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Research paper

Design, analysis, and experimental validation of an active constant-force system based on a low-stiffness mechanism

Zhen Liu*, Fuliang Niu, Haibo Gao, Haitao Yu, Liang Ding, Nan Li, Zongquan Deng

State Key Laboratory of Robotics and System, Harbin Institute of Technology, Harbin 150001, China

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In low-gravity suspension simulation experiments, the partial gravitational forces of tested objects are balanced by the constant vertical forces on cables generated by constant-force systems. To improve system robustness against external payload disturbance, such systems usually employ low-stiffness mechanisms. The schematic diagram of our proposed low-stiffness mechanism is derived from an energy approach, which is especially preferable when the low-stiffness mechanism comprises two kinds of elastic components. The mechanism uses a combination of an axially arranged torsion bar and a group of radially arranged springs. While the former exhibits high energy density and generates major output force, the latter offers a negative stiffness to shape the output force curve so that it resembles a constant one. The mechanism has a comparatively smaller overall size, lower stiffness, and wider adjustable force range. The low-stiffness mechanism is used to form an active constant-force system. The system, as well as its dynamic model and controller, are also detailed in this paper. Experimental results demonstrate that the active constant-force system can be robustly controlled by a proportional-derivative controller with incomplete derivation to generate a high-accuracy dynamic force.

1. Introduction

One of the most significant current discussions in astronaut training and spacecraft reliability assessment is the ground simulation of a low-gravity environment on the planet's surface [1, 2]. An alternative method for the simulation is using slings to apply vertical forces on tested objects to compensate for their partial gravity [3, 4]. Such a method usually consists of a horizontal position tracking system maintaining the sling vertical and a constant-force system keeping the sling force constant. The performance of a constant-force system is a key factor to ensure high-fidelity simulation. Two requirements for constant-force systems are high steady-state force accuracy and excellent dynamic response.

Currently, research results regarding to constant-force systems can be classified into passive constant-force systems and active constant-force systems (ACFSs). Passive constant-force systems are generally classified in terms of compensation components into two types: counterweights [5–7] and buffer springs [8–10]. They are simple in structure; however, the counterweight inertia and spring deformation significantly affect the force accuracy and dynamic characteristics. Consequently, they are suitable for low-speed or low-force accuracy applications. ACFSs are divided on the basis of drive source and composition

* Corresponding author.
E-mail address: zhenliu.hit@gmail.com (Z. Liu).

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Nomenclature

L	Lagrangian operator (J)
V_{LSM}	Elastic potential energy of LSM (J)
q	Generalised degree of freedom and angle between input and output terminals of low-stiffness mechanism (LSM) (°)
$Q(q)$	Generalised force (N m)
C	Stored elastic energy at initial working point (J)
Q_C	Setting generalised force (N m)
T'	Torsional stiffness of torsion bar (N m/°)
q_{st}	Initial angle of torsion bar (°)
V_{tor}, V_{spr}	Elastic potential energy of torsion bar and compressed spring (J)
V_{ACFS}	Elastic potential energy of ACFS (J)
V_{rem}	Elastic potential energy of elastic elements except torsion bar (J)
V_{rmax}	Maximum elastic potential energy of elastic elements except torsion bar (J)
C_1	Initial elastic potential energy of elastic elements except torsion bar (J)
$q_{max(V_{rem})}$	Angle under the maximum elastic potential energy of remaining elastic elements (°)
q_H	Maximum angle of one side of the initial position of LSM(°)
$T_{E_{ACFS}}$	Torque error function of ACFS (N m)
$F_{E_{ACFS}}$	Force error function of ACFS (N)
K_{ACFS}	Equivalent stiffness of LSM and ACFS (N/mm)
d	Torsion bar diameter (mm)
L_{tor}	Torsion bar length (mm)
k	Stiffness of compressed spring (N/mm)
l_0, l_{st}	Rest and installed lengths of compressed spring (mm)
F_{pre}	Preload of compressed springs (N)
$l(q)$	Length function of compressed spring (mm)
e	Distance from outer joint of compressed spring to rotation axis of LSM (mm)
r	Distance from inner joint of compressed spring to rotation axis of LSM (locking bar with constant length) (mm)
$\alpha(q)$	Angle function between compressed spring axis and locking bar direction (°)
F_{spr}	Force of compressed spring (N)
R	Reel radius (mm)
τ	Motor torque (N m)
J_1	Equivalent moment of inertia of motor shaft, brake, reducer, and input terminal of LSM (kg cm^2)
J_2	Equivalent moment of inertia of reel and output terminal of LSM (kg cm^2)
J_{motor},J_{brake},$J_{reducer}$,$J_{coupling}$	Moment of inertia of motor, brake, reducer, and coupling (kg cm^2)
i	Transmission ratio
J_{In}	Equivalent moment of inertia of input terminal of LSM (kg cm^2)
μ_1	Equivalent viscous friction coefficient of motor shaft, brake, reducer, and input terminal of LSM (N s/mm)
μ_2	Equivalent viscous friction coefficient between input and output terminals of LSM (N s/mm)
μ_3	Equivalent viscous friction coefficient between output terminal of LSM and support (N s/mm)
θ_1	Rotation angle of input terminal of LSM (°) (Positive direction is clockwise viewed from the motor to the LSM)
θ_2	Rotation angle of output terminal of LSM (°) (Direction is the same asθ_1)
T_{LSM}	Actual torque of LSM (N m)
F_{LSM}	Actual force of LSM (N)
F_{st}	Required force of LSM (N)
F	Sling force (N)
F_{Δ}	Sling force error (N)
f	Friction force (N)
F_C	Coulomb friction force (N)
v	Relative sliding velocity (m/s)
μ	Viscous friction coefficient (N s/mm)