SIMULATION AND OPTIMIZATION OF SPACE DOCKING MECHANISM

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Abstract

Dynamic simulation is essential to the study on the space docking mechanism, docking dynamics and the development of new design concepts for advanced docking system. This paper presents the research on the simulation and optimization of the most widely used space docking system-androgynous peripheral assembly system (APAS). Two analysis methods were used to evaluate damping system in APAS-Automatic Dynamic Analysis of Mechanical System (ADAMS) and simplified dynamics model. This article describes the damping mechanism in brief, outlines each of the methods and compares their analysis results. Certain kinematic and dynamic simulation results are presented which are valuable to investigate insight into the system’s behaviour, study contingency simulation scenarios, and improve the fidelity of dynamic simulation of the docking event.

Introduction

Space docking technique, one of the keys to rendezvous and docking, is an essential element for in-orbit operation, such as platform servicing and construction of large space station. Research and development of this technique has been imperative for the advancement of aerospace technology and manned space technology. The system used to implement two spacecrafts’ docking mission is called as “Docking Mechanism”. A Russian-designed and -built mechanism (APAS) was used in 1975 for Apollo-Soyuz Test Project, and its hybrid version in 1995 for the U.S. Space Shuttle Atlantis docking to Russian space station Mir, and it continued to play an important role in establishing International Space Station (ISS). During rendezvous, damping mechanism in APAS arrests the relative motion of the two vehicles and prevents them from colliding. Therefore, damping mechanism is the most critical section of berthing mechanism in both function and structure. The research and development of the performance capabilities of damping system is of great significance. However, it is difficult and costly to conduct a ground simulation and test for the study on the dynamic behavior of the hardware. Hence, a detailed computer simulation is a necessity.

Docking dynamics has been the subject of several investigations, which presented analysis of docking. Several authors studied the contact dynamics of two spacecrafts. As a result of these investigations, several computer routines have been developed for the dynamic analysis of docking process. It should be noted that the most previous studies on the docking dynamic simulation were based on the simplified description of the damping mechanism, and the degree of accuracy of such calculations is reduced. So far, however, no quantitative studies have been made on the performance and
characteristics of the damping system. This motivates the work reported in this paper.

This paper is organized as follows: followed the introduction, the paper briefly explains the damping mechanism and discusses its operation principle. Then, two analysis models are described, respectively: simplified dynamic model based on the virtual work principle, and virtual prototype built on the SGI workstation by using ADAMS. The paper explains how simulation results from the two models correlates each other, details the analysis performed, and offers an overview of the insights gained by virtual test on the ADAMS model. The last section is Summary.

Mechanism Description

During docking process, the orbiter is maneuvered to bring the interfaces of the docking vehicle in contact with the target vehicle with six-degree of freedom (dof) while eliminating the relative misalignments between the interfaces. Therefore, the capture ring on the active docking vehicle should be displaced and rotated freely. In APAS unit, a three-petal andrognous capture ring is mounted on six interconnected ball screw shock absorbers. Three gear assemblies between every two of the six absorbers are located on the capture ring to implement the displacement difference between two adjacent ball screw/nut assemblies, and another two gear differential assemblies are located between each two adjacent ball screw/nut assemblies to achieve the displacement difference of them. Each ball screw/nut assembly is connected on one end by a universal joint to the supporting truss structure and on the other end by a spherical joint to the capture ring. Those difference assemblies make the capture ring have six-dof: three translations and three rotations.

Simplified Dynamic Model

Referring to the global coordinate system \(O-x_1y_1z_1\) on the supporting truss structure and the local coordinate system \(O-x_3y_3z_3\) on the capture ring, as shown in Fig. 1, the following equation holds for a branch of the mechanism

\[ \mathbf{L}_i = \mathbf{R}_i \mathbf{m}_i + \mathbf{P}_i - \mathbf{b}_i \]  

(1)

Where \( \mathbf{P} \) is the 3 by 1 vector and \( \mathbf{R} \) is the 3 by 3 rotation matrix, which define the position and orientation of the capture ring relative to the base, \( \mathbf{m}_i \) is the 3 by 1 vector representing the position of the \( i \)th joint on the capture ring in the local coordinates, \( \mathbf{b}_i \) is the 3 by 1 vector representing the position of the \( i \)th joint attached to the base in the global coordinates, \( \mathbf{L}_i \) is 3 by 1 vector representing the \( i \)th link.

All the springs, electromagnetic dampers and gears’ kinematic models are obtained as a function of the links’ displacement \( l_i \):

\[ c_{li} = f((l_1, \ldots, l_6), I_{cl}) \]

\[ \omega_{di} = f((l_1, \ldots, l_6), I_{di}) \]

(2)

where, \( I_{cl}, I_{di} \) is the transmission ratio to the corresponding springs and dampers, respectively.

The damping mechanism has six-dof, and to model this in the traditional Lagrange’s formulation would be nearly impossible. Numerous bodies with inertial
characteristics varying by several orders of magnitude and undergoing constrained motion, combined with stiff components and damping elements, would make the numerical solution extremely time-consuming and would make post-processing overly tedious. The primary goal for the damping system simulation is to analysis the damping system rapidly, and to obtain the attenuation characteristics, which will be used to demonstrate the correlation of the numerical model and ADAMS model. With this in mind, the configuration and symmetry of the mechanism allows for a significant reduction in the analytic description of the mechanism. The capture ring is much heavier than all the other bodies in the mechanism, hence the bodies mass are negligible. In order to obtain the equivalent stiffness and damping coefficients, static test should be made on the system. In that case, virtual work principle could be adopted to model the damping system.

The following equation holds for the infinitesimal displacements of links \( \delta l \) and the infinitesimal motion of the capture ring \( \Delta P \):

\[
\delta l = J \Delta P \quad (3)
\]

In this paper, we refer to \( J \) defined by eq. (3) as the Jacobian of the parallel link structure. Consequently, we obtain the infinitesimal displacements of the attenuation components as function of \( \delta l \):

\[
\delta cl_{i} = \frac{\partial \delta l_{i}}{\partial l_{i}} \delta l_{i} \quad (4)
\]

\[
\delta d_{i,j} = \frac{\partial \delta d_{i,j}}{\partial l_{i}} \delta l_{i} \quad (4)
\]

Where, \( \delta cl \) , \( \delta d \) are, respectively, the infinitesimal displacements of springs and electromagnetic dampers.

Using the principle of virtual work, we have the following relation:

\[
\sum_{i=1}^{n} f_{i} \delta l_{i} = \sum_{i} M_{c,i} \delta cl_{i} + \sum_{j} M_{d,j} \delta d_{j} \quad (5)
\]

Where, \( M_{c,i} \) , \( M_{d,j} \) are the moments of springs and electromagnetic dampers, respectively.

Substituting eq. (4) into eq. (5), we have

\[
f = M \cdot [A]^{p \times 6} \quad (6)
\]

Where, \( [A]^{p \times 6} \) is the transformation matrix between displacements of attenuation components and links, and every elements of the matrix is the function of the pitch of the screw and the corresponding transmission ratio.

Using eq. (3) and virtual work principle to the parallel structure, the relationship between the links forces and the forces/moments of the capture ring can be expressed as:

\[
F = J^{T} f \quad (7)
\]

\[\text{ADAMS Model}\]

Although physical prototype has been successfully operated for demonstration purposes, and a simplified algebraic model has been formulated, several issues remain with the past designs, design methodologies, and mathemathical models. These issues need to be addressed in order to design and implement a powerful APAS.

The goal of the virtual prototyping program is to provide engineers a tool to create a complete, accurate model of APAS to be used early in the design process. The model provides a baseline that an engineer can change in order to verify the APAS design's functionality and to optimize its performance.

Fig.2 shows the steps in modelling and simulating. Models of individual dynamic components are being developed and
Then component models are integrated into an overall model of APAS. The APAS under development consists of 86 moving parts, 6 cylindrical joints, 91 revolute joints, 6 screw joints, 49 couplers and 6 gears. Sensors are included to detect the displacement of screws and springs. In order to define the moments applied by several springs and electromagnetic dampers, numerical procedures must be adopted. In this paper, a general formulation of the attenuation forces using spline functions is implemented. And two important issues must be considered. The first is the setting of the initial value of springs, and the second is the smoothness of the splines to avoid numerical problems that result from irregularities that do not influence the gross motion of the mechanism.

The virtual prototype showed an excellent correlation with simplified models results. Fig. 3 shows the damping force comparison between VP and math model prediction. The force is representative of the load imposed on the capture ring by the attenuation components when the capture ring has misalignment. Since forces are represented by a curve in terms of movement, two indices are introduced to evaluate the fidelity of the two models. They are the mean value and standard deviation. The mean value indicates the quasi-static force, as it is the averaged force difference over the entire motion period. The standard deviation represents the dynamic force difference fluctuation around the mean value.

To fig. 3, the mean value is -8.2, sd is 4.8. Therefore, Fig. 3 provide a good approximation for VP and the math model. Meanwhile, we can discover that the results of VP show the dynamic characteristics of the system more deliberately. In fact, the attenuation characteristics of the system changes throughout the workspace of the capture ring. They are different with respect to position and orientation of capture ring.

**Description of Analysis Performed**

The virtual prototype was used for the detailed investigation of the mechanism’s kinematic and dynamic performance. The obvious benefit of the VP was that an infinite number of simulations could be performed and viewed at a fraction of the cost of time and modification of the math models.

Fig. 4 shows the maximum displacement of the capture ring along axis Y. The curve is characterized with three sections along the mechanism’s longitudinal axis. The middle section is measured because the movement of the mechanism stops by the sensors monitoring the springs, the other two are because of the sensors monitoring the lengths of screws. The result shows that capture and attenuation should occur within the middle height so that it is not so easy to damage the mechanism.

Fig. 5 shows the kinematic coupling of the capture ring for a case that the impact force is along axis Y. The coupling displacement along axis X is much smaller than that along axis Z, and more stronger the impact force is, much bigger the main displacement is.

Fig. 6 shows the simulation result of mechanism dynamics. They are the equivalent attenuation forces along axis X and axis Y, respectively.

The mechanism settings for the presented results were identical for all simulations. Various trades were performed as gear ratios were adjusted; component damping and stiffness characteristics were parameterised. By analysing the mechanism’s behaviour when it was set for different system parameters, the results
Fig. 2 Flow Chart of ADAMS Modeling and Simulating

Fig. 3 VP and Math Model Results-Equivalent Force

Fig. 4 Maximum displacement along axis Y

Fig. 5 Coupling motion of the capture ring
showed how the hardware’s performance would respond to the internal factors.

Various characteristics of the mechanism’s response were investigated and explained with the VP. The engineers focused on obtaining a combination of system parameters that made the mechanism have the cost-and-effective capabilities. These findings have been essential in ensuring a satisfied structure and functionality of the damping mechanism.

**Summary**

In this paper, two methods with their methodology for analyzing the kinematics and dynamics of damping system in APAS have been presented. The advantage of the method based on ADAMS is that the modeling is simple by knowing the operation principle. The virtual prototype allows the engineers to have a look at the simulation process.

Two kind of analysis tool are developed to evaluate the performance capability of the damping mechanism in APAS: simplified mathematical model and virtual prototype. This paper outlines the establishment of the two models, describes the analysis performed, and presents the simulation results. The virtual prototype is proved to be valuable aids in investigating and demonstrating the complicated kinematics and dynamics of the docking mechanism.

**References**

5. ADAMS User Manual.