ADAMS/Simulink interface for Dynamic Modeling and Control of Closed Loop Mechanisms

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***Abstract* – In this work, computer aided methods for dynamics modeling and control of a closed loop mechanisms are presented. The software, ADAMS, was used to generate automatically the dynamic model. This model was then used under the Matlab/Simulink environment to control the mechanism. Simulation results are presented to illustrate the behavior of the whole mechatronic system under different control strategies. The crank slider mechanism was chosen to illustrate our approach. In particular, it is shown that simple PD controller is sufficient to yield acceptable dynamic behavior of a statically balanced mechanism even when gravity was taken into account.**

Mechatronic systems – Dynamic modeling – Control – Closed loop mechanisms.

I. INTRODUCTION

Closed loop mechanisms are present in virtually all mechanical systems. These mechanisms are usually synthesized to follow a prespecified trajectory. Synthesis of a mechanism controller and the simulation of the behavior of the whole mechatronic system is 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. 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 performance 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 make 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 mechatonic 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. Finally, section V contains some concluding remarks.

II. 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. 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.

A. *Description of the crank slider mechanism*

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

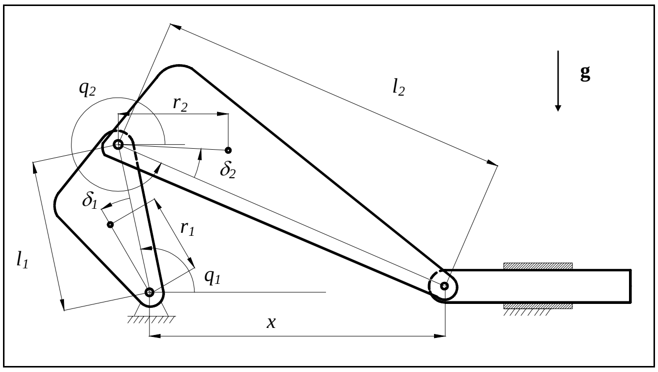

Fig. 1:schematic of the crank slider mechanism

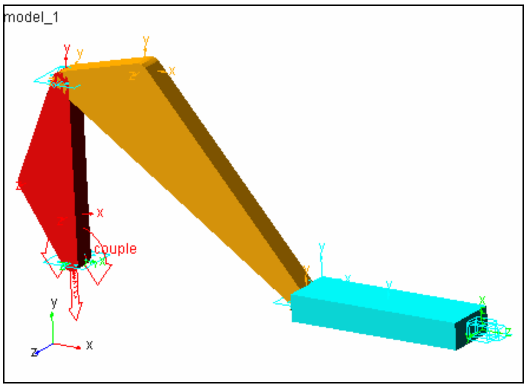

Fig. 2: ADAMS model of the crank slider mechanism

This parametrized model is convenient for testing different designs by simply modifying the values of the different parameters of the system. 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 I. 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.

TABLE I

Geometric parameters of the Crank slider mechanism

Parameter	**Crank**	**Connect. Rod**	**Slider**
Length (m)	$l_1 = 0.1$	$l_2 = 0.4$	$l_3 = 0.4$

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 II.

TABLE II

Inertia parameters of the Crank slider 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)	$I_{zz} = 0.04$	$I_{zz} = 0.1$	

B. *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 [1] 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:

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

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 following relation:

$$q_2 = \arcsin\left(\frac{l_1 \sin(q1)}{l_2} \right) \tag{2}$$

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

$$\begin{aligned} U &= m_1 r_1 g \sin(q_1 + \delta_1) + \\ &m_2 g \left(l_1 \sin(q_1) + r_2 \sin\left(\arcsin\left(\frac{l_1 \sin(q_1)}{l_2} \right) + \delta_2 \right) \right) \end{aligned} \tag{3}$$

The balancing condition can be expressed as:

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

where:

$$\begin{aligned} \frac{dU}{dq_1} &= m_1 r_1 g \cos(q_1 + \delta_1) \\ &+ m_2 g \left(l_1 \cos(q_1) + \frac{r_2 \cos\left(\arcsin\left(\frac{l_1 \sin(q_1)}{l_2} \right) + \delta_2 \right) l_1 \cos(q_1)}{l_2 \sqrt{1 - \left(\frac{l_1 \sin(q_1)}{l_2} \right)^2}} \right) \end{aligned}$$

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:

$$\delta_1 = \pi,\ r_1 = \frac{m_2 l_1}{m_1}$$
$$r_2 = 0 \qquad (5)$$

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 III presents the values of the parameters used in the simulation of the balanced crank slider mechanism.

TABLE III
Inertia parameters of the balance Crank slider mechanism

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

C. The dynamic model generated by ADAMS
Once the model is finalized, the dynamic model is generated to be exported under the MATLAB/Simulink environment.

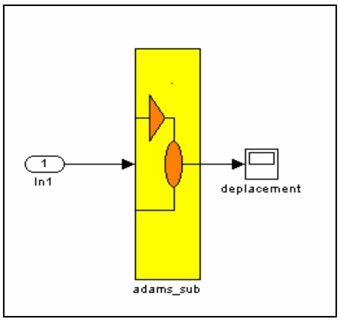

Fig. 3: the encapsulated dynamic model generated by ADAMS and shown under Simulink

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.

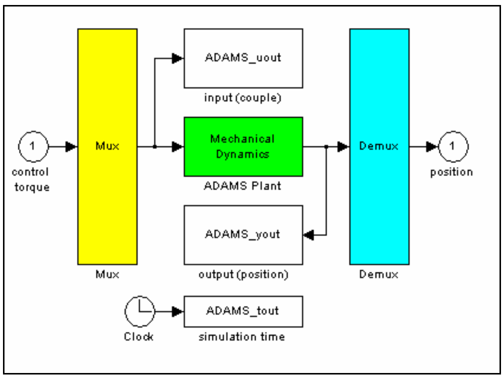

Fig. 4: architecture of the adams_sub block

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.

III. The Control Scheme

The Simulink library contains several control schemes, e.g., linear, non linear, fuzzy logic,…, in this paper we will be investigating the PD and the PID controller.
The PID control algorithm is given by:

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

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:

$$e(t) = x_d - x(t) \qquad (7)$$

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.

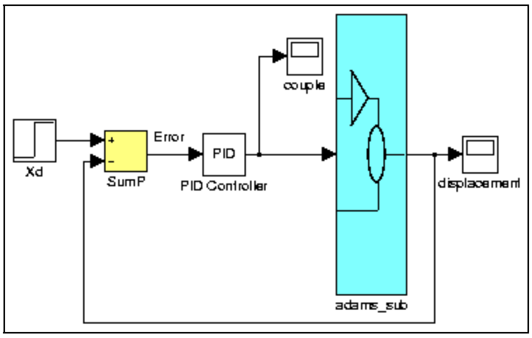

Fig. 5: the control scheme of the mechanism

IV. 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 I and Table II. The second one is a statically balanced crank slider mechanism whose parameters are given on Table I and Table III.
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 I and II, when the simple PD controller is used. It can be noticed that the static error is important (around 16 mm).

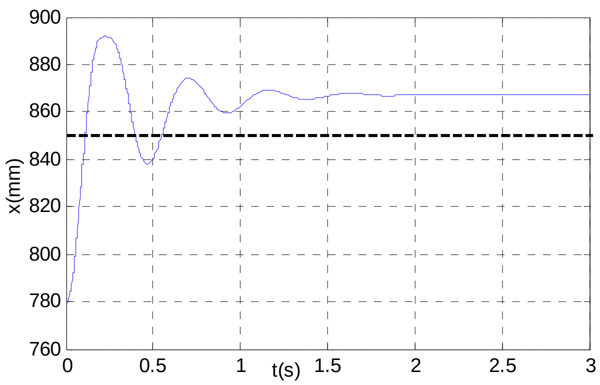

Fig. 6: Dynamic response of the crank-slider mechanism for (KP=90,KD=20)

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.

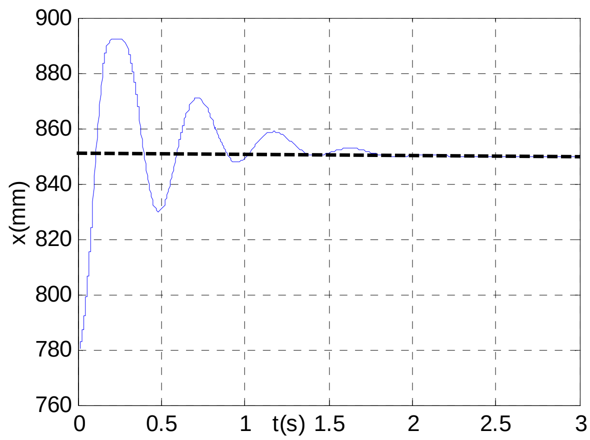

Fig. 7: Dynamic response of the crank-slider mechanism for (K_P=90,K_D=20, K_I=150)

However, if we balance this mechanism, the simple PD controller can yield satisfactory results without having the need to add the integrator. Fig. 8 shows such dynamic response.

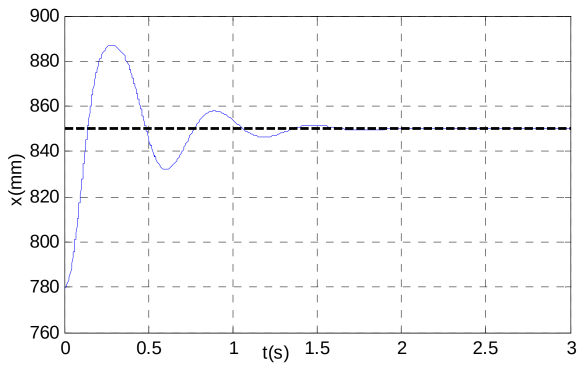

Fig. 8: Dynamic response of the balanced crank-slider mechanism for (K_P=90,K_D=20)

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 slight 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.

One of the main problems, that needs further investigation, is the dynamic behavior of the sevo-motor. As a future work, we propose incorporating the dynamic model of the servo-motor to take into account all the components of the mechatronic system.

V. Conclusion

In this work, we designed an interface between the software ADAMS and the Simulink environment. This interface allowed us to control and simulate the dynamic behavior of different designs. A crank slider mechanism was used to illustrate our approach. This mechanism was designed under ADAMS then its dynamic model was exported to Simulink for simulation. We showed that the behavior obtained by a simple PD controller is not 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 obtaine 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.

This interface between ADAMS and Simulink can be very useful to facilitate the approach known as “Design for control” (DFC). Complex mechanisms can be designed and tested very easily using this interface.

Acknowledgment

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