Kinematical analysis of crank slider mechanism using MSC Adams/View

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Abstract

The aim of this article is to develop a functional model of centric crank mechanism in ADAMS/View software, and its following complete kinematical analysis. We deal directly with modeling the crank mechanism in ADAMS/View software. The next stage is the simulation with a set of different parameters to obtain its kinematical analysis. Finally the data gathered in this process is compared and evaluated.

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Nomenclature

\begin{itemize}
  \item \( x \) displacement (mm)
  \item \( v \) velocity (mm/s)
  \item \( a \) acceleration (mm/s\(^2\))
  \item \( \phi \) angle (degree)
  \item \( \omega \) angular velocity (rad/sec)
  \item \( \alpha \) angular acceleration (rad/sec\(^2\))
  \item \( m \) mass (kg)
\end{itemize}

1. Introduction

The rapid evolution brings along a sharp increase in requirements for faster production with increasingly greater accuracy at the lowest possible cost. This is associated with the rising demands of customers, market dynamism but mainly by globalization. To satisfy these increasing demands some new, faster and more efficient methods for solving complex problems must be found.

The rapid development of computer technology allowed the development of different kinds of simulation software. By computer simulation can be solved a wide range of very difficult dynamic tasks. A computer model is compiled on the basis of mathematical analysis, defining the model having the properties of a real object. Since the created model has the same
characteristics as the real object, the computer simulation yields the same results as the actual simulation model. This computer simulation is carried out more cost effectively and can easily be repeated with modified model parameters. We can simulate the same model in different environments and under various external influences at the same cost which is not possible with a real model. Another advantage is that the computer simulation model is virtual in its nature and as such cannot be damaged and left useless for further simulations.

2. Kinematical analysis of crank slider mechanism

Crank slider mechanism is a simple machine used to transform the rectilinear translational (sliding) motion to rotary motion or vice versa. It is a simple mechanism but it has a very wide use [1-6].

In kinematical analysis the trigonometric method is used in coordinate system 0, x, y, [7-12].

From the Fig. 1:

\[ r \sin \varphi_{21} = l \sin \psi_{31} \]  \hspace{1cm} (1)

For point A stands:

\[ \sin \psi_{31} = \frac{r}{l} \sin \varphi_{21} = \frac{r}{l} \sin \omega_{2} t \]  \hspace{1cm} (2)

\[ \cos \psi_{31} = \sqrt{1 - \sin^2 \psi_{31}} = \sqrt{1 - \left( \frac{r}{l} \right)^2 \sin^2 \varphi_{21}} = \sqrt{1 - \left( \frac{r}{l} \right)^2 \sin^2 \omega_{2} t} \]

Parametric equations of trajectories of points A, B and C are:

\[ x_{A} = r \cos \varphi_{21} = r \cos \omega_{2} t \]  \hspace{1cm} \[ y_{A} = r \sin \varphi_{21} = r \sin \omega_{2} t \]  \hspace{1cm} (3)

\[ x_{B} = r \cos \varphi_{21} + l \cos \psi_{31} = r \cos \omega_{2} t + l \sqrt{1 - \left( \frac{r}{l} \right)^2 \sin^2 \omega_{2} t} \]  \hspace{1cm} (4)

\[ y_{B} = 0 \]

\[ x_{C} = r \cos \varphi_{21} + b \cos \psi_{31} = r \cos \omega_{2} t + b \sqrt{1 - \left( \frac{r}{l} \right)^2 \sin^2 \omega_{2} t} \]  \hspace{1cm} (5)

\[ y_{C} = d \sin \psi_{31} = \frac{rd}{l} \sin \omega_{2} t \]

The velocities of points A, B and C are:
\[ v_A = \sqrt{x_A^2 + y_A^2}, \quad v_B = \sqrt{x_B^2 + y_B^2}, \quad v_C = \sqrt{x_C^2 + y_C^2} \] (6)

And the accelerations of points A, B and C are:
\[ a_A = \sqrt{x_A'^2 + y_A'^2}, \quad a_B = \sqrt{x_B'^2 + y_B'^2}, \quad a_C = \sqrt{x_C'^2 + y_C'^2} \] (7)

We obtain angular trajectory, angular velocity and angular acceleration of the member 3:
\[ \varphi_{31} = \varphi_{31}(t), \] (8)
\[ \varphi_{31} = \pi - \psi_{31}, \] (9)
\[ \psi_{31} = \arcsin \left( \frac{r}{l} \sin \varphi_{21} \right) = \arcsin \left( \frac{r}{l} \sin \omega_{21} t \right), \] (10)
\[ \omega_{31} = \dot{\varphi}_{31} = -\dot{\psi}_{31} = -\frac{r \omega_{21} \sin \omega_{21} t}{l \sqrt{1 - \left( \frac{r}{l} \right)^2 \sin^2 \omega_{21}}}, \] (11)
\[ \alpha_{31} = \ddot{\varphi}_{31} = \ddot{\psi}_{31} = -\ddot{\psi}_{31} \] (12)

3. Compilation of a model in MSC Adams

In this chapter we describe the compilation of MSC Adams model of the crank slider mechanism [13]. This procedure consists of several steps, which are further described. The assembled model will be ready to simulate the movement of the mechanism to be analyzed (Fig. 2).

3.1. Members of the crank slider mechanism

Individual members of mechanisms are rendered in the plane by marking the essential key points, so called design points. In our case these are the points of joints.
The next step is the insertion of members of the crank slider mechanism. In the Main window we select Toolbox to insert a solid body called Link. A table of parameters for the solid body (Link) shows up to numerically enter the width and thickness of the given member. By a left click we place the member connecting the first and the second design point. Another member is also created by combining the second and the third design point. Inserting the individual members of the mechanism is shown in Fig. 3.

3.2. Determining the links

To enable the member movement the linkages must be defined for their respective joints. They define the relative positions of the two bound bodies. There are revolute joints between members of the crank mechanism. There are also links between the parts of the mechanism and the desktop. To create them, in the command window we select the Main Toolbox for determining rotational bonds. In the table with options for determining link properties we select the option for determining the joint by a defined point. Subsequently the bonds are defined by clicking the left mouse button on the first design point, which ensures that the first member will rotate about the first design point where the drive of the mechanism will be placed later. The second link is defined in the second design point, which produces the rotational joint between members. Last revolute joint is placed in the third design point (Fig. 4). It will place a revolute join between the second member and the member performing translational motion.

3.3. Defining the motion

For performing the kinematic analysis of the mechanism we have to define its movement. This is achieved by defining joint motion, thus ensuring the rotation of the driven member and the consequent movement of the whole mechanism. To define it we select the Toolbox command in the Main window. Then we select the rotational drive and in the table with the kinematic parameters we set rotation speed of 30 rad/s. By the Left click we place it to the first design point which represent the first revolute joint (Fig. 5).
3.4. Verification of the model functionality

After these steps the model definition is completed and the model is ready for simulation and kinematic analysis. Another task before the actual simulation is the possibility of modification of some parameters of the mechanism. Therefore, the next steps describe the creation of a table of parameters. We show how it is defined, what its purpose is and how it simplifies the change of parameters of the crank slider mechanism.

4. Modifying model parameters in MSC Adams/View

This chapter describes the creation of a table of parameters in which we can easily change the dimensions of the members of the mechanism and the initial position of the driven member. This table will be integrated into top bar menu to use it at any stage of the model analysis (Fig. 6).

4.1. Variable parameters of the model

To enable the modification of dimensions of the crank mechanism by using the table of parameters we have to first create and define the variable parameters and define their properties. In the top bar with the command Build we select the option Design Variable which starts the creation of a new parameter. A window opens where the first line defines the name of the parameter. In our case, we call it R1, which will define the length of the first member. The Standard Value setting defines the default value of the parameter. The next line Value Range defines the change of the parameter – we chose the absolute method from a minimum to a maximum i.e. Absolute Min and Max Values. The numeric values are entered in the next fields and confirmed by the Apply button. With the same procedure we create the next parameter R2, which represents the length of the second member. The last parameter will define the initial position of the driven member.

4.2. Equations of design points

These equations are used to define the positions of the design points associated with members to correspond to the changed parameters of the model. Equations in which the parameters define the position of design points are compiled. To enter the equations we open the Equation Editor Table which is selected in the top bar in the Tools window Fig. 7.
4.3. Table of model parameters

Table of model parameters serves for the simple and fast change of parameters of the crank mechanism. It contains the code which by entering the corresponding parameter values changes the appropriate model parameters. It is created by selecting Tools, the Dialog Box and then Create. A window appears with commands to create a parameter table that is shown in Fig. 8.

4.4. The finished model with a parameter table

Fig. 9 shows the model of the crank slider mechanism and the table of parameters to change the corresponding model properties that enter the kinematic analysis.

5. Graphical representation of the simulation results

The most important kinematic variable of the crank slider mechanism are those of the member performing translational motion. Will investigate its displacement, velocity and acceleration during the simulation time. To obtain a better understanding of effects of various model parameters to kinematic variables we will compare two different crank mechanisms in each graph. These vary either by structure or by parameters of the drive.
5.1. Effect of the geometry of the model

The first pair to compare is a couple of crank slider mechanisms, where the second crank mechanism has the length of the driven member $R_1$ twice as long as the first one (Fig. 11). Using the table of parameters the desired configuration is achieved efficiently. In this case, the dimensions chosen are shown in Fig. 10.

![Fig. 10](image)

Fig. 10. Defining the joints between members of the crank slider mechanism for (a) rotational joints and (b) translational joint.

![Fig. 11](image)

Fig. 11. Setting simulation model parameters.

The graph in Fig. 12 a) shows the displacement of the sliding member performing translational motion during one revolution of the driven crank. Red solid line in the graph represents the values of the model with the short driven crank, the dashed blue line represents the values for the model with the longer crank. It is clear from the graph, that the change in the length of the driven member causes a change of the length of translational displacement of the sliding member. It is also evident, that the change of the driven crank of about 150 mm extended a total distance of translational member of about 300 mm. It is a linear relationship and it means that the quantitative change of the crank will cause twice as large change in the distance travelled by the slider.

![Fig. 12](image)

Fig. 12. Illustration of crank slider mechanism (a) position of sliding member of the crank slider mechanism and (b) velocity of sliding member of the crank slider mechanism.

The Fig. 12 b) shows the dependence of velocity of the slider during one revolution of the crank. It can be seen that changing the crank length from 150 mm to 300 mm which is doubling its size, the velocity of the slider increases two and a half times. This dependence is not linear but exponential. A small extension of the crank will result a higher increase of the speed of the slider. The Fig. 13 shows the dependence of the acceleration of the slider during one revolution of the crank. The obtained graphs show that doubling the size of the crank causes the rise of the acceleration of the slider in some places almost sixfold. With this rapid increase of acceleration increase the inertial forces according to the known relationship. It follows that the masses of crank slider mechanisms should be as low as possible to avoid large inertial forces.
5.2. Effect of the speed of rotation of the crank

The second pair of crank slider mechanisms compared in the analysis will have the same structural parameters and will only vary in the speed of rotation of the crank. The speed of rotation of the crank of the second mechanism will be two times greater than the speed of rotation of the first one. The first crank rotates at speed 5/min and the second at 10/min.

The graph in Fig. 14 a) shows the displacement of the sliding member performing translational motion during one or two revolutions of the driven crank of the first and second model respectively. We can say from the graph that the amplitude of the path traveled by the slider does not change. The difference is that it takes the half of time to complete one period of motion for the slider of the second model. The Fig. 14 b) shows the velocity of the slider during one revolution of the crank for the first model and during two revolutions for the second model. The maximum speed of the second slider is twice as high as that of the first one. The relationship between the change of speed of the crank and the resulting change of speed of the slider is linear.

In Fig. 15 there is the graph of acceleration of the slider of the first and second model respectively. We can tell from the graph that the acceleration of the second slider is almost four times higher than the acceleration of the first one. The relation between the rise of speed of the crank and resulting increase of speed of slider is exponential.

5.3. Comparison of the centric and eccentric crank slider mechanism

The last pair to compare is the centric and eccentric crank slider mechanism. We will investigate the changes resulting from moving the center of rotation of the crank from the axe of a certain amount of eccentricity. Compared crank mechanisms have the same dimensions. The second crank’s center of rotation is shifted by the value of 100 mm from the center position.
Graph in Fig. 16 a) shows the position of slider of the centric (red) and eccentric (dashed blue) crank slider mechanism during one revolution of the crank. Effect of the eccentricity is only in shifting the end positions of the slider. The value of this shift is proportional to the size of the eccentricity of the eccentric crank slider mechanism. The graph of the velocity of the slider shown in Fig. 16 b) shows that the eccentricity causes the change of its shape. The main difference between concentric and eccentric crank slider mechanism is that during one cycle the slider of the centric mechanism reaches the maximum two times. The eccentric crank slider mechanism has two maxima that are not the same.

Fig. 17 shows the acceleration of sliding member of the centric (red) and eccentric (dashed blue) crank slider mechanism. In this case the eccentricity causes the change of the graph shape.

5.4. Evaluation of data obtained from the simulation

In previous analysis we found that even small change in one of the characteristic parameters can significantly change the characteristics of behavior of the crank slider mechanism. These findings can help in the design stage of the crank slider mechanism showing the maxima of the mechanism load caused by inertial forces. The characteristics of position, velocity and acceleration are also important parameters in designing a crank slider mechanism.

Conclusion

MSC ADAMS / View contains a specialized interface for creating virtual objects consisting of rigid and deformable parts linked to each other with different kinematic links. This allows to create static, kinematic and dynamic analysis of virtual prototypes by computer simulation. The advantage is the compatibility with CAD file formats and the ability to import geometry directly into the ADAMS interface.

We investigated a functional model of a crank slider mechanism. Creating the table of parameters enabled us to change the dimensional parameters of the model. Subsequent simulation of several configurations of the crank slider mechanism and the comparison of the outputs showed the impact of the change of the respective parameter. The whole modeling, parameterization and subsequent simulation was carried out by simulation program MSC ADAMS/View.

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