Mechanical Analysis of A Ride Entertainment System

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Abstract

Many carnivals, parks and museums carry ride entertainment systems that provide simulated action-type entertainment such as race car driving, airplane acrobatics, etc. These entertainment systems provide pitch, roll, yaw and heave motions. Unlike air-crew-member training simulators, there are no standard design guide lines for the ride entertainment system. More state regulatory agencies are demanding safety analysis of these systems. Analysis for such a system has been carried out for developing design guide lines for entertainment simulators.

The present paper describes the analysis of an entertainment system capable of pitch, roll and heave. The motion is provided by three hydraulic actuators. The actuators connect the base frame to the motion frame with gimbals and pins simulating universal joints. The unit is capable of carrying fifteen passengers with motion ranges of $\pm 20^0$ in pitch and roll, and fourteen inches of heave. The velocity ranges experienced by the passengers span zero to one mile per hour in heave and 0.5 rad/s in pitch and roll Passenger accelerations of up to 0.8 g are attainable, with maximum angular acceleration restricted to 3 rad/s^2 in pitch and 3.5 rad/s^2 in roll. The system rests on docking pads while loading and unloading passengers. These pads also serve as limits for motion in roll and pitch.

The system is complex and needs accurate modeling to be able to precisely predict the performance of the system. The dynamic analysis requires mass properties that are difficult to physically measure. The mass properties have been determined by modeling the parts in I-DEAS™-Master Modeler module. The geometric and kinematic relationships among the parts were established in I-DEAS™ - Master Assembly module by creating subassemblies and the final assembly. A mechanism containing the mass property information representing the simulator was modeled in I-DEAS™- Mechanism Design module. This mechanism was then transferred into ADAMS™ for dynamic analyses. Roll, Forward and Aft pitch motions were analyzed.

ANSYS™ has been used to determine the stresses caused by the loadings obtained from the dynamic analysis with full passenger loads. Factors of safety for parts perceived as critical based on past experience with the simulators have been computed.

I. Modeling using I-DEAS™

The modeling consisted of creating the solid models of 58 parts comprising motion system. Using these parts, six subassemblies , namely, base frame, motion frame, front hydraulic cylinder , rear hydraulic cylinder, rear A-frame , and cabin assemblies, were created in the Master Assembly module of the I-DEAS™. These subassemblies were assembled to make the motion system. Using the Master Mechanism module, kinematic and dynamic constraints were added at the joint locations in the final assembly so as to enable the motion system to respond to kinematic and dynamic inputs. These constraints subsequently allowed the joints loads be evaluated. The mechanism so created was transferred to ADAMS for dynamic solution.

II. Force Analysis:

The motion frame and the passenger cabin form a dynamic system and undergo a range of angular and linear motion within the range of operation. The system is analyzed for a Run Away condition, which forms the most severe loading condition. The run away motion is stopped by three docking stands.

Forward Pitch: The motion starts from the midpoint of the piston travel. The front hydraulic unit retracts to a velocity of 0.39 m/s (maximum allowed velocity) while the rear hydraulic assembly extends to the rate of 0.37 m/s. The motion is stopped simultaneously by the A-frame hitting the front docking stand and by the cushion provided at the rear pistons when they extend to the maximum travel. It was experimentally observed that the front pistons do not reach the bottom cushion during this motion. The ADAMS™ Model was built to conform to this condition.

Aft Pitch: The motion starts from the midpoint of the piston travel. The front hydraulic unit extends to a velocity of 0.39 m/s (maximum allowed velocity) while the rear hydraulic assembly retracts to the rate of 0.37 m/s. The motion is stopped simultaneously by the A-frame hitting the rear docking stands and by the cushion provided at the front piston when it extends to the maximum travel. It was experimentally observed that the rear pistons do not reach the bottom cushion during this motion. The ADAMS™ Model was built to conform to this condition.

Roll: The motion starts from the midpoint of the piston travel. The front hydraulic unit stays at this position while one of the rear hydraulic assemblies extends and the other rear hydraulic assembly retracts to a the maximum allowed velocity of 0.37 m/s as above. The motion is stopped simultaneously by the A-fame hitting one of the rear docking stands and the cushion provided at the end of travel of the other rear hydraulic unit.

Forces: The motion system is actuated by the expanding/retracting hydraulic units. A constant force resulting in the specified motion was applied at the pistons/cylinder (sliding) joint. The impact between the motion frame and the docking stands was modeled as constant force assuming that the motion to stop when the shock pads are compressed to half their thickness. The cushion forces were modeled the same way now assuming that the motion to stop when the cushion between the piston and the cylinder end is 1.125 in. Some of the revolute joints were replaces as bushings to include dynamic effects on the joints.

Ground Reaction: The base frame was considered as bolted to the ground by four bolts. The forces needed to hold the motion system to ground were obtained from the ADAMS results as shown.

III. Stress Analysis

ANSYS™ was used to determine the maximum stress in each part. ANSYS™ element solid-92 was used. The results indicated varying degrees of SEPC (Strain Energy percent error). which represents the error relative to the mesh discretization. In the solution the mesh size was adjusted to minimize the SEPC. Multiple runs with different mesh patterns and sizes were needed to reduce the SEPC error to an acceptable level.

IV. Acknowledgment: The work and the report is supported by a grant from SPIR (Strategic Partnership for Industrial Resurgence) from the State of New York and Doron Precision Systems.