δ_2.

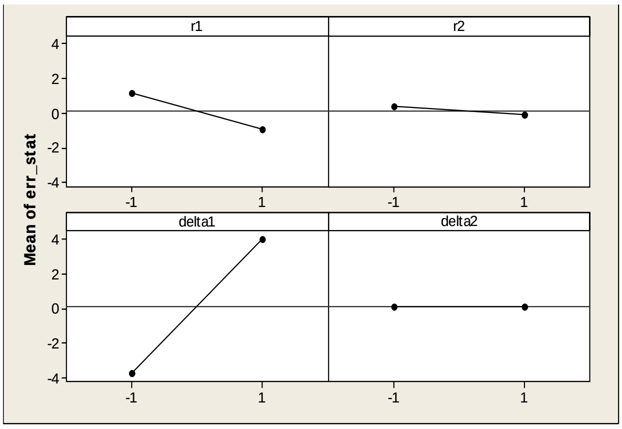

Fig. 9: Effect of the factors on Es

The model that gives the variation of steady state error according to the variables of Yates associated to the different factors are given by [9]:

$$E_s = E_0 + a_1 x_1 + a_2 x_2 + a_{12} x_1 x_2 + \dots \quad (8)$$

E_0 presents the average of the steady state errors of the different factors

a_i is the average effect of the factor *i*

a_{ij} is the average effect of the interaction between x_i and x_j

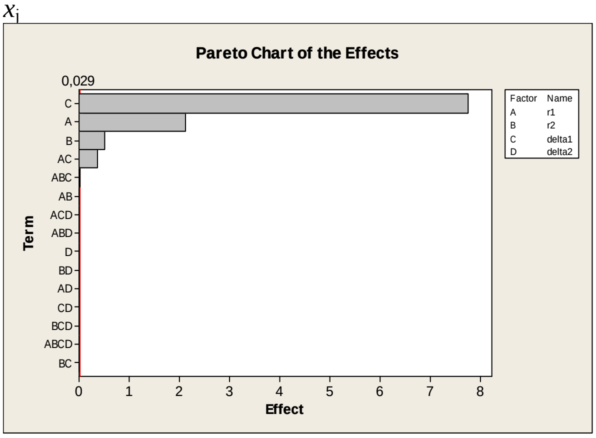

Fig. 10: Pareto chart of effect on Es

The analysis of the different tests, allows us to have a model that characterizes the effects of the different factors on the static error of the mechatronic system. This model is given by:

$$E_s = 16.625 - 0.941 r_1 - 0.052 r_2 + 0.022\delta_1 + 0.004 r_1 \delta_1 \quad (9)$$

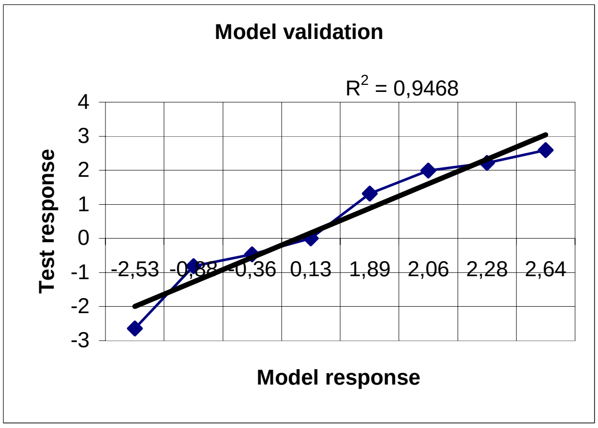

Fig 8: validation of the empiric model

To test the validity of this model, we conducted some simulations which will be compared with those given by the developed model (eq. 9). The correlation coefficient will be taken as an appreciation criterion. Different tests were done and the results are presented in fig. 8.

We can note that the correlation coefficient is $R^2 \approx 0.94$ which is acceptable. Therefore Eq 9 can be used with a fairly good approximation to calculate the necessary tolerances on the different parameters of the mechanism, for a give static error E_s.

6. Conclusion

In this work, we developed a program capable of interfacing the software ADAMS with the Simulink environment. This tool was then used to optimize the dynamic behavior of different mechanisms. A crank slider mechanism was used to illustrate our approach. This mechanism was designed under ADAMS environment then its dynamic model was exported to Simulink for simulation. We showed that the behavior obtained by a simple PD controller is not generally acceptable and a PID controller was necessary to have the adequate response. Then the crank slider mechanism was redesigned by applying a negative mass distribution approach to obtain a statically balanced mechanism. As a result the gravity term disappeared from the Lagrange equations of the system. This simplification of the dynamic equations resulted in an improvement of the dynamic behavior of the system. Indeed, a simple PD controller yielded satisfactory motion tracking performance.

A sensitivity analysis was then performed to generate an empirical model relating the static error of the system to its physical parameters. It was shown that the steady state error is very sensitive to the location of the center of gravity of the crank.

In conclusion, the proposed tool can be very useful to facilitate the approach known as “Design for control” (DFC). Complex mechanisms can be designed using ADAMS software and then simulated using the Simulink environment. The effect of any change in the physical parameters of the system can be analyzed immediately.

One of the main problems, that need further investigation, is the dynamic behavior of the whole mechatronic system, which incorporates the sevo-motor along with the mechanical system.

7. Acknowledgements

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Biography

Zouhaier AFFI, received his undergraduate degree (Diplôme National d'Ingénieur) in mechanical engineering from the Ecole Nationale d'Ingénieurs de Monastir (Tunisia), in 1997, and his Masters and PhD at the same university in 1999 and 2004, respectively.
He is currently an Assistant Professor in the Department of Mechanical Engineering at the Ecole Nationale d'Ingénieurs de Monastir.
His research interests are kinematics, dynamics and control of parallel manipulators.
He has participated in several publications in the aforementioned area.

Lotfi ROMDHANE, received his undergraduate degree (Diplôme d'Ingénieur Principal) at the Ecole Nationale d'Ingénieurs de Tunis (Tunisia), in 1984, and his PhD in Mechanical engineering at the University of Florida, USA, in 1989. In the same year he accepted a post-doctoral fellowship, for two years, from the department of orthopaedics at the University of Florida.
In 1990, he joined the department of mechanical engineering at King Saud University as an assistant professor.
In 1992, he joined the department of mechanical engineering at the Ecole Nationale d'Ingénieurs de Monastir (Tunisia), where he has been a professor since 2001.
His research interests are kinematics, dynamics and control of robotic mechanical systems, biomechanics, and design of mechatronic systems. His work in the aforementioned areas has been the subject of several publications in international conferences and journals.