CUBESPACE

Characterization of Reaction Wheels for High Performance Imaging Missions

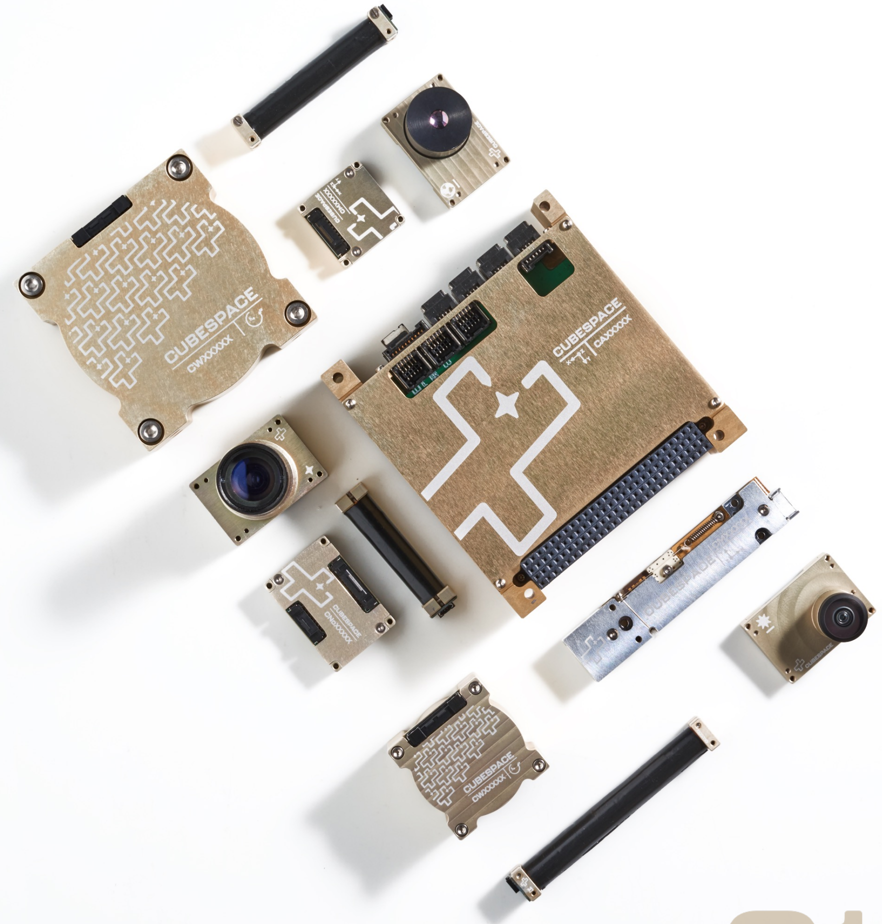
Marisa du Plessis – AIT Engineer

2024/04/24

CUBESPACE SATELLITE SYSTEMS

INTRODUCTION

- Design and manufacture ADCS for Small Satellites
- Started as a university spin out in 2014
- Controlling over 350 satellites to date
- Supplied over 3000+ ADCS parts to date
- Started with CubeSats, now scaling into MicroSats
- Expanding to EU and US in 2024



CUBESPACE

SATELLITE POINTING STABILITY

RELATION TO REACTION WHEELS

- High ground resolution specifications - stable platform
- Influenced by micro-vibrations emanating from subsystems
- Most prominent source: Reaction Wheels

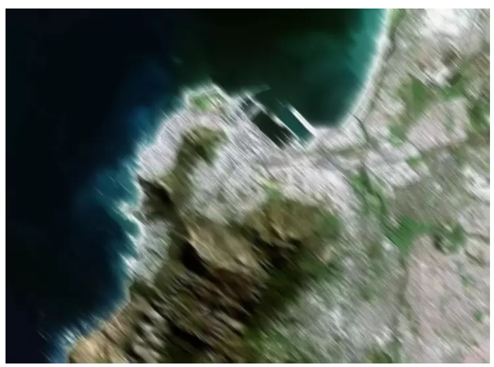
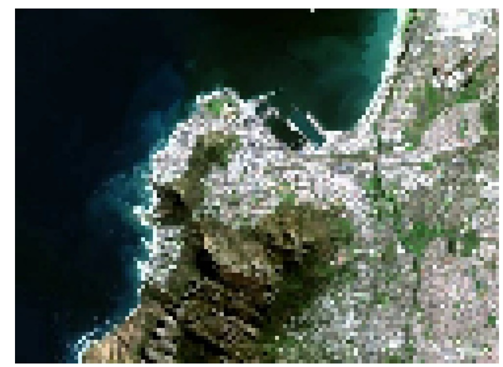


IMAGE SMEAR

IMAGE BLUR



RESOLUTION DEGRAGATION

RESEARCH OBJECTIVES

OVERVIEW

Comprehensive Literature Review

Develop a Reaction Wheel Model

Classify origins of micro-vibrations.

Experimental Measurements

Analyze Measured Data

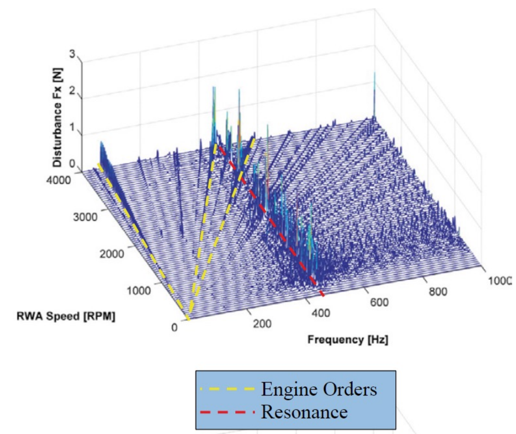
Investigate effect of micro-vibrations on a satellite system

Micro-vibration Mitigation

MICRO-VIBRATION SOURCES

INTRODUCTION

- Periodic micro-disturbances presenting as (F_X, F_Y, F_Z) and (M_X, M_Y, M_Z)
- Low amplitude mechanical vibrations
- From 1 Hz up to 1 kHz
- Signals with distinct characteristics
  - Structural Cage Modes
  - Engine Orders
- Signal clearly visualized and identified in waterfall plots

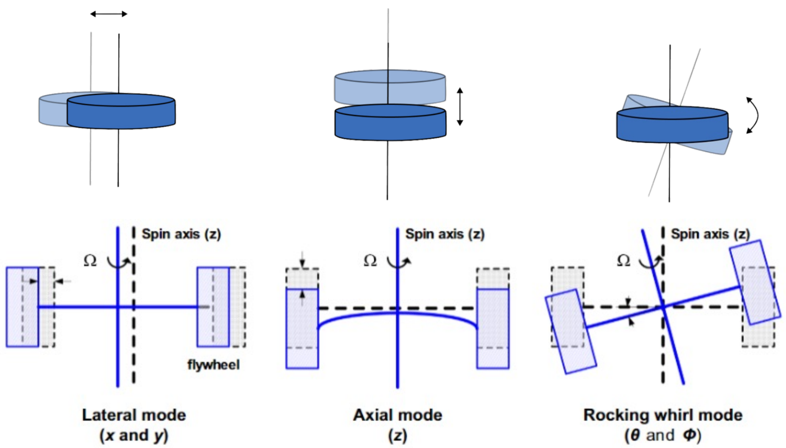


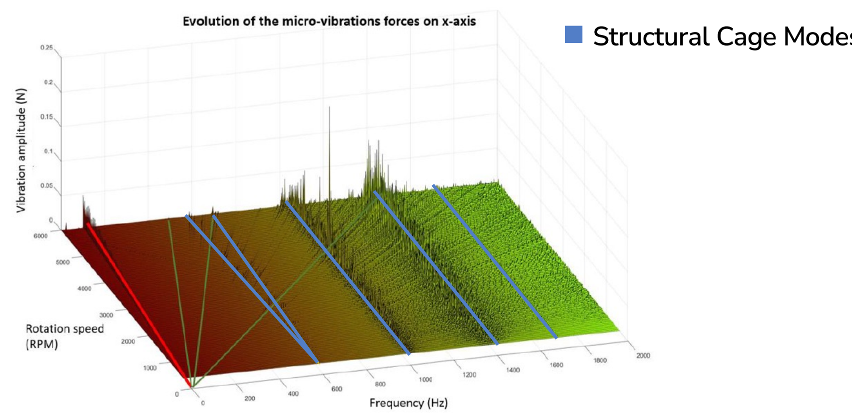
CUBESPACE

MICRO-VIBRATION SOURCES

STRUCTURAL CAGE MODES

- Natural Frequencies or Eigenmodes
- Specific modes of vibration exhibited by the structure of mechanical system
- Three modes to investigate,
- 5 degrees of freedom in the system:



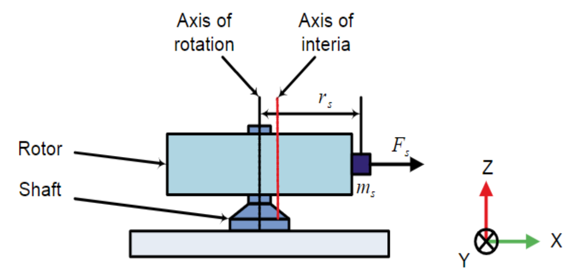


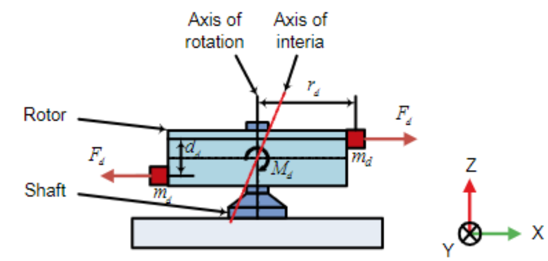
MICRO-VIBRATION SOURCES

FLYWHEEL UNBALANCE

- Often most significant source of micro-vibration emitted by a reaction wheel
- Caused by residual uneven distribution of mass around the rotor's axis of rotation

Static Unbalance	Dynamic Unbalance
Uneven distribution of mass in the plane of rotation	Uneven distribution of mass in the planes parallel to the rotational plane of the flywheel
$F_s = m_s r_s \Omega^2 = U_s \Omega^2$	$M_d = m_d r_d d_d \Omega^2 = U_d \Omega^2$

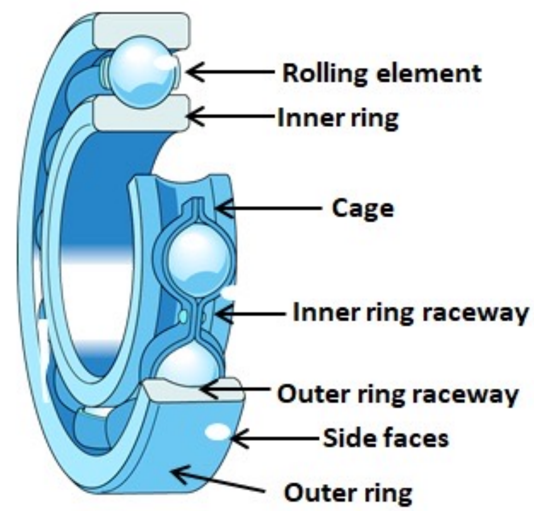




MICRO-VIBRATION SOURCES

BALL BEARING IMPERFECTIONS

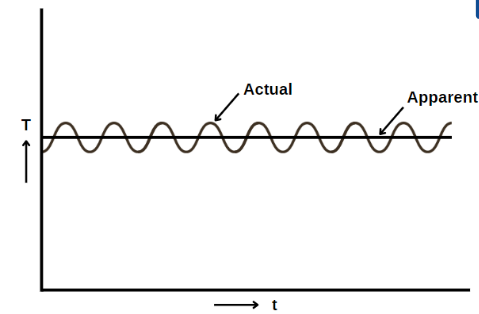
Race Ball Pass Frequency	Fundamental Cage Frequency	Ball Spin Frequency
Corresponds to number of rolling elements passing a set point of the **inner/outer race** each time the shaft makes a complete turn	Corresponds to number of turns the bearing cage make each time the shaft makes a complete turn	Corresponds to the Number of turns that rolling element makes each time the shaft makes a complete turn (rolling element failing frequency)
$-\frac{N\ \mathrm{fr}\left(\frac{\mathrm{dd}\cos(\alpha)}{\mathrm{D}}-1\right)}{2}$ $\frac{N\ \mathrm{fr}\left(\frac{\mathrm{dd}\cos(\alpha)}{\mathrm{D}}+1\right)}{2}$	$-\frac{\mathrm{fr}\left(\frac{\mathrm{dd}\cos(\alpha)}{\mathrm{D}}-1\right)}{2}$ $\frac{\mathrm{fr}\left(\frac{\mathrm{dd}\cos(\alpha)}{\mathrm{D}}+1\right)}{2}$	$-\frac{\mathrm{D\ fr}\left(\frac{\mathrm{dd}^2}{\mathrm{D}^2\cos(\alpha)^2}-1\right)}{2\,\mathrm{dd}}$



MICRO-VIBRATION SOURCES

TORQUE RIPPLE

- Variation in torque output of a motor as it rotates through a complete revolution
- Primarily caused by interaction between magnetic fields of rotor and stator



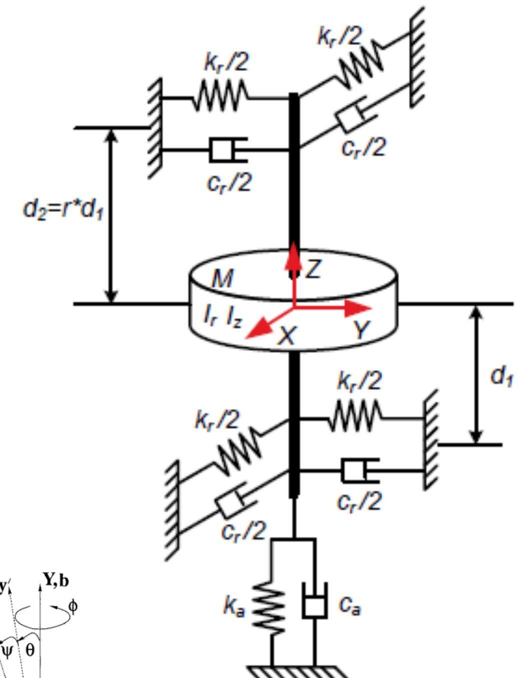
Torque Ripple

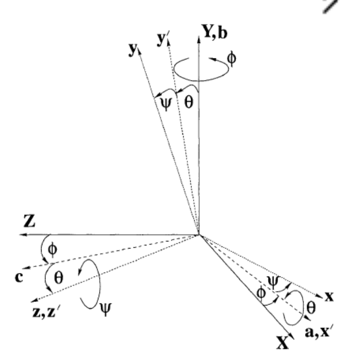
Cogging Torque	Commutation Torque	Zero Crossing Stiction	Harmonic Torque Ripple
Interaction: permanent magnets on the rotor and the teeth on the stator	High frequency stator phase switching. Inversely proportional to the speed.	Static Friction	Variations in torque due to presence of higher order electromagnetic harmonics in the motor
$f_{cogging} = N_{poles}\,\Omega$			

THEORETICAL MODEL

EQUATIONS OF MOTION: BALANCED FLYWHEEL

- Reference Frame: Euler Rotation Y – X – Z
- Derive using Lagrange Equation
- Determine Kinetic Energy *(T)* and Potential Energy *(V)*
- Lagrangian: $L = T - V$
- External forces generated by the viscous dampers: δW
- For each degree of freedom of the system, an EOM is derived from the Euler-Lagrange equation: $\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}}\right) - \frac{\partial L}{\partial x} = \frac{\partial(\delta W)}{\partial(\delta x)}$





CUBESPACE

THEORETICAL MODEL

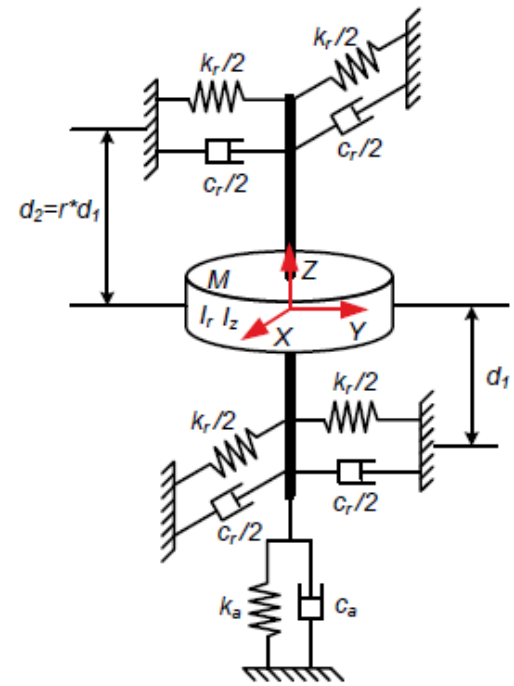
EQUATIONS OF MOTION: BALANCED FLYWHEEL

The set of 5 equations of motions presenting the balanced rotor-bearing system is linearized and rearranged into matrix form to become:

$$[\boldsymbol{M_b}] \cdot \boldsymbol{a} + [\boldsymbol{C_b}(\Omega, t)] \cdot \boldsymbol{v} + [\boldsymbol{K}] \cdot \boldsymbol{u} = [\boldsymbol{0}]$$

Where:

$$\boldsymbol{a} = \begin{matrix} \ddot{x} \\ \ddot{y} \\ \ddot{z} \\ \ddot{\theta} \\ \ddot{\Phi} \end{matrix}, \qquad \boldsymbol{v} = \begin{matrix} \dot{x} \\ \dot{y} \\ \dot{z} \\ \dot{\theta} \\ \dot{\Phi} \end{matrix}, \qquad \boldsymbol{u} = \begin{matrix} x \\ y \\ z \\ \theta \\ \Phi \end{matrix}$$

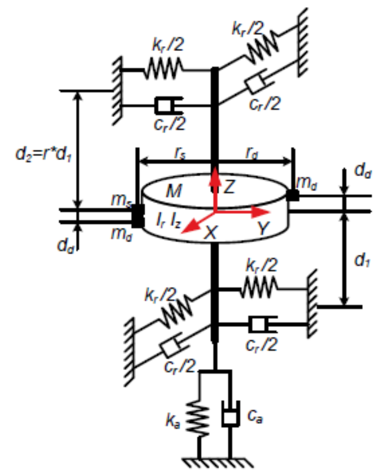


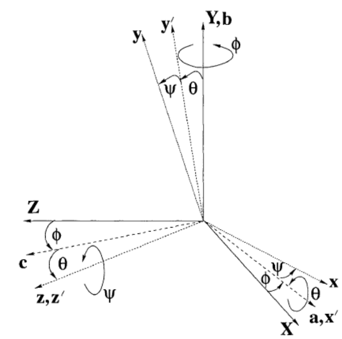
THEORETICAL MODEL

EQUATIONS OF MOTION:

UNBALANCED FLYWHEEL

- Add unbalance masses m_s and m_d
- $M_t = M_b + m_s + 2m_d$
- Translational motions now includes total mass and excitation of the static unbalance, U_s.
- Rocking motions now considers additional inertia and the excitation of the dynamic unbalance, U_d.





$$\begin{pmatrix} x_{\mathrm{dd}} + \frac{\mathrm{cr}\, x_d}{\mathrm{Mt}} + \frac{\mathrm{kr}\, x}{\mathrm{Mt}} - \frac{\mathrm{cr}\, d_1\, \phi_d\, (r-1)}{2\,\mathrm{Mt}} - \frac{d_1\, \mathrm{kr}\, \phi\, (r-1)}{2\,\mathrm{Mt}} = -\frac{\Omega^2\, \mathrm{Us}\, \sin(\Omega\, t)}{\mathrm{Mt}} \\ y_{\mathrm{dd}} + \frac{\mathrm{cr}\, y_d}{\mathrm{Mt}} + \frac{\mathrm{kr}\, y}{\mathrm{Mt}} - \frac{\mathrm{cr}\, d_1\, \theta_d\, (r-1)}{2\,\mathrm{Mt}} - \frac{d_1\, \mathrm{kr}\, \theta\, (r-1)}{2\,\mathrm{Mt}} = \frac{\Omega^2\, \mathrm{Us}\, \cos(\Omega\, t)}{\mathrm{Mt}} \\ z_{\mathrm{dd}} + \frac{\mathrm{ca}\, z_d}{\mathrm{Mt}} + \frac{\mathrm{ka}\, z}{\mathrm{Mt}} = 0 \\ \theta_{\mathrm{dd}} - \frac{\theta_d\, \sigma_1}{\mathrm{Ir}} - \frac{\mathrm{Iz}\, \Omega\, \phi_d}{\mathrm{Ir}} - \frac{\mathrm{cr}\, d_1\, y_d\, (r-1)}{2\,\mathrm{Ir}} - \frac{d_1\, \mathrm{kr}\, y\, (r-1)}{2\,\mathrm{Ir}} + \frac{{d_1}^2\, \mathrm{kr}\, \theta\, \left(r^2+1\right)}{2\,\mathrm{Ir}} = \frac{\Omega^2\, \mathrm{Ud}\, \cos(\Omega\, t)}{\mathrm{Ir}} \\ \phi_{\mathrm{dd}} - \frac{\phi_d\, \sigma_1}{\mathrm{Ir}} + \frac{\mathrm{Iz}\, \Omega\, \theta_d}{\mathrm{Ir}} - \frac{\mathrm{cr}\, d_1\, x_d\, (r-1)}{2\,\mathrm{Ir}} - \frac{d_1\, \mathrm{kr}\, x\, (r-1)}{2\,\mathrm{Ir}} + \frac{{d_1}^2\, \mathrm{kr}\, \phi\, \left(r^2+1\right)}{2\,\mathrm{Ir}} = \frac{\Omega^2\, \mathrm{Ud}\, \sin(\Omega\, t)}{\mathrm{Ir}} \end{pmatrix}$$

where

$$\sigma_1 = \mathrm{Iz}\,\Omega\, \sin(2\,\Omega\, t) - \frac{\mathrm{cr}\, {d_1}^2\, \left(r^2+1\right)}{2}$$

THEORETICAL MODEL

EQUATIONS OF MOTION: UNBALANCED FLYWHEEL

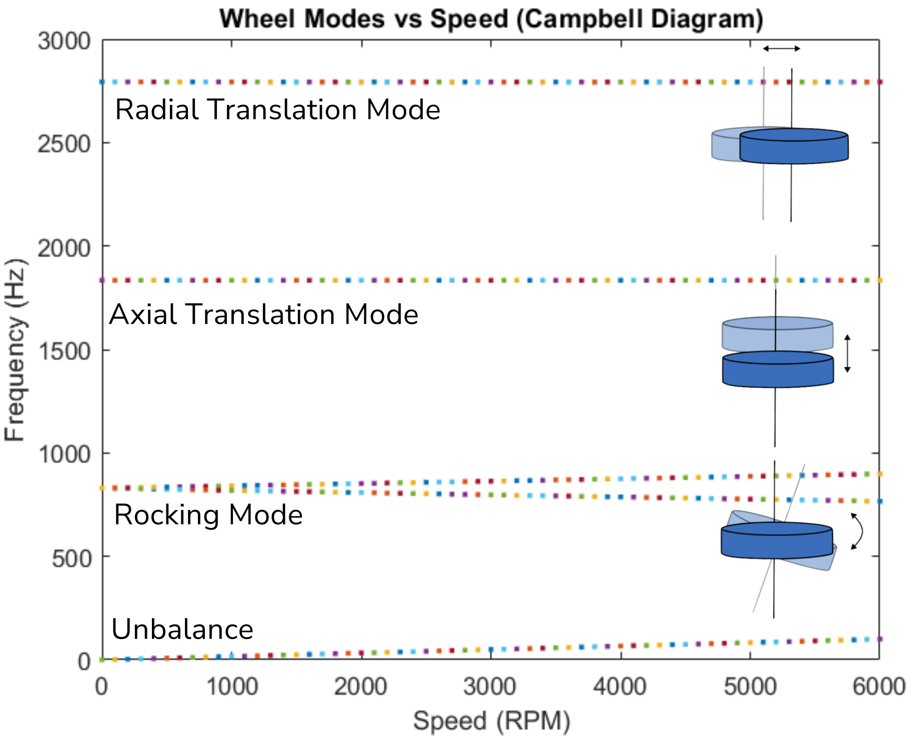
The set of equations are considered the basic form of the theoretical micro-vibration model, and is presented in matrix form as:

$$\boldsymbol{a} + [\boldsymbol{C}_{ub}(\Omega, \boldsymbol{t})] \cdot \boldsymbol{v} + [\boldsymbol{K}] \cdot \boldsymbol{u} = [\boldsymbol{Ub}(\Omega, \boldsymbol{t})]$$

Where: $\boldsymbol{a} = \begin{matrix} \ddot{x} \\ \ddot{y} \\ \ddot{z} \\ \ddot{\theta} \\ \ddot{\Phi} \end{matrix}, \quad \boldsymbol{v} = \begin{matrix} \dot{x} \\ \dot{y} \\ \dot{z} \\ \dot{\theta} \\ \dot{\Phi} \end{matrix}, \quad \boldsymbol{u} = \begin{matrix} x \\ y \\ z \\ \theta \\ \Phi \end{matrix}$

THEORETICAL MODEL

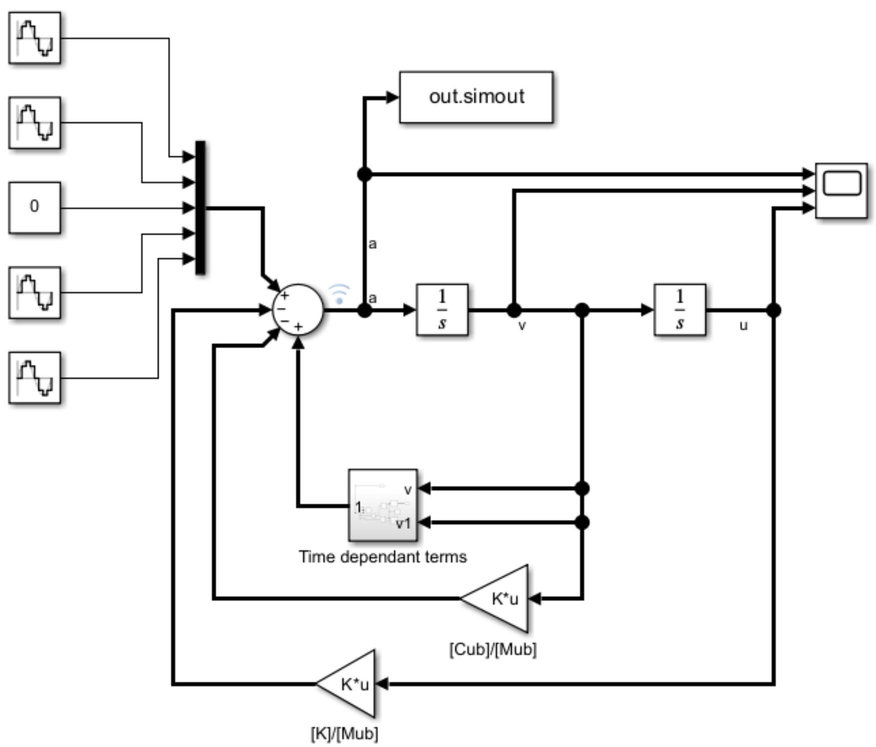
MODEL OUTPUT: CAMPBELL DIAGRAM



- Presentation of resonant frequencies as function of speed
- $c_r = c_a = 0$ and $U_s = U_d = 0$
- State matrix $\boldsymbol{A} = \begin{bmatrix} -[\boldsymbol{C_{ub}}(\Omega)] & -[\boldsymbol{K}] \\ [\boldsymbol{I_5}] & [\boldsymbol{0}] \end{bmatrix}$
- Calculate eigenvalues of A to find resonant frequencies

THEORETICAL MODEL

MODEL OUTPUT: FFT RESPONSE AT SET SPEEDS



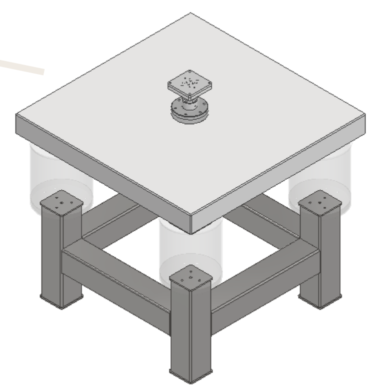
- Solve Simulink model – set speeds
- Aacceleration $\boldsymbol{a} = [\ddot{x} \quad \ddot{y} \quad \ddot{z} \quad \ddot{\theta} \quad \ddot{\Phi}]^T$ determined
- Acceleration multiplied by either mass of moment of inertia
- Obtain forces and torques as a function of time
- Transform into frequency domain and FFT
- Plots for each speed can be stacked to create waterfall plot comparable to experimental data

CUBESPACE

EXPERIMENTAL SETUP

DATA ACQUISITION SYSTEM

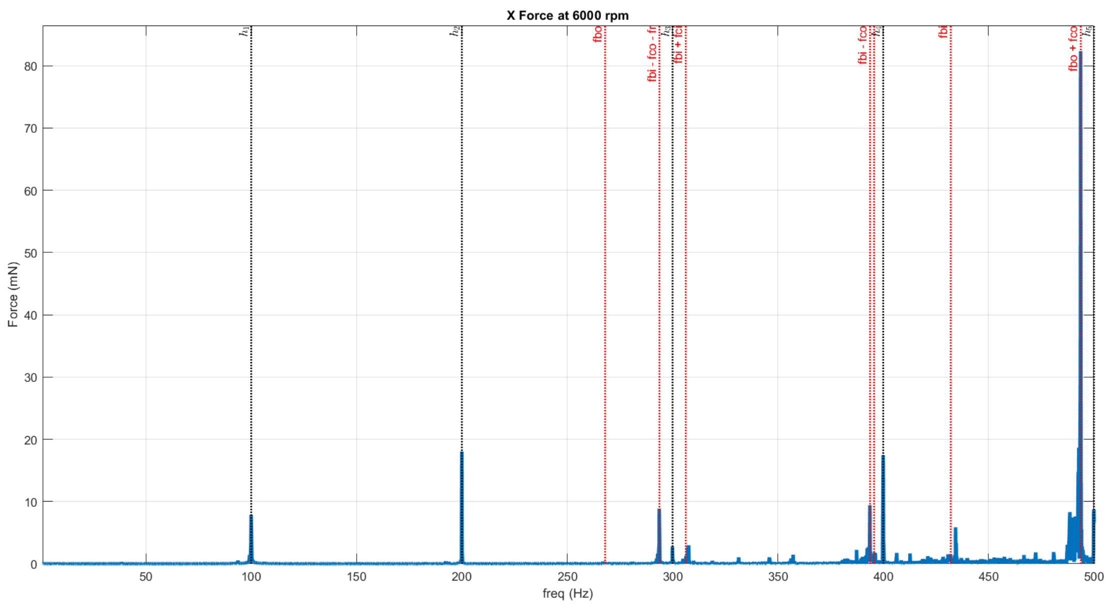
Components
Kistler piezoelectric force-torque sensor
Kistler Multichannel Laboratory Charge Amplifier
Kistler Laboratory Charge Amplifier
Pneumatic Isolation System
Processing Software

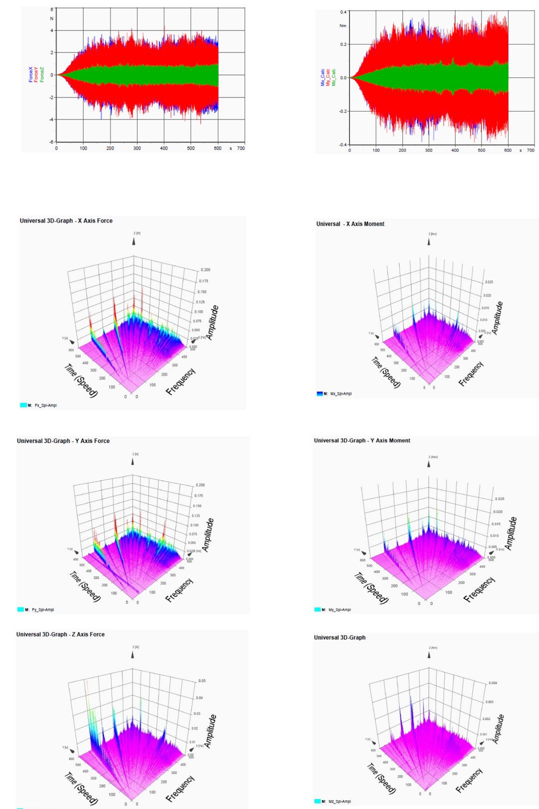


CUBESPACE

PRELIMINARY RESULTS

DATA REPRESENTATION





CUBESPACE

CONCLUSION
SUMMARY AND FUTURE WORK
Characterizes Micro-vibration Sources in Reaction Wheels
Refine Theoretical Model through further experiments and analysis
Developed Basic Theoretical Micro-vibration Model
Investigate Propagation of Micro-vibrations through Satellite
Design and Implementation of Experimental Setup
Successful Mitigation of Micro-vibrations
CUBESPACE
17

CUBESPACE

GET IN TOUCH

INFO@CUBESPACE.CO.ZA

@cubespace_adcs

@cubespaceadcs

@cubespace adcs

@cubespace_adcs

www.cubespace.co.za