High Speed Elevator

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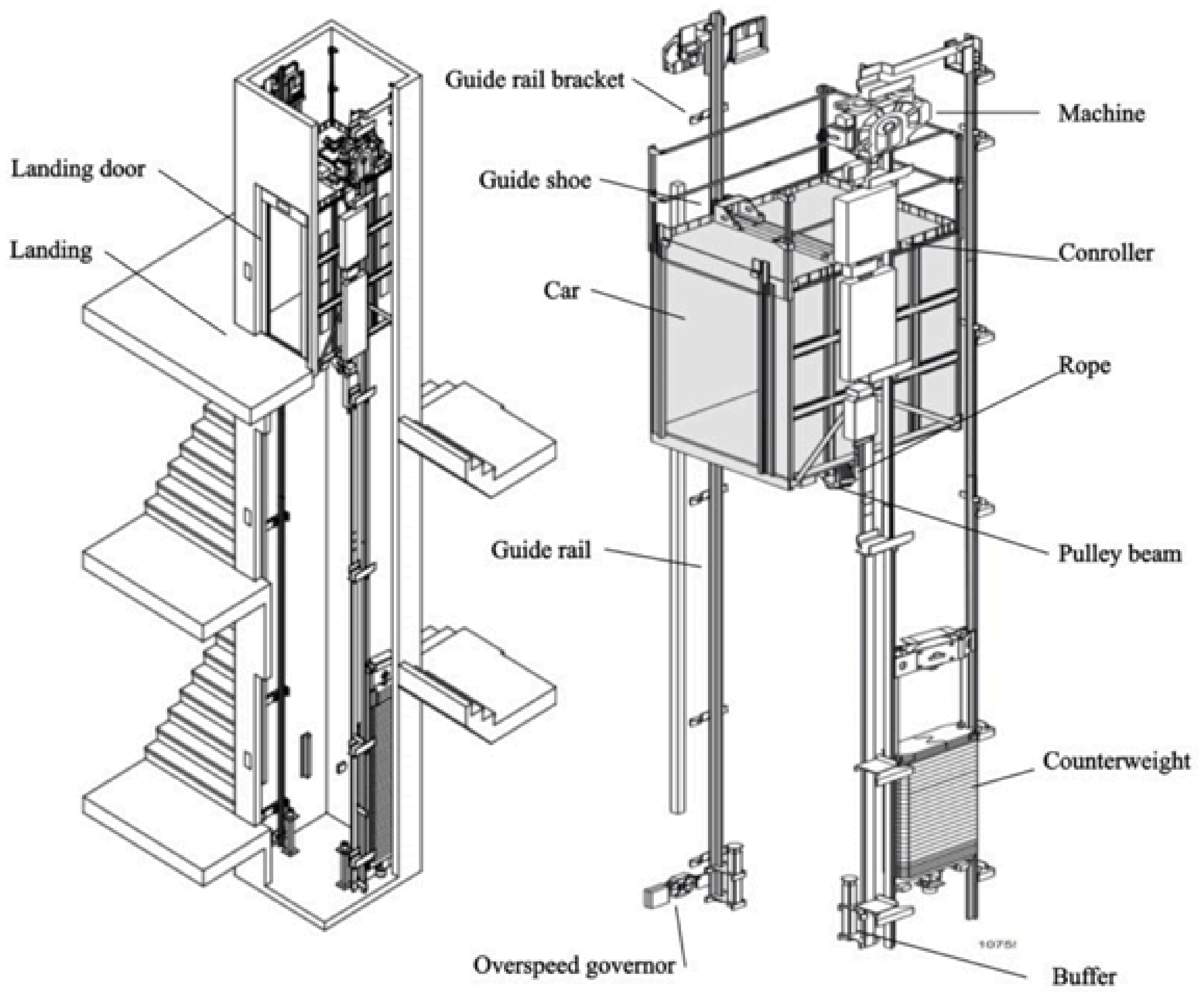
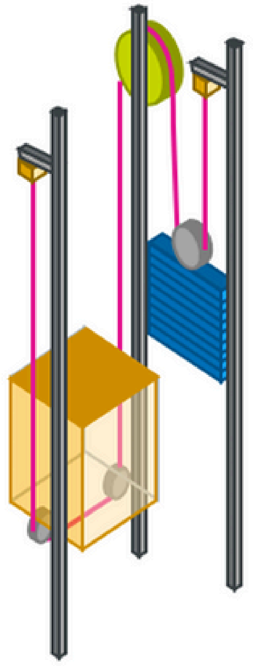
High Speed Elevator System

Schematic Diagram





Components of Elevator

Hoisting & Drive System

Machine: The motor that powers the elevator by turning the sheave that moves the ropes

Rope: The steel cables that connect the car and counterweight, used to lift and lower them.

Pulley Beam: A structural support at the top of the hoistway for the machine or pulleys.

Moving Components & Balancing

Car: The cabin that transports passengers or freight between floors.

Car frame : It is the steel structural framework that supports the elevator car and connects it to the guide rails.

Counterweight: A heavy mass that balances the car's weight to save energy.

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High Speed Elevator System

Guidance System

Guide Rail: The vertical steel tracks that guide the car and counterweight.

Guide Shoe: Components fixed to the car and counterweight that slide along the guide rails for alignment.

Guide Rail Bracket: The fixtures that secure the guide rails to the hoistway walls.

Control & Safety Systems

Controller: The "brain" of the elevator that manages calls, speed, direction, and safety.

Overspeed Governor: A safety device that trips the brakes if the car descends too fast.

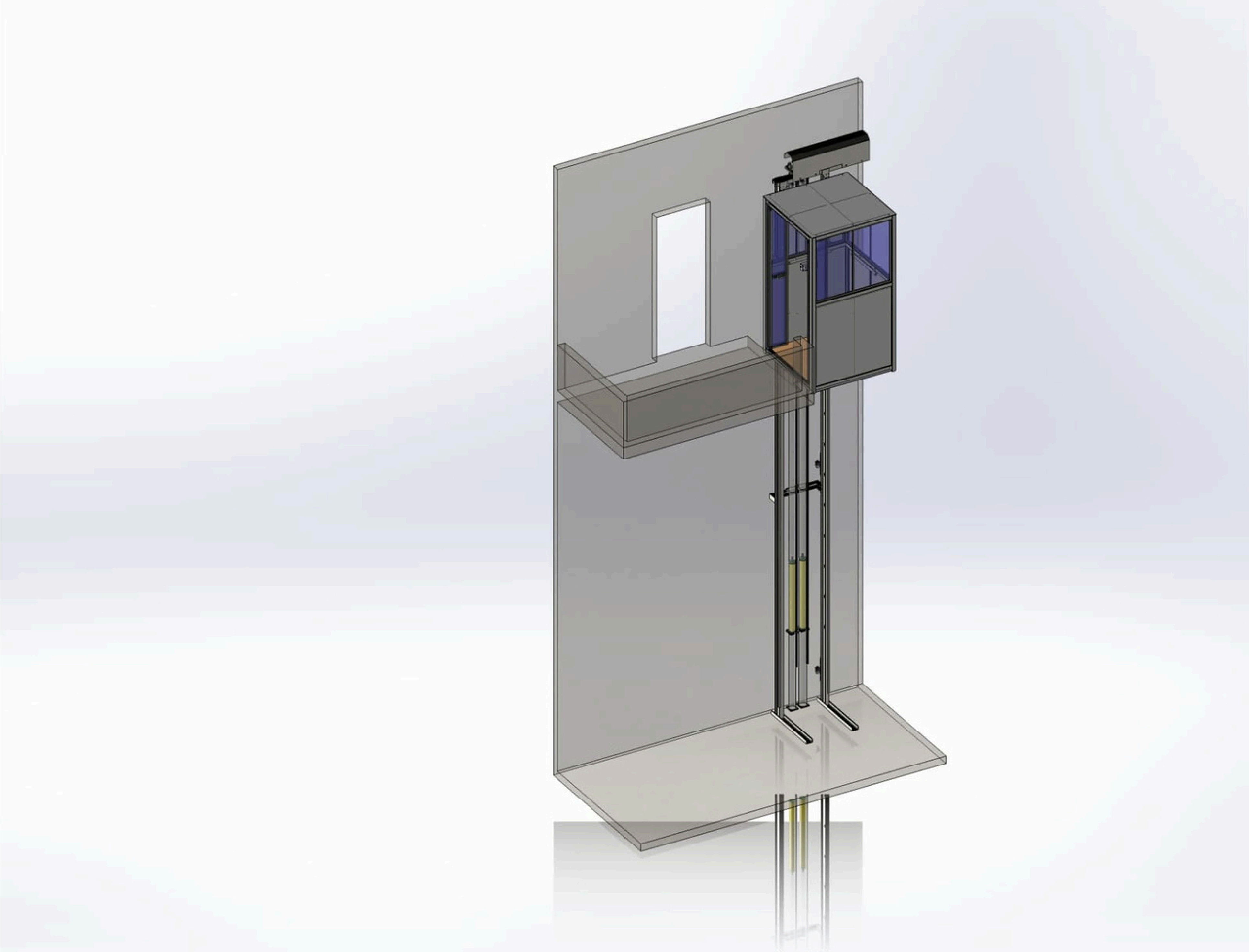
Buffer: A shock absorber at the bottom of the hoistway to safely stop the car or counterweight.

Building Integration

Landing: The floor area outside the elevator shaft.

Landing Door: The doors on each floor that provide access to the elevator car.

CAD File



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High Speed Elevator System

Made in Fusion360

High-Speed Elevator: General Overview

- High-speed elevators are advanced vertical transportation systems designed to move passengers rapidly between floors of high-rise and supertall buildings(skyscrapers).
- Their speeds typically range from 5 m/s to over 20 m/s, offering efficient, safe, and comfortable travel.
- High-speed motion causes mechanical vibrations that affect comfort and stability
- Vertical vibrations arise from Uneven Tension or Elastic Stretch in Hoisting Ropes, traction and motor irregularities.
- Horizontal vibrations result from guide rail misalignment and cabin sway.
- Damping and isolation systems are used to reduce these effects.

Core Objectives

- Speed with Safety: Achieving higher velocities without compromising passenger comfort.
- Energy Efficiency: Use of regenerative drives to reduce energy consumption.
- Ride Comfort: Smooth acceleration/deceleration and minimized sway.
- Vibration & Noise Control: Implementing vibration isolation models and aerodynamic cabin designs.

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High Speed Elevator System

COMPONENT DEFINITIONS – ELEVATOR VIBRATION MODEL (3-DOF LUMPED MASS SYSTEM)

- **Elevator Car (Cabin)**

A rigid body that carries passengers; modeled as a mass moving vertically under gravity and rope tension.

- **Counterweight**

A solid mass connected to the car by ropes over the traction wheel; moves in the opposite direction to balance the car's weight.

- **Rope Stiffness (k)**

A property representing how much force is needed to stretch the rope by a unit displacement; determines how vibration transmits through the rope.

- **Rope Damping (C)**

Coefficient representing energy loss due to internal friction within the rope material during vibration.

- **Guide Rails**

Vertical tracks that guide the car and counterweight, preventing lateral motion and keeping the movement purely vertical.

- **Traction Wheel (Sheave)**

A rotating pulley with moment of inertia and radius ; transmits motor torque or braking torque to the rope for motion control.

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Types of Vibrations

Study on Theoretical Model and Test Method of Horizontal Vibration of Elevator Traction System

Background:

Modern high-speed elevators experience **horizontal vibrations** due to interaction between the **car, guide rails, guide shoes,** and **building sway**.

Primary Causes:

- **Guide-rail irregularities –** misalignment or waviness induce periodic lateral forces.
- **Nonlinear guide-shoe stiffness –** variable spring and rubber elements create asymmetric restoring forces.
- **Car mass asymmetry –** uneven load or low structural stiffness amplifies lateral motion.
- **Building sway –** wind and seismic excitation transmit low-frequency motion through rail brackets.

Secondary Factors:

Rope coupling, aerodynamic forces in the shaft, and material stiffness changes with temperature.

Impact:

Horizontal vibration affects **ride comfort**, **safety**, and **system stability**, making it a critical research focus in elevator dynamics.

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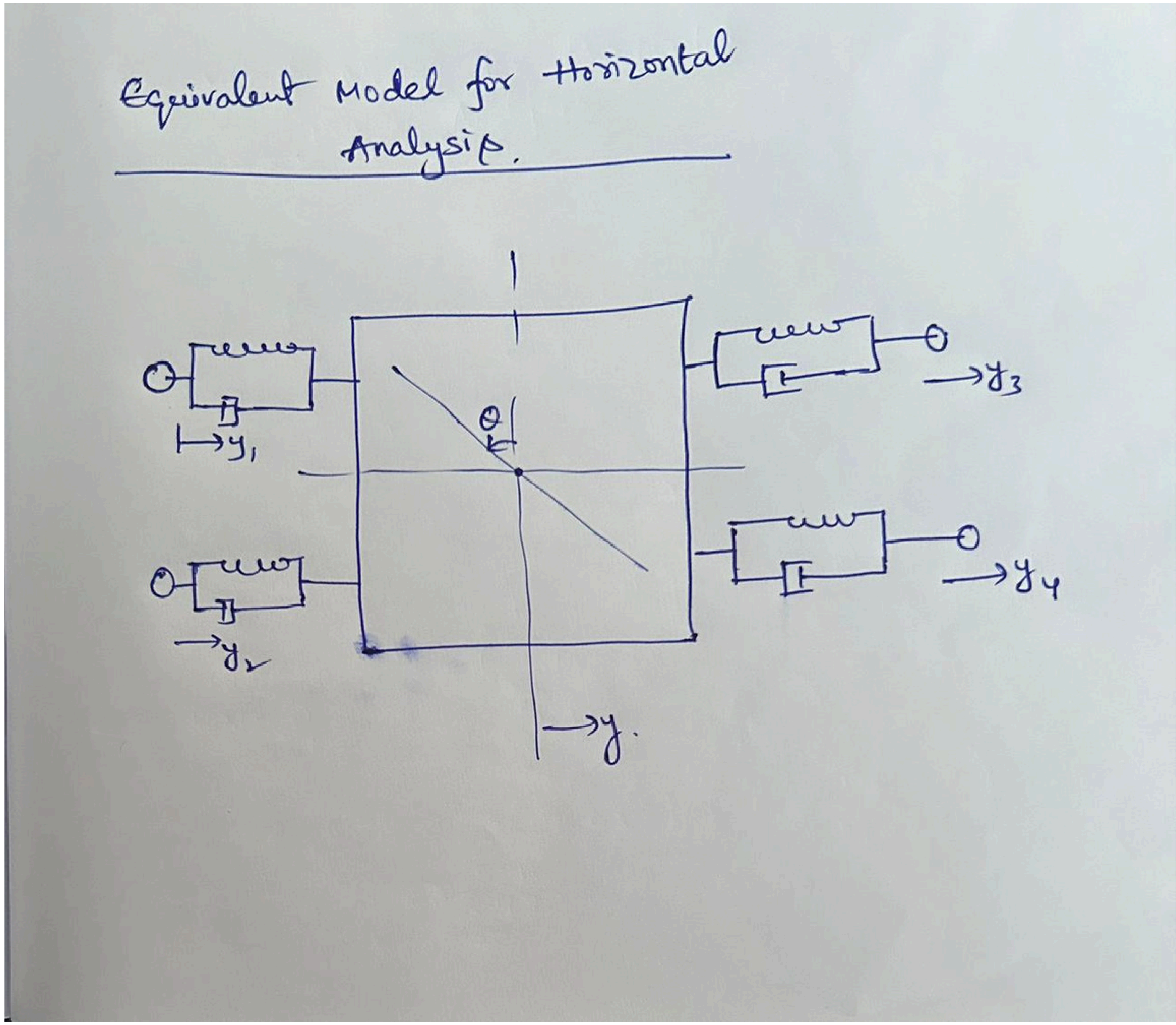
Horizontal Vibrations

Simplified Model for Horizontal Dynamic Analysis



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Horizontal Vibrations

Key Components in Lateral Vibrations Coupling

- **Elevator Car:**
- Modeled as a rigid body with **6 DOF** (three translational + three rotational).
- Lateral motion dominated by **y-direction translation** and **yaw rotation.**

- **Guide Shoes:**
- Four contact points (upper & lower).
- Modeled using **spring–damper systems** with equivalent stiffness $k_g = \frac{k_s k_r}{k_s + k_r}$ and damping $c_g = c_s + c_r.$

- Nonlinear contact characteristics cause parametric excitation.
- **Guide Rails & Brackets:**
- Modeled as Euler–Bernoulli beams with multi-support conditions:

$$EI\frac{\partial^4 y_r}{\partial z^4} + \rho A\frac{\partial^2 y_r}{\partial t^2} = q(z,t)$$

- Building sway imposes base motion $y_b(z,t) = Y_b(z)\sin(\omega_b t + \phi(z))$
- **Building Structure:**
- Transfers **low-frequency excitation** to rails.
- **System Coupling:**
- Car–rail interaction forms a **coupled dynamic system**, solved via numerical integration.

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Horizontal Vibrations

Mathematical Modeling

Car Dynamic Equation:

$$m_c\ddot{y}_c + c_g(\dot{y}_c - \dot{y}_r) + k_g(y_c - y_r) = 0$$

Rail Equation:

$$EI\frac{\partial^4 y_r}{\partial z^4} + \rho A\frac{\partial^2 y_r}{\partial t^2} = \sum_i F_i\delta(z - z_i)$$

Combined System:

$$\mathbf{M}\ddot{\mathbf{y}} + \mathbf{C}\dot{\mathbf{y}} + \mathbf{K}\mathbf{y} = \mathbf{F}(t)$$

Matrix Form:

$$\begin{bmatrix} m & 0 \\ 0 & J \end{bmatrix}\begin{bmatrix} \ddot{y} \\ \ddot{\theta} \end{bmatrix} + \begin{bmatrix} 4c & 2c(l_2 - l_1) \\ 2c(l_2 - l_1) & 2c(l_1^2 + l_2^2) \end{bmatrix}\begin{bmatrix} \dot{y} \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} 4k & 2k(l_2 - l_1) \\ 2k(l_2 - l_1) & 2k(l_1^2 + l_2^2) \end{bmatrix}\begin{bmatrix} y \\ \theta \end{bmatrix}$$

$$= \begin{bmatrix} \sum_{i=1}^{4}(c\dot{y} + ky_i) \\ cl_2(\dot{y}_2 + \dot{y}_4) - cl_1(\dot{y}_1 + \dot{y}_3) + kl_2(y_2 + y_4) - kl_1(y_1 + y_3) \end{bmatrix}.$$

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Horizontal Vibrations

Influencing Factors of High Speed Elevator Horizontal Vibration

Influence of the Car system on HsEHV

Moment of Inertia: A higher moment of inertia resists changes in rotational motion, thereby reducing horizontal vibration sensitivity but making response slower.

Mass & Load: Increasing the mass or load generally lowers the natural frequency, reducing vibration amplitude but potentially causing more sluggish dynamic response.

Influence of the Lifting speed on HsEHV

Lifting Speed: As the elevator speed increases, aerodynamic and guide interaction forces rise, which can amplify horizontal vibrations unless damping and alignment are optimized.

Influence of the Guide system on HsEHV

Roller Guide Shoe Parameters: Stiffer or poorly damped roller guide shoes transmit more vibration, while optimized stiffness and damping reduce horizontal oscillations.

Guide Shoe Position: Placing guide shoes farther apart increases stability and lowers horizontal vibration, while closer positioning makes the system more prone to sway.

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Horizontal Vibrations

IMP code Snippets for Horizontal Analysis

```
% --- NATURAL FREQUENCY ANALYSIS ---
%used modal analysis
% This script calculates the natural frequencies of the 2-DOF elevator model
% for two cases:
% 1. BEFORE: Using initial parameters from Table 1
% 2. AFTER: Using optimized parameters from Section 5.3
%
% The calculation is based on the system matrices [M] and [K] from
% Equation (10) in the paper. It solves the generalized eigenvalue
% problem: (K - w^2*M)u = 0


% --- Common Parameters (from Table 1)
m = 1535;     % Mass of the car system (kg)
J = 8090;     % Moment of inertia of the car (kg*m^2)

% 1. "BEFORE" OPTIMIZATION (Initial Parameters from Table 1)


fprintf('--- 1. BEFORE Optimization Frequencies ---\n');

% Initial Parameters
k_init = 1e5;      % Initial stiffness (N/m)
l1_init = 2.8;     % Initial l1 (m)
l2_init = 3.7;     % Initial l2 (m)

% Build the Mass and Stiffness matrices (from Equation 10)
M_init = [m, 0; 0, J];
K_init = [4*k_init, 2*k_init*(l2_init - l1_init);
          2*k_init*(l2_init - l1_init), 2*k_init*(l1_init^2 + l2_init^2)];

% Solve the generalized eigenvalue problem: K*v = lambda*M*v
% where lambda = w^2 (angular frequency squared)
eig_vals_init = eig(K_init, M_init);

% Convert eigenvalues (w^2) to natural frequencies in Hz (f)
% f = w / (2*pi) = sqrt(lambda) / (2*pi)
freqs_init_hz = sort(sqrt(eig_vals_init) / (2 * pi));
```

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Horizontal Vibrations

Vertical Vibration of Elevator Traction System

Why Vibrations Happen

- **Rope elasticity (spring effect):** Wire ropes act like elastic springs — they stretch and rebound.
- **Sudden torque/force changes:** Especially during emergency braking → creates shock.
- **Inertia of components:** Car, counterweight, and traction wheel resist sudden motion changes.
- **Uneven rope tension:** Rope may slack (upward braking) or tighten (downward braking).
- **System non-stationarity:** The rope length changes with car movement → system parameters (stiffness, mass distribution) vary with time.

Real-World Effect

During upward emergency braking, **acceleration fluctuation** and **jerk** are significantly higher than in downward braking — causing passenger discomfort and safety concerns.

What We Aim to Do

1. **Model** the elevator's vertical vibration system mathematically**.**
2. **Analyze** vibration using theoretical equations (Lagrange's method).
3. **Simulate** system motion in MATLAB using ODE solvers.
4. **Measure experimentally** (using accelerometers) to validate the theory.
5. **Suggest methods to minimize vibration** (vibration isolation and damping improvement).

Why This Study Is Needed

Traditional studies ignored **traction wheel rotation**, **rope slip**, and **time-varying length**.
This work considers all — making the model **non-stationary**, realistic, and experimentally validated.

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Vertical Vibrations

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Vertical Vibrations

Type of Model Used: Lumped Mass (Centralized Mass Discretization) Model

Model Type	Description	Reference
Lumped Mass Model (Chosen)	Rope modeled as a series of **spring–damper–mass** elements — system represented as **discrete DOFs** (3 main masses).	Watanabe & Okawa – vertical vibration under braking
Distributed Mass Model	Rope treated as a **continuous flexible beam**, with variable length (complex PDEs).	Gaiko & van Horssen – flexible beam model

Reason for Choosing Lumped Model:

Simpler, accurate for short to mid-rise elevators, easier to implement in MATLAB, and allows clear vibration isolation design.

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Vertical Vibrations

3-DOF System Components

Component	Symbol	Role	Key Parameters
Elevator Car	x_1	Main body (passenger cabin)	m_c, k_c, C_c
Counterweight	x_2	Balancing mass	m_w, k_w, C_w
Traction Wheel	x_3	Rotating pulley transmitting torque	$J_{Tr}, R_{Tr}, M(t)$

Each component connected by **rope stiffness** and **damping**, forming a **3-DOF vibration system.**

04 Vertical Vibrations

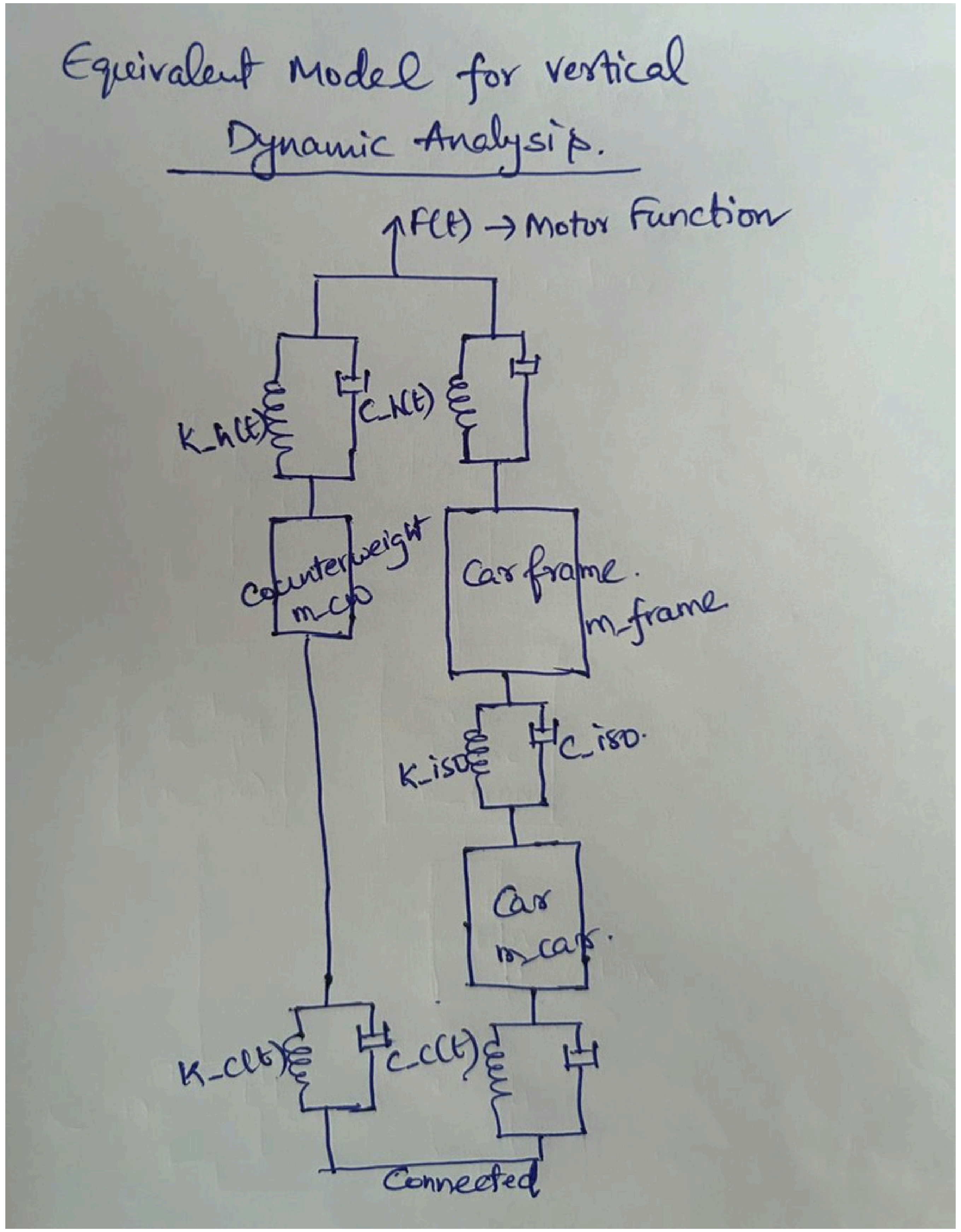
System Assumptions

- Rope always in tension, follows Hooke's law.
- Car & counterweight → rigid bodies.
- Rope segment mass ignored in simplified form.
- Wheel and pulley are rigid.
- **Lateral vibration ignored** (only vertical).
- **System is non-stationary:** Rope length changes with car motion.

Concept of Vibration Isolation

- By modeling damping and stiffness properly, the **vibration energy** transmitted from the wheel or rope to the car can be **isolated or absorbed.**
- Helps separate the **noisy frame** (structural vibrations) from the **smooth cabin** (passenger zone).

Simplified model for Vertical Dynamic Analysis



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Vertical Vibrations

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Vertical Vibrations

MATHEMATICAL ANALYSIS

$$m_c\ddot{y}_c = F_{iso} - F_{gc}$$

$$m_f\ddot{y}_f = F_{hf}(t) - F_{iso} - F_c(t) - F_{gf}$$

$$m_w\ddot{y}_w = F_{hw}(t) + F_c(t) - F_{gw}$$

Isolation Force: $F_{iso} = k_{iso}(y_f - y_c) + c_{iso}(\dot{y}_f - \dot{y}_c)$

Hoist Force (Frame): $F_{hf}(t) = k_{hf}(t)(y_{sh} - y_f) + c_{hf}(t)(\dot{y}_{sh} - \dot{y}_f)$

Hoist Force (CWT): $F_{hw}(t) = k_{hw}(t)(y_{sh\_w} - y_w) + c_{hw}(t)(\dot{y}_{sh\_w} - \dot{y}_w)$

Compensation Force: $F_c(t) = k_c(t)(y_f - y_w) + c_c(t)(\dot{y}_f - \dot{y}_w)$

Gravity Forces: $F_{gc} = m_c g$, $F_{gf} = m_f g$, $F_{gw} = m_w g$

IMP Code Snippets For Dynamic analysis

```
%% --- 2. DEFINE SYSTEM PARAMETERS (as individual variables) ---
% --- Physical Constants ---
g = 9.81; % Acceleration due to gravity (m/s^2)

% --- Hoistway and Geometry ---
L_total = 200; % Total hoistway height (m)

% --- Mass Properties (3-DOF) ---
m_cabin = 1000; % Mass of passenger cabin + load (kg)
m_frame = 500;  % Mass of the structural car frame/sling (kg)
m_cwt = 2000;   % Mass of counterweight (kg)

% --- Rope Properties (Stiffness & Damping) ---
AE_hoist = 9e7; % Stiffness property (AE) of hoist ropes (N)
AE_comp  = 7e7; % Stiffness property (AE) of comp. ropes (N)
damping_ratio = 0.02; % Modal damping ratio (zeta) for ropes

% --- Isolation Pad Properties (The core of this model) ---
iso_freq_hz = 3.0; % Target isolation frequency (Hz)
iso_zeta = 0.4;  % Target isolation damping ratio (0.4 = 40%)

omega_iso = iso_freq_hz * (2*pi); % Convert to rad/s
k_iso = omega_iso^2 * m_cabin; % Stiffness (N/m)
c_iso = 2 * iso_zeta * omega_iso * m_cabin; % Damping (Ns/m)

fprintf('Isolation Pads: k_iso = %.0f N/m, c_iso = %.0f Ns/m\n', ...
        k_iso, c_iso);

% --- Drive System (Input Profile) ---
V_max   = 8.0;  % Max constant velocity (m/s)
A_max   = 1.0;  % Max acceleration (m/s^2)
J_max   = 0.8;  % Max jerk (m/s^3)

% --- Disturbance (Motor Ripple) ---
f_ripple = 30;   % Frequency of motor ripple (Hz)
```

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Vertical Vibrations

IMP Code Snippets For Dynamic analysis

```
%% CALCULATE TIME-VARYING PARAMETERS
% Rope Lengths with time
% ROPES ARE ATTACHED TO THE FRAME (y_frame), NOT THE CABIN (y_cabin)
L_hoist_frame = max(1e-3, L_total - y_frame); % Hoist, frame side
L_hoist_cwt   = max(1e-3, L_total - y_cwt);   % Hoist, cwt side
L_comp_frame  = max(1e-3, y_frame);           % Comp, frame side
L_comp_cwt    = max(1e-3, y_cwt);             % Comp, cwt side

% --- Time-Varying Stiffness (k = AE/L) ---
k_hf = AE_hoist / L_hoist_frame;  % Hoist, frame side
k_hcw = AE_hoist / L_hoist_cwt;   % Hoist, cwt side
k_c = 1 / ( (L_comp_frame / AE_comp) + (L_comp_cwt / AE_comp) ); % Comp.

% --- Time-Varying Damping (c = 2*zeta*sqrt(k*m)) ---
c_hf = 2*zeta*sqrt(k_hf * (m_cabin + m_frame));
c_hcw = 2*zeta*sqrt(k_hcw * m_cwt);
c_c = 2*zeta*sqrt(k_c * (m_cabin + m_frame + m_cwt) / 2);
```

```
%% CALCULATE FORCES
% Force 1: Isolation Pads (connects Cabin to Frame)
F_iso = k_iso * (y_frame - y_cabin) + c_iso * (v_frame - v_cabin);
% Force 2: Hoist Ropes (Frame)
F_hoist_frame = k_hf * (y_sheave_car - y_frame) + c_hf * (v_sheave_car - v_frame);
% Force 3: Hoist Ropes (CWT)
F_hoist_cwt = k_hcw * (y_sheave_cwt - y_cwt) + c_hcw * (v_sheave_cwt - v_cwt);
% Force 4: Compensation Ropes (connects Frame to CWT)
F_comp = k_c * (y_frame - y_cwt) + c_c * (v_frame - v_cwt);

%Force 5: Gravitational Forces
F_grav_cabin = m_cabin * g;
F_grav_frame = m_frame * g;
F_grav_cwt   = m_cwt * g;
```

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Vertical Vibrations

IMP Code Snippets For Dynamic analysis

```
%% CALCULATE ACCELERATIONS (3 EOMs)
% EOM 1: Cabin (m_cabin)
% m*a = F_iso - F_gravity
a_cabin = (1 / m_cabin) * (F_iso - F_grav_cabin);

% EOM 2: Frame (m_frame)
% m*a = F_hoist - F_iso(reaction) - F_comp - F_gravity
a_frame = (1 / m_frame) * (F_hoist_frame - F_iso - F_comp - F_grav_frame);

% EOM 3: Counterweight (m_cwt)
% m*a = F_hoist_cwt + F_comp(reaction) - F_gravity
a_cwt = (1 / m_cwt) * (F_hoist_cwt + F_comp - F_grav_cwt);
```

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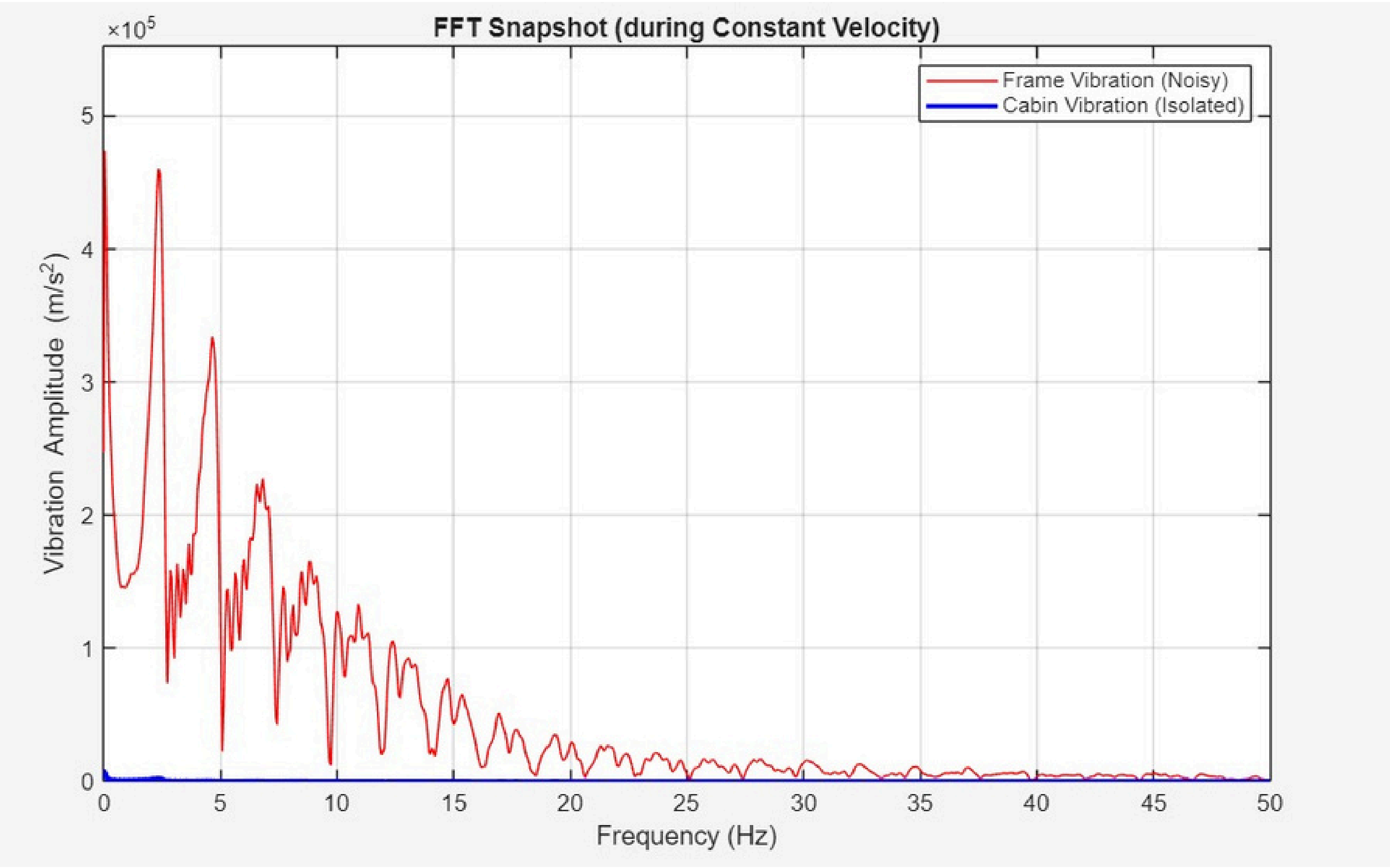
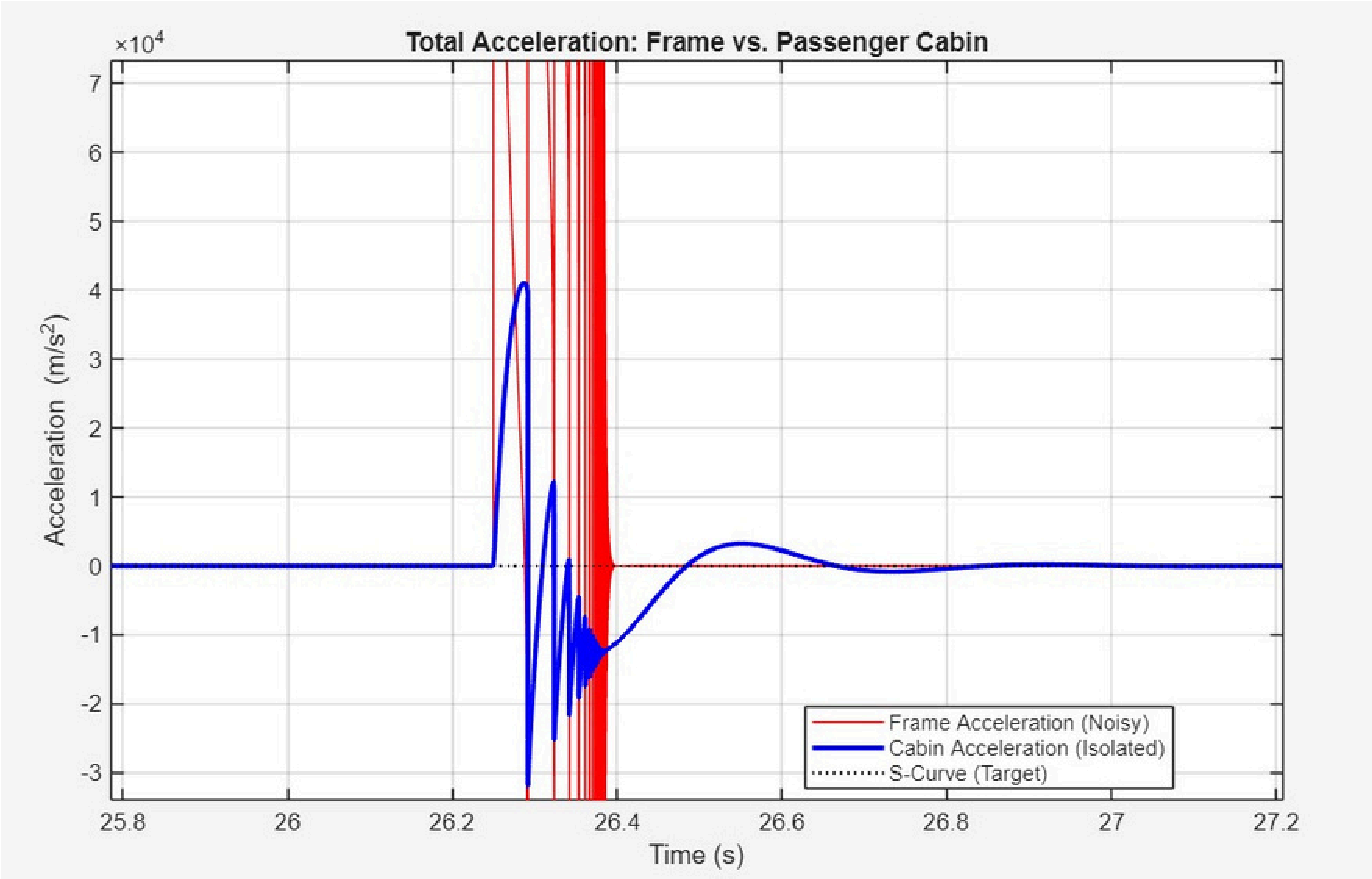
Vertical Vibrations



Vibration Frequency Plot



Total Acceleration: Frame vs. Passenger Cabin

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Vertical Vibrations

Optimisations

The optimization objective of the horizontal vibration reduction should be as follows :

- Peak-to-peak value of the horizontal vibration ac-celeration is minimized.
- Natural frequency of the system is maximized, or its opposite number is minimized

Rayleigh Damping Method:

- **Key Formula:**

$$[C] = \alpha[M] + \beta[K]$$

- [C] → Damping matrix
- [M], [K] → Mass and stiffness matrices
- α, β → Damping coefficients (to be tuned)

- Damping quantifies the **energy absorption** of the elevator system (ropes, guides, and structure).
- **Goal:** Reduce vibration amplitude by increasing the damping ratio (ζ) through proper tuning of α and β.
- Damping ratio for the i-th mode:

$$\zeta_i = \frac{\alpha + \beta\omega_{ni}^2}{2\omega_{ni}}$$

- **Optimization Strategy:**
- Tune α and β to maximize damping for the **most critical vibration mode** (highest energy mode, typically 5–7 Hz).
- This adjustment minimizes **horizontal vibration response**, especially during braking and transient conditions.
- Ensures smoother operation, improved ride comfort, and reduced structural stress.

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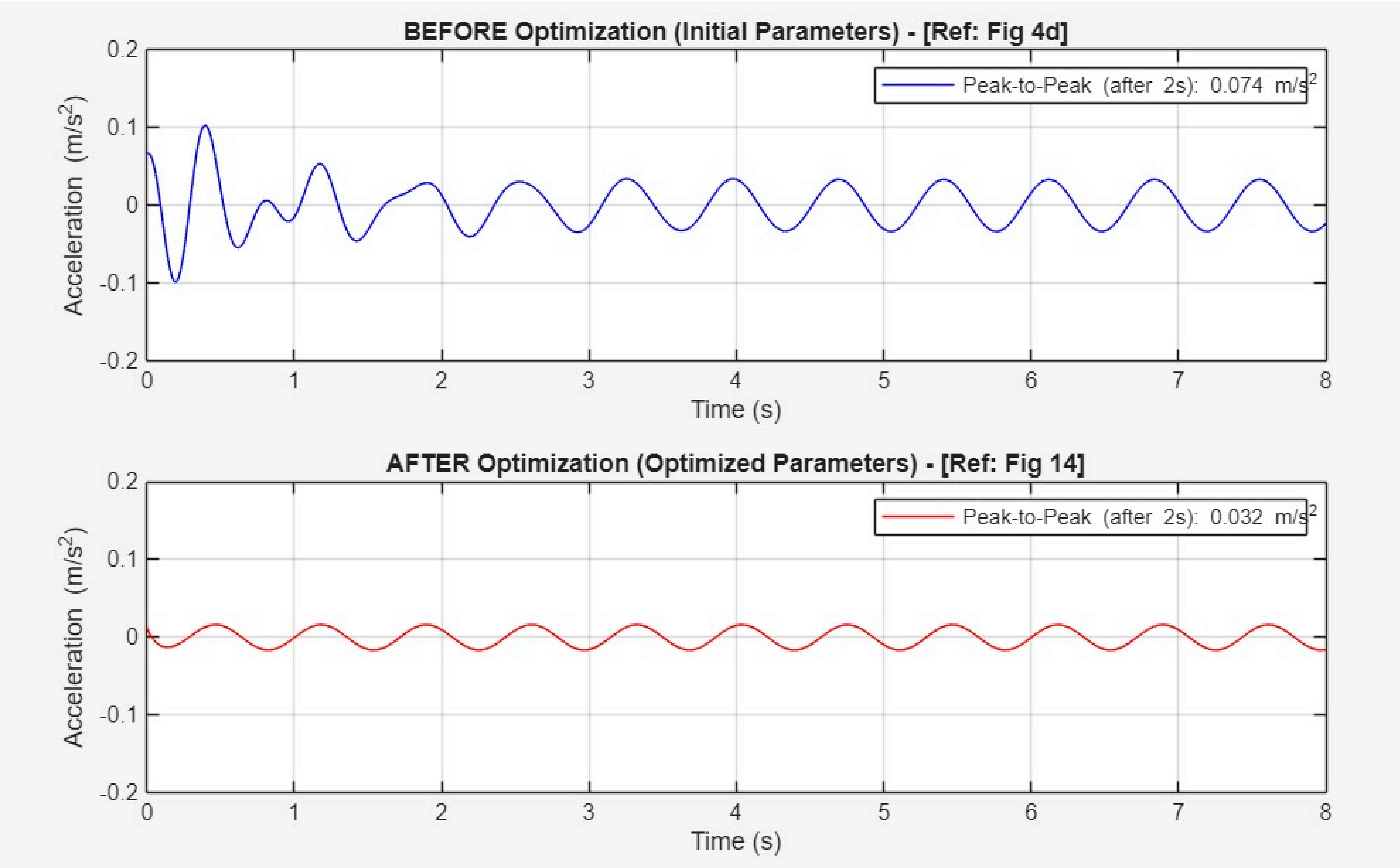
Optimisations

Optimization Result of Horizontal Vibrations



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Optimisations

References

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- https://www.researchgate.net/publication/339940862_Study_on_Theoretical_Model_and_Test_Method_of_Vertical_Vibration_of_Elevator_Traction_System
- https://www.researchgate.net/publication/352446837_Dynamic_Modelling_Experimental_Identification_and_Computer_Simulations_of_Non-Stationary_Vibration_in_High-Speed_Elevators

THANK YOU