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MECHANICAL DESIGN OF NASA AMES RESEARCH CENTER VERTICAL MOTION SIMULATOR

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ABSTRACT

NASA has designed and is constructing at the Ames Research Center a new flight simulator with large vertical travel. Several aspects of the mechanical design of this Vertical Motion Simulator (VMS) are discussed, including the multiple rack and pinion vertical drive, a pneumatic equilibration system, and the friction-damped rigid link catenaries used as cable supports.

INTRODUCTION

Existing flight simulators are inadequate for a number of critical tasks. Among these are simulation of aircraft flare and touchdown, particularly for V/STOL and carrier aircraft landings, and control with degraded longitudinal stability. Extended vertical travel is necessary to accomplish this class of simulations with sufficient fidelity. To meet this need, NASA has designed the Vertical Motion Simulator (VMS), which is now under construction at the Ames Research Center.

GENERAL DESCRIPTION

The VMS motion generator provides six degrees of freedom for a fully outfitted cab. The motion generator will be installed in a tower that is approximately 22m (73 feet) long by 11m (36 feet) wide by 34m (110 feet) high. Vertical motion is the primary degree of freedom. A vertical platform is the basic structure driven vertically and all other degrees of freedom are assembled on top of it. The vertical platform, fabricated primarily from aluminum plate and angles, spans the tower and is supported by two vertical drive columns spaced to minimize deflections. Eight DC servo-motors drive the platform vertically for a total usable travel of 18m (60 feet) through gear reducers, pinions, and racks which are attached to the columns (see Figure 1 and Table 1).

Principal design calculations used the English system of units.

Table 1. Vertical Motion Simulator Parameters

Item	Value	
Vertical Travel, m (feet)	18	(60)
Lateral Travel, m (feet)	12	(40)
Maximum Vertical Acceleration, m/sec^2 (ft/sec^2)	10	(32)
Maximum Lateral Acceleration, m/sec^2 (ft/sec^2)	7	(24)
Effective Mass for Vertical Acceleration, kg (lb_m)	$5x10^4$	($1.1x10^5$)
Mass for Lateral Acceleration, kg (lb_m)	$1.2x10^4$	($2.6x10^4$)
Maximum Vertical Velocity, m/sec (ft/sec)	6	(20)
Maximum Lateral Velocity, m/sec (ft/sec)	3	(10)
Equilibrator Pressure, MPa (lb/in^2)	2.41	(350)

Four torque tubes mounted in bearings at the tower floor extend out to the drive columns. The vertical drive racks engage pinions on the ends of the torque tubes forcing them to rotate as the platform moves vertically. The torque tubes synchronize the columns driving the platform and react roll moments induced when the cab is off the center position.

Two guide rails attached to the east and north walls of the tower provide continuous support points for the horizontal loads on the vertical platform. Wheel assemblies on the vertical platform transfer these loads to the rails.

Lateral motion capability of 12m (40 feet) is provided by a lateral platform which is driven across the vertical platform. Four DC servo-motors on the lateral platform drive through gear reducers and pinions to a fixed rack on the vertical platform.

A commercially available six degree of freedom motion generator will be mounted on the lateral platform to provide longitudinal and rotational motions. The hydraulic power supply for this unit will be installed in a separate room adjacent to the tower. Hydraulic and electrical power and signals are transmitted through lines mounted in two catenaries linking the lateral platform to the tower walls (Figure 1).

An equilibration system, used to uncouple gravity forces from the vertical drive, uses air pressure inside the drive columns to provide a constant upward force equal to the effective weight of the vertically moving components of the simulator.

VERTICAL DRIVE

Each tubular vertical drive column is 35m (81 feet) long and is loaded by internal air pressure as well as by mechanical loads from the drive and torque tube pinions. Two racks are attached to each column. The pinions which engage them are mounted in diametrically opposed pairs. Pinion separating forces are then balanced, eliminating the need for heavily loaded guide rollers. A small (eight degree) pressure angle rack reduces squeeze on the column from pinion separating forces.

The drive columns are critical components of the simulator from a safety standpoint. Failure of a column would be catastrophic. For this reason extra care in analysis of the stresses was needed. Flanged connections were designed using the ASME Boiler and Pressure Vessel Code (Reference 1), and stress levels were further checked by finite element analysis using the NASTRAN program. The tubular part of the columns was sized using hand calculations, but the complicated geometry of the rack attachment and multiple pinion load points resulted in a decision to refine that analysis through the finite element approach.

All pinions for each column are mounted in a carrier which encompasses the column and reacts the separating forces internally (Figure 2). The column is positioned laterally relative to the pinion carrier by polyurethane coated guide wheels mounted in the pinion carrier. Pinion carriers are gimbal mounted on the floor to accommodate small angular misalignments and crookedness of the columns. Axial alignment of columns with their pinion carriers is also achieved by the guide wheels. Pinions are flexibly spline-mounted on their shafts to assure tooth contact across the full 10cm (4-inch) rack width. Drive motors, reduction gears, and torque tubes are mounted rigidly to the floor and connected to the pinion shafts by gear couplings.

One of the design criteria for the VMS is to provide a mechanically stiff system in order to keep its fundamental frequency well above the desired servo-controlled operating frequency of about 2 Hz. Individual components were designed to have a frequency of about 15 Hz, recognizing that when they are combined in a single structure their combined compliances would result in a lower resonant frequency. A NASTRAN analysis was done to find the frequencies and mode shapes for the complete VMS structure, giving a first natural frequency of 7.5 Hz. Figure 3 shows the undeformed and deformed plots of the vertical platform for the 7.5 Hz fundamental frequency.

EQUILIBRATION SYSTEM

The purpose of the equilibration system is to support the dead weight of the simulator so that the drive system sees nearly identical inertial loads whether driving up or down. The effective weight of the vertically moving parts of the simulator is about 4.9×10^5 newtons (110,000 lb). Equilibrating forces for the VMS arise from pressurized nitrogen gas contained inside the drive columns. Each drive column fits over a stationary inner tubular column which is connected by piping to a gas storage volume of 28 m^3 (1000 ft^3). A sliding seal at the bottom of each drive column allows vertical motion with little gas leakage. Changes in volume of the gas container resulting from vertical motion are relatively small ($\pm$ 5% from mid-travel position) resulting in a nearly uniform equilibrating force.

FRICTION DAMPED CATENARIES

Description

The total vertical and lateral excursions of the VMS motion system are

22m (72 feet) and 14m (46 feet) respectively. A number of schemes to provide electrical power and control cables, instrumentation leads, and hydraulic lines to the lateral carriage from various points in the tower were investigated. The chain-like apparatus shown in Figures 1, 4 and 5 was selected as the best candidate for the job. The two flexible "conduits" are suspended between the support structure on the lateral carriage and two pivot points which are located 16m (52 feet) above the floor on the north and south tower walls. The two catenaries are constructed much the same as a roller chain. The links are approximately 0.9m (3 feet) long from pin to pin and wide enough to provide 0.6m (2 feet) of clear space for the attachment of cables and hoses. In order to prevent the catenaries from whipping and oscillating as the motion system goes through its various gyrations, spring loaded brake discs will be installed at the hinge points of each link. They are arranged as shown in the exploded view and section in Figure 4.

Urethane bumper pads and stops will be installed at each of the hinge points on both sides of the links to prevent the catenaries from reversing curvature during large, downward, vertical excursions when the acceleration is greater than one "g". (This is outside the scope of the present motion generator but within the scope of an upgraded version.)

The catenaries consist of 24 links each and have a total length, from the pin joint on the tower wall to the pin joint at the lateral carriage, of 22m (73 feet).

DYNAMIC ANALYSIS

Concern about the dynamic behavior of the catenaries during various simulator motions resulted in the concept of the coulomb friction damped catenaries described above. Extraneous forces from whipping and flailing cables and hoses would produce severe negative effects on servo-system performance. The purpose of the dynamic analysis was to give assurance of adequate performance by the catenary system.

A numerical analysis was performed using the idealized model described in Figure 5. Masses are considered to be lumped at the joints of the 24-link catenary. The links are all of equal length and are considered rigid.

K_i represents the torsional spring coefficient in a joint. The spring stiffness is derived from the cables and from the "anti-reverse curvature" urethane bumpers. Figure 6 shows the variation of the spring stiffness as a function of $\Delta\theta_i = \theta_i - \theta_{i-1}$.

$$K_i = \frac{K_P + K_N}{2} + \frac{K_P - K_N}{2} \operatorname{sign}(\Delta\theta_i)$$

Springs mounted on the building wall to cushion impacts of the joints with the wall were also included in the model, but details of their incorporation are omitted for brevity.

Frictional damping in the joints is denoted by C_i. Because of the discontinuous nature of friction force with reversing velocity and numerical analysis considerations, the following function was used to represent friction torque versus $\Delta\theta_i = \theta_i - \theta_{i-1}$:

$$F_T = T_0 \left(\frac{2}{\pi}\right) \arctan [S(\Delta\theta_i)]$$

X (t) and Y (t) are the prescribed motions of the last joint. In reality, X and Y can be any functions of time that the simulator servo-system can generate. In this case $X = A \cos \omega_1 t$, $Y = B \cos \omega_2 t$ were chosen because of the mathematical simplicity of the functions. With the simultaneous application of X and Y motions and appropriate selection of A, B and $\omega_{1,2}$ the extremes of displacement, velocity, and acceleration can be obtained.

Lagrange's equation was used to obtain the differential equations of motion for each joint, resulting in a system of 24 second order, non-linear differential equations. These were converted to 48 first order equations and solved numerically.

Although printed output was obtained, the most interesting and useful results were the 16 mm movies of the computed joint trajectories. They were generated by plotting joint positions at 1/96 second intervals resulting in movies with the motion slowed by a factor of 4.

After analyzing cases with and without friction damping and anti-reverse curvature springs, it appears that incorporation of these concepts will result in satisfactory operation of an otherwise ill-behaved cable support system, using only about 15% of the 110 newton-m (1000 in-lb) friction damping torque available at each joint.

REFERENCES

1. ASME Boiler and Pressure Vessel Code, Section VIII, Division I, Appendix II, Part B, 1971

Figure 1. Scale model of Vertical Motion Simulator

Figure 2. Vertical drive pinion carrier of Vertical Motion Simulator model

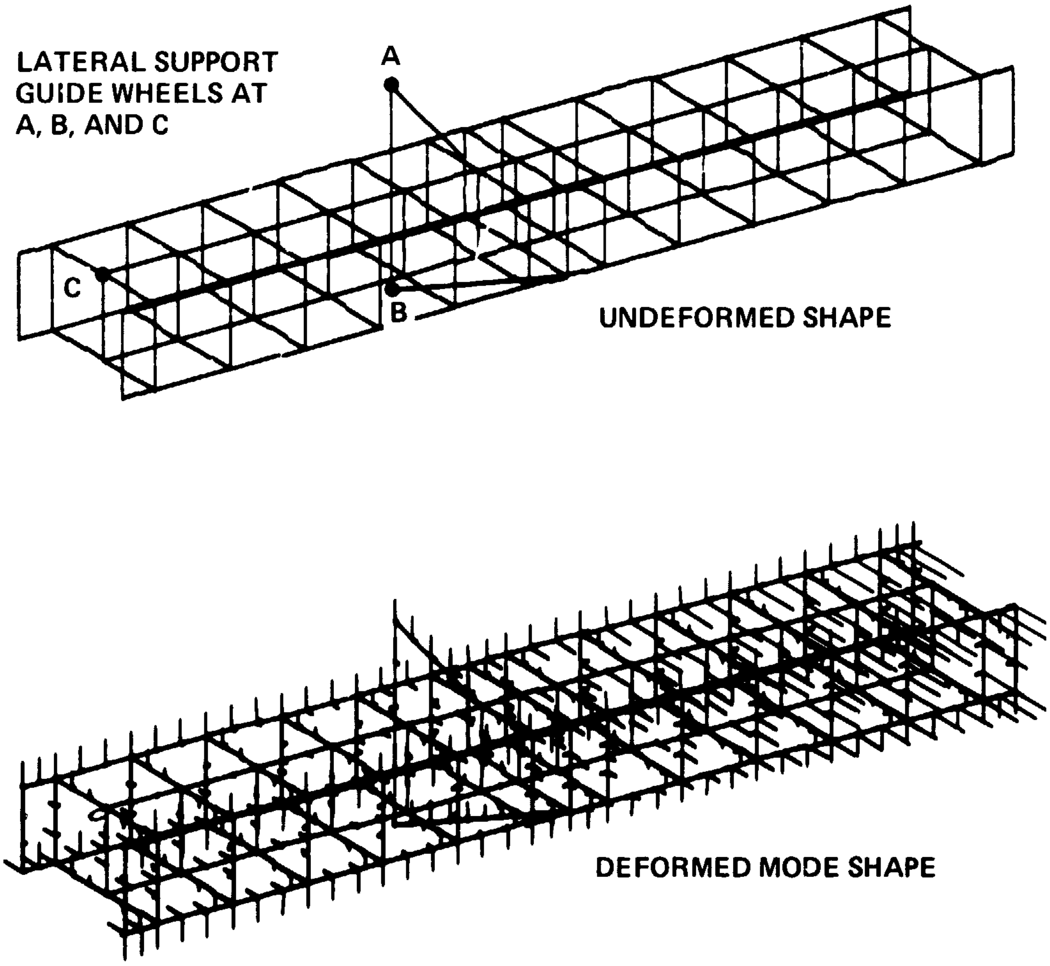

Figure 3. Structural plots of the vertical platform

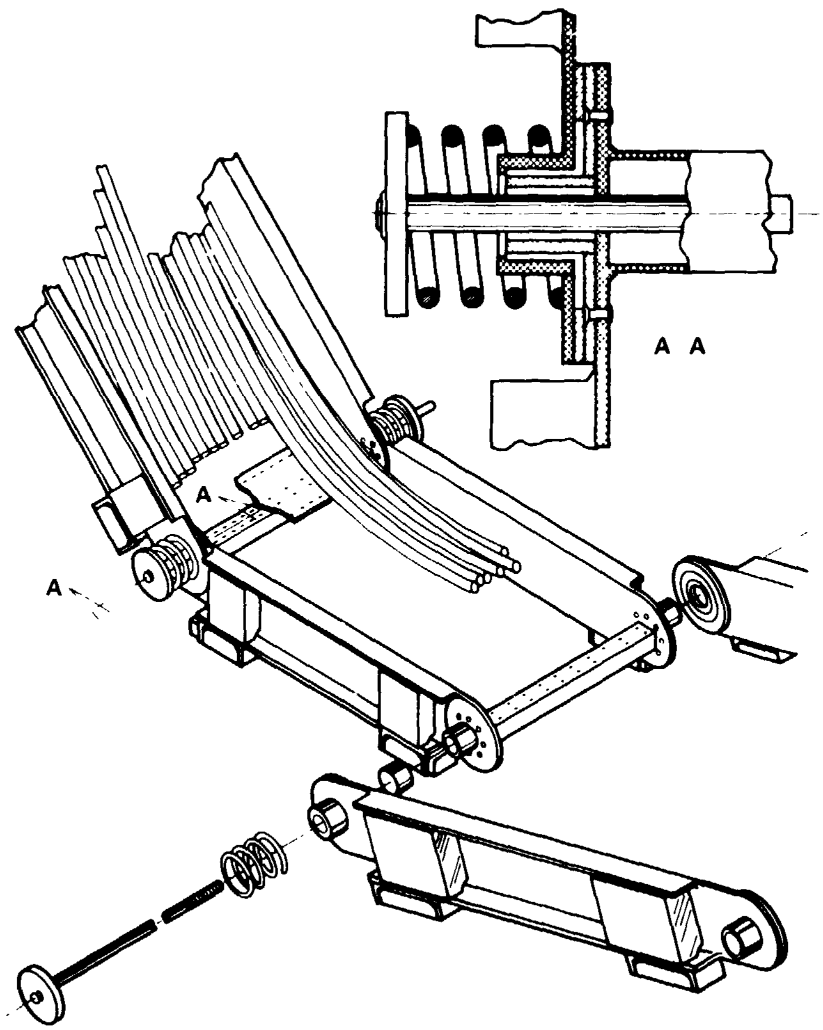

Figure 4. Catenary link detail

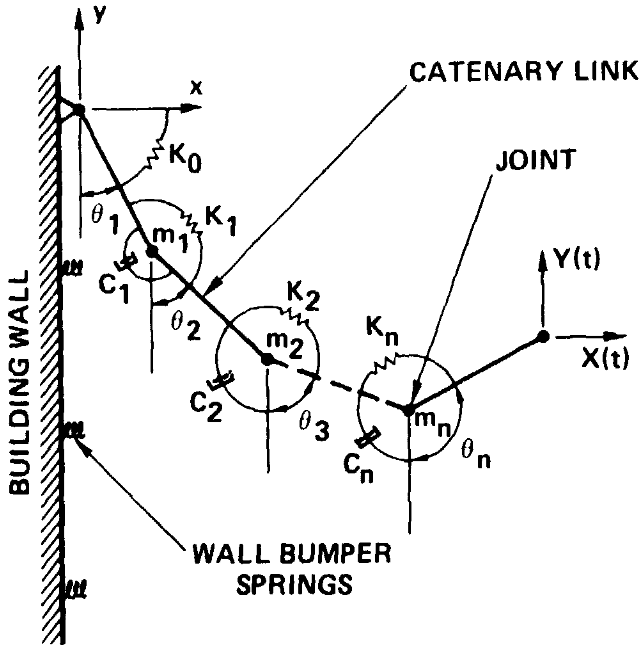

Figure 5. Catenary mathematical model

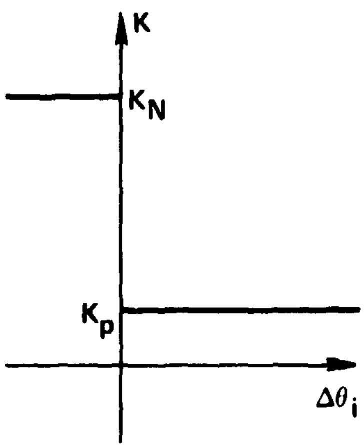

Figure 6. Catenary torsional spring characteristics

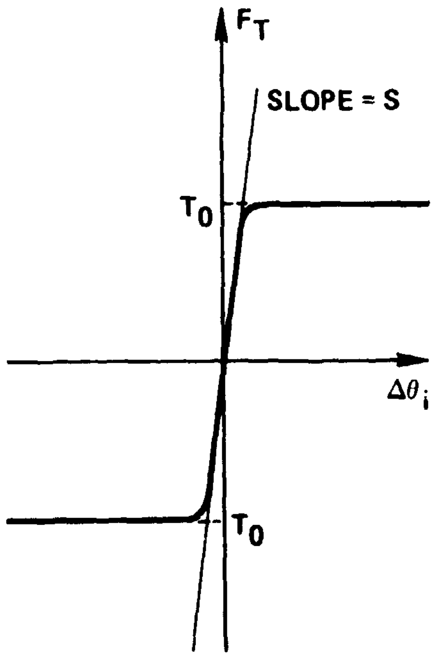

Figure 7. Catenary frictional torque