Benha University

Benha Faculty of Engineering

Mechanical Engineering Department

Report about the following:

"Hydraulic Bascule Bridge Design & Vibro-Acoustic Mitigation Analysis"

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Chapter 1

Introduction

1.1 Project Overview

This report consolidates the design, analysis, and optimization of a Hydraulic Bas- cule Bridge system. The project is divided into two primary phases: the mechanical and hydraulic design of the bridge lifting mechanism, and a specialized analysis of vibro-acoustic phenomena in hydraulic circuits to ensure operational longevity and compliance with noise regulations.

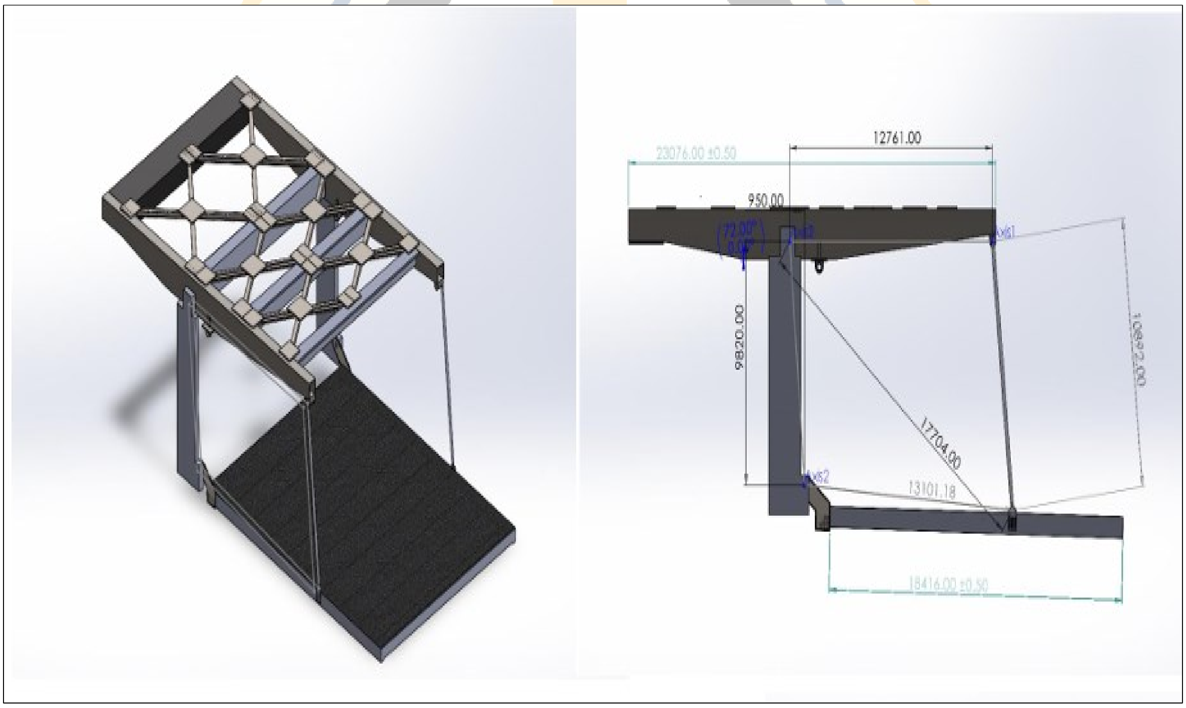


Figure 1.1: 3D CAD Model of the Hydraulic Bascule Bridge

1.2 Bascule Bridge Fundamentals

A bascule bridge, often referred to as a drawbridge or lifting bridge, is a type of movable bridge equipped with a counterweight that continuously balances a span, or ”leaf,” throughout its upward swing. This design allows for the efficient clear- ance of boat traffic.

The term ”bascule” is derived from the French term for a balance scale, utilizing the same physical principle. These bridges are the most common type of movable span due to their ability to open quickly and operate with relatively little energy input, while providing unlimited vertical clearance for marine traffic.

1.3 Scope of the Report

The report addresses the following key areas:

- Given Data & Geometric Parameters: Analysis of the bridge weight, wind loads, and operational angles.
- Hydraulic Actuator Design: Detailed calculations for the cylinder, piston, and rod to withstand the required forces.
- Circuit Design: Selection of pumps, valves, and transmission lines (pipes and hoses).
- Control System: PLC logic and I/O assignment for system automation.
- Vibro-Acoustic Analysis: A theoretical and numerical investigation into noise generation within the hydraulic circuit.
- Mitigation Strategies: Engineering solutions to reduce fluid-borne and structure- borne noise.

Chapter 2

Design Parameters and Loads

2.1 Given Data

The design is constrained by the following fixed parameters provided for the project:

1. Bridge Weight: The base weight is 156 tons. A safety factor of +20% is ap- plied for design calculations.
2. Wind Load (Closing): The wind effect while the bridge is in the closing position is 200 kg/m^2.
3. Wind Load (Opening): The wind effect during the opening operation is 40 kg/m^2.
4. Maximum Opening Angle: The bridge must open to an angle of 70°.

2.2 Operational Cases

The calculations consider four distinct operational scenarios to ensure safety un- der all conditions:

- Case A: Bridge opens with no wind load.
- Case B: Bridge opens with wind acting in the same direction (assisting), with a value of 50 kg/m^2.
- Case C: Bridge opens with wind acting in the opposite direction (resisting), with a value of 50 kg/m^2.
- Case D: Bridge is in the closed position, and the pressure in the hydraulic cylinders is zero.

2.3 Design Methodology

The calculation workflow follows this sequence:

1. Input of fixed data and material yield stresses.
2. Determination of geometric parameters.
3. Calculation of forces and reactions.
4. Stress analysis and Factor of Safety (F.S.) verification for critical components.
5. Special calculations for thermal management and auxiliary systems.

Chapter 3

Hydraulic Actuator Design

3.1 Force and Stroke Requirements

Based on the geometric analysis, the hydraulic cylinder must meet the following requirements:

- Required Force (F): 90 tons.
- Retract Speed (v): 0.5 m/min.
- Stroke Length (L): 2.3 m.
- Maximum System Pressure (P_{max}): 200 bar.

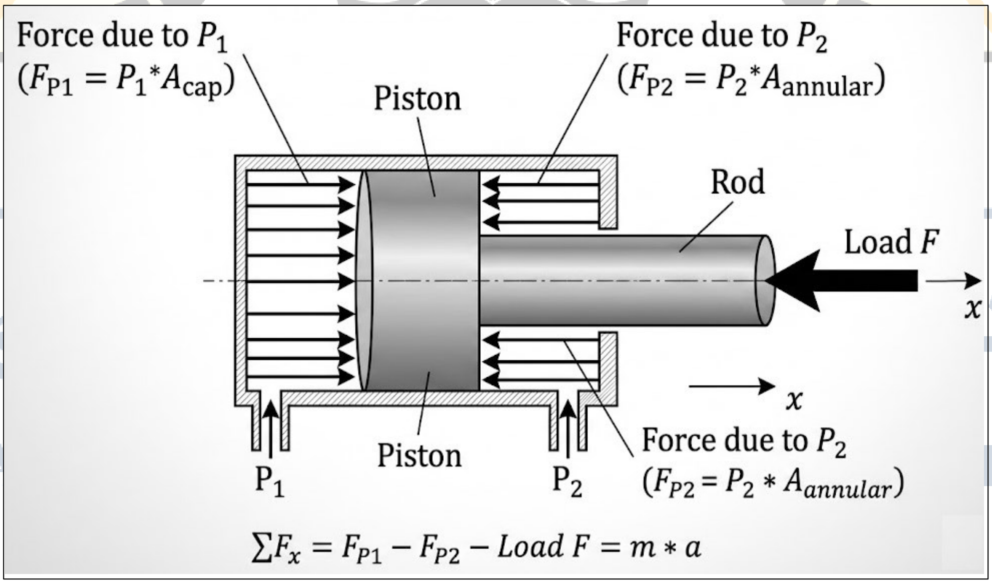


Figure 3.1: Hydraulic Cylinder Free Body Diagram

3.2 Rod Design and Buckling Analysis

The piston rod is a critical component subject to compressive loads, making buck- ling the primary failure mode. The mounting arrangement is a clevis type, pivoting at the eye-end.

3.2.1 Buckling Parameters

- Effective Buckling Length (L_k): For two ends pivoted, $L_k = L = 2.3\ m$.
- Rod Material: AISI 1045.
- Yield Strength (R_e): 530 MPa.
- Modulus of Elasticity (E): 205 GPa.
- Poisson's Ratio (ν) : 0.29

3.2.2 Slenderness Ratio Calculation

The limiting slenderness ratio (λ_g) is calculated as:

$$\lambda_g = \pi\sqrt{1.25 \times \frac{205 \times 10^9}{530 \times 10^6}} = \pi\sqrt{483.5} \approx 69.1$$

.

Assuming the rod is slender ($\lambda > \lambda_g$), we use Euler's equation to find the initial diameter:

$$F = \frac{\pi^2 EJ}{nL_k^2} \tag{3.2}$$

Where $n = 3.5$ is the safety factor. Solving for diameter d_r:

$$90 \times 10^3 \times 9.81 = \frac{\pi^2 \times 205 \times 10^9 \times \frac{\pi}{64} d_r^4}{3.5 \times (2.3)^2} \tag{3.3}$$

This yields a theoretical diameter of 113 mm. According to standard catalogs, the nearest standard size is selected.

- Initial Selection: $d_r = 125\ mm$.

3.2.3 Buckling Verification

We verify the assumption $\lambda > \lambda_g$:

$$\lambda = \frac{4L_k}{d_r} = \frac{4 \times 2.3}{0.125} = 73.6 \tag{3.4}$$

Since $73.6 > 69.1$, the rod is in the Euler region. So , safe selection diameter

$$F = \frac{\pi d_r^2 (335 - 0.62\lambda)}{4n} \tag{3.5}$$

Recalculating requires an iterative approach. To satisfy the safety factor and load requirements safely:

- Final Selected Rod Diameter (d_r): 180 mm.

3.3 Piston Design

The piston diameter (d_p) is determined by the required force and the maximum operating pressure.

$$F = P \times A_p \tag{3.6}$$

$$90 \times 10^3 \times 9.81 = P \times \frac{\pi}{4} d_p^2 \tag{3.7}$$

Using an initial catalog ratio for a rod of 125mm yielded a piston of 200mm, which required 281 bar (too high). With the revised rod diameter of 180 mm, we select a larger piston.

- Selected Piston Diameter (d_p): 280 mm.

Verification of Pressure:

$$P = \frac{F}{A_p} = \frac{90 \times 10^3 \times 9.81}{\frac{\pi}{4}(0.28)^2} = 144\ bar \tag{3.8}$$

3.4 Cylinder Wall Design

The cylinder wall thickness is calculated using the Lamé (or Bernie's) equation for thick-walled cylinders.

- Material: St52 (Yield Strength = 530 MPa).
- Design Stress (σ_d): $530/2 = 265\ MPa$.
- Internal Pressure (P_i): $200\ bar$ (Design pressure).

Thickness Calculation:

$$t = \frac{d_i}{2}\left[\sqrt{\frac{\sigma_d + (1-v)P_i}{\sigma_d - (1+v)P_i}} - 1\right] \tag{3.9}$$

$$t = 140\left[\sqrt{\frac{265+(1-0.29)20}{265-(1+0.29)20}} - 1\right]$$

$$t = 140\left[\sqrt{\frac{279.2}{239.2}} - 1\right] = 140\left[\sqrt{1.167} - 1\right] = 140[1.08-1] \approx 11.2\ mm$$

for structural rigidity and to accommodate the 316mm OD standard, we maintain a conservative thickness.
Substituting values yields $t = 11.2\ mm$.

Selected Thickness (t) = 18 mm

- Outer Diameter (d_o): $280 + 2(18) = 316\ mm$.

Hoop Stress Check:

$$\sigma_H = P\frac{d_o^2 + d_i^2}{d_o^2 - d_i^2} = 166\ MPa \tag{3.10}$$

Since $166\ MPa < 175\ MPa$, the cylinder is safe against bursting.

3.5 Summary of Actuator Dimensions

Table 3.1: Final Hydraulic Actuator Dimensions

Parameter	Value
Rod Diameter (d_r)	180 mm
Piston Diameter (d_p)	280 mm
Outer Diameter (d_o)	316 mm
Stroke Length	2300 mm
Design Pressure	200 bar

Chapter 4

Hydraulic Circuit Design

4.1 Circuit Overview

The system is powered by an electric motor driving a gear pump. The circuit in- cludes pressure relief valves, flow dividers for synchronization, and directional control valves.

4.2

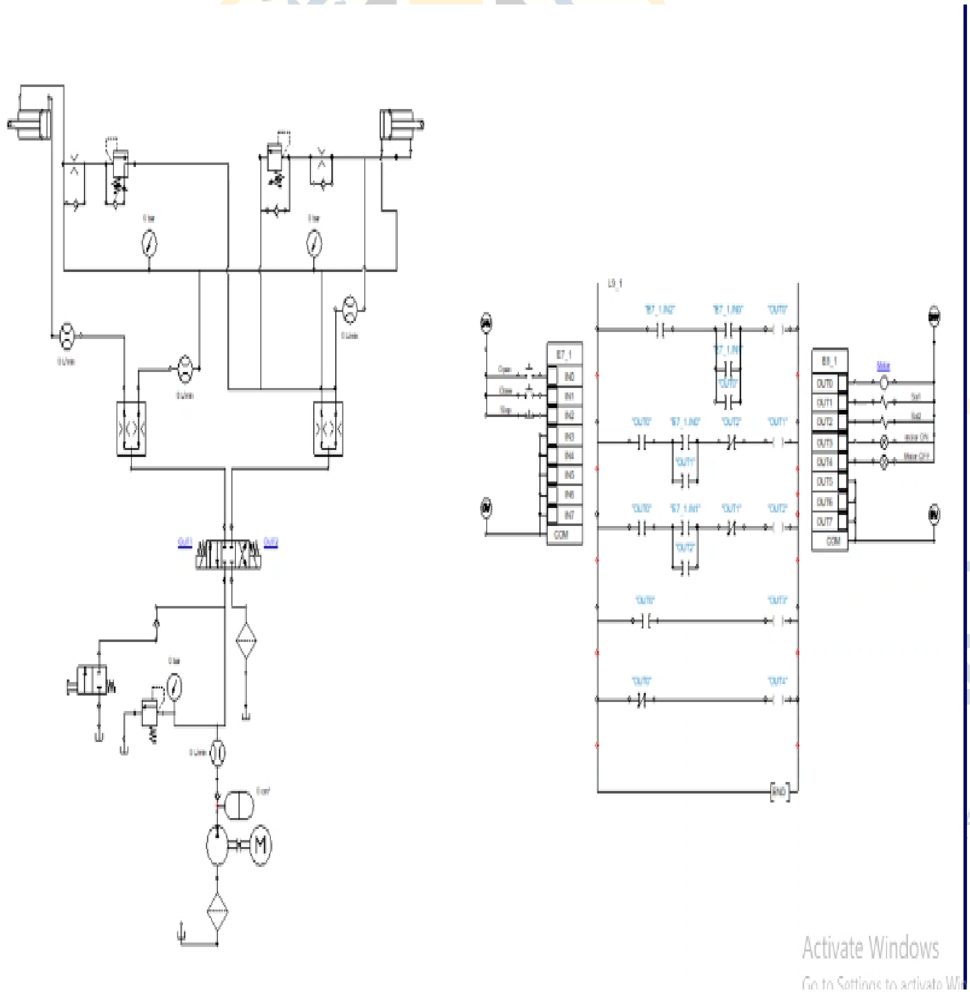


Figure 4.1: Hydraulic Circuit Diagram

Flow Rate Calculation

To achieve the required lifting speed, the flow rate (Q_1) is calculated based on the extension velocity.

- Velocity (v): 0.5 m/min.
- Piston Area (A_p): $\frac{\pi}{4}(0.28)^2 = 0.0615\ m^2$.

$$Q_1 = v \times A_p = 0.5 \times 0.0615 \times 1000 \times 60 \approx 30.78\ L/min \tag{4.1}$$

Since the bridge typically utilizes two cylinders operating in parallel:

- Total Required Flow: 62 L/min.

4.3 Transmission Line Design

The piping system connects the power unit to the actuators. Proper sizing is crucial to minimize pressure drops and prevent excessive heat generation.

4.3.1 Pressure Lines (Rigid Tubes)

- Recommended Velocity (v): $2 - 6\ m/s$.
- Maximum Flow (Q_{max}): 80 L/min (Safety margin included).

Calculating the diameter:

$$d = \sqrt{\frac{4Q_{max}}{\pi v}}$$

For $v = 6\ m/s$, $d_{min} = 16.8\ mm$. For $v = 2\ m/s$, $d_{max} = 29.13\ mm$. We select a standard metric tube size within this range.

- Selected Tube: $25 \times 4\ mm$ (ID = 25mm, OD = 33mm).
- Material: Carbon Steel.

4.3.2 Return Lines

Return lines require lower velocities ($0.6 - 1.6\ m/s$) to prevent backpressure.

- Calculated Range: $32.57\ mm$ to $53\ mm$.
- Selected Tube: $50 \times 3\ mm$ (ID = 50mm).

4.4 Hose Selection

Flexible hoses are required at moving interfaces.

- Pressure Line Hose: 1 inch (25.4 mm ID). Outer Diameter 39.7 mm.
- Return Line Hose: 2 inch (50.8 mm ID).

4.5 Pressure Drop Analysis

Friction losses were calculated using the standard laminar flow equation for hy- draulic oil (Viscosity $v = 9.7\ mm^2/s$ at $40^\circ C$).

$$\Delta P = 128 \mu Q L / \pi D^4$$

Result:

- Pressure Line Drop: $1152\ Pa/m$.
- Return Line Drop: $72\ Pa/m$.
- Total System Pressure Drop: Approx $0.57\ bar$.

Chapter 5 Component Selection

Pump Selection

The system requires a robust pump capable of delivering at least 62 L/min at pres- sures exceeding 160 bar.

Selected Unit: Marzocchi External Gear Pump

- Model: ALP3A-D-94.
- Displacement: 61 cm^3/rev.
- Flow at 1500 rpm: 87 L/min.
- Max Pressure (P_1): 190 bar.

Peak Pressure (P_2): 205 bar.

ALP3AD94	61	87	190	205	220	2800	79	158,5	30,2	58,7	7/16	33	26,19	52,37	27

Figure 5.1: Marzocchi Pump Dimensional Drawing

5.1 Valve Selection

Components were selected from the Rexroth catalog to match the system flow and pressure requirements.

- Relief Valve: Pilot operated, Type DB10. Max Pressure 350 bar.
- Check Valve: Type SL15. Cracking pressure 5 bar.
- Directional Control Valve: 4/3 Spool Valve, Type WE10. Solenoid actuated.
- Flow Divider: Spool type L06A3 to synchronize the two cylinders.

5.2 Tank Design and Thermal Analysis

The reservoir must be sized to handle the fluid volume and dissipate heat gener- ated by system inefficiencies.

5.3 Heat Generation

- Input Mechanical Power: 19 kW.
- System Efficiency: 77%.
- Heat Load (N_{heat}): $(1 - 0.77) \times 19 = 4.37\ kW$.

5.3.2 Heat Dissipation Calculation

Assuming a square tank section with side a and length $2a$. The total heat dissipa- tion capability (H_t) is the sum of horizontal and vertical surface radiation.

$$H_t = 3420 \times a^{1.75}\ Watts \tag{5.1}$$

Equating heat generation to dissipation:

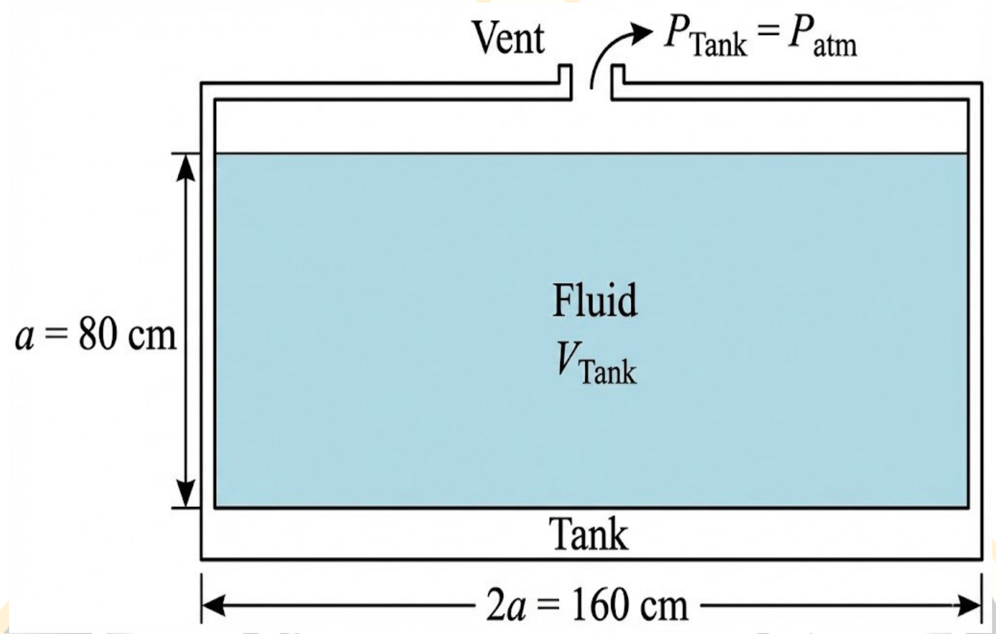
$$4370 = 6840 \times a^{1.75} \tag{5.2}$$

Solving for a yields approx 77.41 cm.

5.1.1 Tank Dimensions

To ensure adequate cooling and volume:

- Dimensions: 80 $cm \times 80\ cm \times 160\ cm$.
- Total Capacity: 1024 Liters.
- This capacity provides sufficient residence time for de-aeration and cooling.



Hydraulic Reservoir Heat Dissipation Geometry

Chapter 6

Control System

6.1 PLC Control Strategy

To ensure safe and synchronized operation of the bascule bridge, a Programmable Logic Controller (PLC) is employed. The control system monitors the position sen- sors, pressure switches, and operator inputs to manage the solenoid valves and motor starters.

6.2 PLC Input/Output Assignment

The following table outlines the I/O address assignment for the bridge control system.

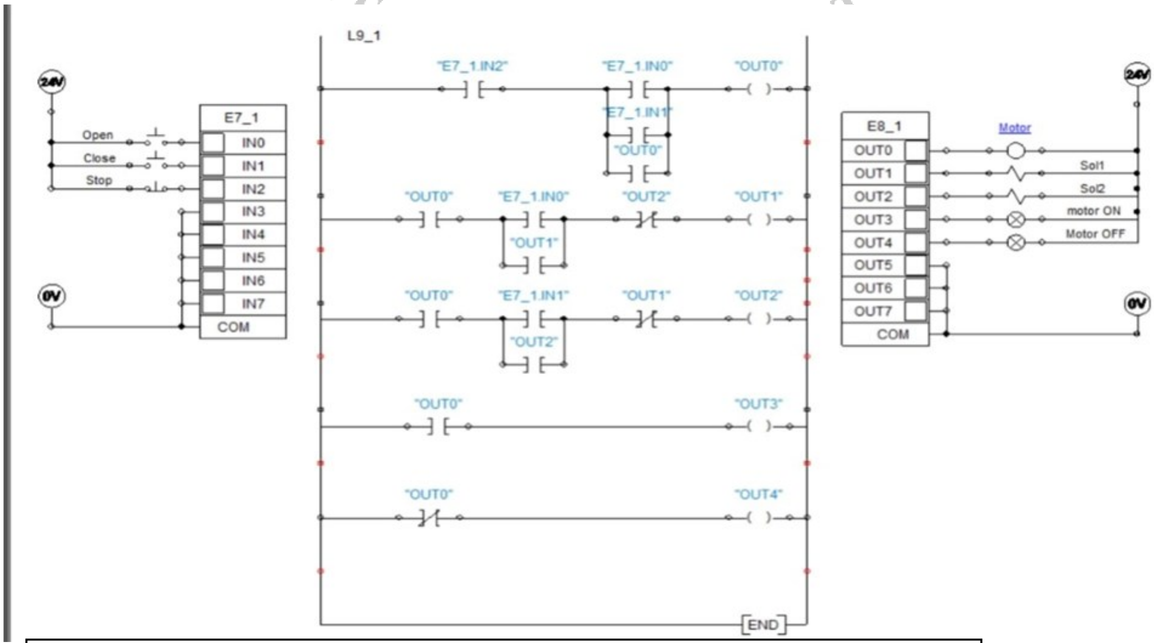


Table 6.1: PLC Input/Output Assignment List

Bridge Control System - I/O Address Assignment Table

Address	Tag Name	Description	Device / Location	Signal Type / Voltage
Discrete Inputs (DI) - RACK 0 - SLOT 1 (Main Control Panel Inputs)				
%IX0.0	PB_ESTOP_MCR	Master Control Relay E-Stop Pushbutton (NC)	Main Console	24VDC (NC)
%IX0.1	SW_AUTO_MAN	Mode Selector Switch (0=Manual, 1=Auto)	Main Console	Selector
%IX0.2	PB_BRIDGE_LIFT	Pushbutton - Initiate Bridge Lift Sequence	Main Console	NO Momentary
%IX0.3	PB_BRIDGE_LOWER	Pushbutton - Initiate Bridge Lower Sequence	Main Console	NO Momentary
%IX0.4	PB_GATES_CLOSE	Pushbutton - Close Traffic Gates	Main Console	NO Momentary
%IX0.5	PB_GATES_OPEN	Pushbutton - Open Traffic Gates	Main Console	NO Momentary
%IX0.6	PB_ALARM_ACK	Pushbutton - Alarm Acknowledge/Silence	Main Console	NO Momentary
Discrete Inputs (DI) - RACK 0 - SLOT 2 (Field Sensors - Bridge Structure)				
%IX1.0	LS_BRIDGE_CLOSED	Limit Switch - Bridge Fully Seated/Closed	Bridge Toe/Lock	N.O. Limit
%IX1.1	LS_BRIDGE_OPEN	Limit Switch - Bridge Fully Raised/Open	Bridge Trunnion	N.O. Limit
%IX1.2	LS_BRIDGE_NEAR_CL	Limit Switch - Near Closed (Deceleration point)	Bridge Structure	N.O. Proximity
%IX1.3	LS_BRIDGE_NEAR_OP	Limit Switch - Near Open (Deceleration point)	Bridge Structure	N.O. Proximity
%IX1.4	PS_HYD_OK	Hydraulic System Pressure OK switch	Hydraulic Unit	Pressure Sw.
%IX1.5	LS_LOCK_ENGAGED	Span Lock Engaged (Bridge secured down)	Span Lock Mech	N.O. Limit
%IX1.6	LS_LOCK_DISENG	Span Lock Disengaged (Ready to lift)	Span Lock Mech	N.O. Limit
Discrete Inputs (DI) - RACK 0 - SLOT 3 (Field Sensors - Traffic & Gates)				
%IX2.0	LS_GATE1_CLSD	Traffic Gate 1 (Near Side) Is Closed	Gate Housing 1	N.O. Limit
%IX2.1	LS_GATE1_OPEN	Traffic Gate 1 (Near Side) Is Open	Gate Housing 1	N.O. Limit
%IX2.2	LS_GATE2_CLSD	Traffic Gate 2 (Far Side) Is Closed	Gate Housing 2	N.O. Limit
%IX2.3	LS_GATE2_OPEN	Traffic Gate 2 (Far Side) Is Open	Gate Housing 2	N.O. Limit
%IX2.4	PE_TRAFFIC_CLR	Photoeye/Loop - Deck Clear of traffic Zone	Bridge Deck	N.O. Contact
%IX2.5	PE_PED_DETECT	Pedestrian Detector - Sidewalk clear zone	Sidewalk Approach	N.O. Contact

Chapter 7

Vibro-Acoustic Phenomena in Hydraulic Systems

7.1 Introduction

The second phase of this project focuses on the ”mitigation of the project,” specif- ically addressing the vibro-acoustic byproducts of high-pressure hydraulic opera- tions. Hydraulic systems are prized for power density but are notorious for gen- erating structure-borne (SBN) and airborne noise (ABN).

7.2 The Physics of Noise Generation

Noise in hydraulic systems is not random; it follows a specific causal chain:

1. Source (Fluid-Borne Noise): The pressure ripple generated by the pump.
2. Transmission (Structure-Borne Noise): The ripple excites the pipes and pump housing.
3. Radiation (Airborne Noise): Vibrating surfaces couple with the air to pro- duce audible sound.

The dominant source is the Pressure Ripple. As gear teeth mesh, they separate high-pressure zones from low-pressure zones. The discrete nature of this process results in a non-constant flow rate, known as flow ripple.

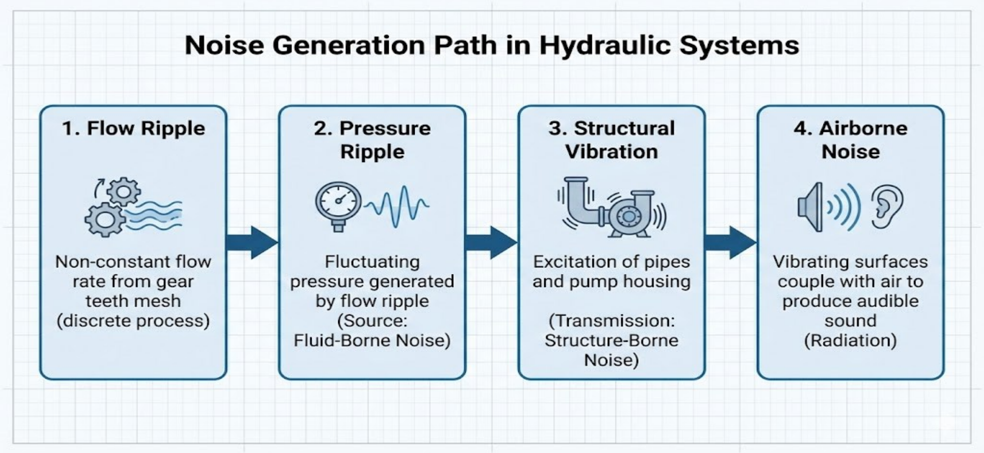


Figure 7.1: Causal Chain of Hydraulic Noise Generation

Chapter 8

System Kinematics and Source Characterization

8.1 Gear Pump Kinematics

To predict the noise spectrum, we calculate the fundamental frequencies derived from the pump's mechanics.

8.1.1 Geometric Displacement

$$V_{dis} = \frac{Q_{nom}}{N} = 1033.54/25=41.34 \text{ CC/rev} \qquad (8.1)$$

This displacement corresponds to a standard Group 2 pump with approximately $z = 12$ teeth.

8.1.2 Spectral Frequency Calculation

The noise spectrum is dominated by the shaft frequency and the gear mesh fre- quency.

- Shaft Frequency (f_{shaft}):

$$f_{shaft} = \frac{N}{60} = \frac{1500}{60} = 25\ Hz \qquad (8.2)$$

Gear Mesh Frequency (f_{mesh}):

$$f_{mesh} = f_{shaft} \times z = 25 \times 12 = 300\ Hz \qquad (8.3)$$

The fundamental noise tone will occur at 300 Hz, with harmonics at 600 Hz, 900 Hz, etc.

8.1 Flow Ripple Magnitude

The flow ripple is the ”forcing function” of the vibration. For spur gears, this ripple is typically 10-15% of the mean flow.

- Peak-to-Peak Flow Ripple:

$$\Delta Q_{p-p} = 0.12 \times 62 = 7.44\ L/min \tag{8.4}$$

This fluctuating flow interacts with the circuit impedance to create pressure waves that travel through the steel pipes.

Chapter 9

Fluid Dynamics and Structural Resonance

9.1 Pipe Flow and Turbulence

Using the project's standard 25mm ID hydraulic tube, we analyzed the flow regime.

9.1.1 Velocity Calculation

Using the maximum pump flow (Q=87 L/min) to determine peak velocity:

$$v = \frac{Q}{A} = \frac{1.45 \times 10^{-3}}{\pi (0.0125)^2} \approx 2.95\ m/s$$

9.1.1 Reynolds Number

Using ISO VG 46 oil at 50◦C (ν =30 cSt):

$$Re = \frac{vD}{\nu} = \frac{2.95 \times 0.025}{30 \times 10^{-6}} \approx 2458$$

Since Re > 2300, the flow is Transitional to Turbulent., the larger 25mm pipe diameter promotes turbulence. This implies that flow noise in the piping itself will be a measurable contributor to the broadband noise spectrum, in addition to valve throttling

Cavitation Risk Analysis

The pressure line is 25mm. To prevent cavitation at the pump inlet with 87 L/min flow, the suction velocity must be < 1.0 m/s

$$D_{req} = \sqrt{\frac{4Q}{\pi v_{target}}} = \sqrt{\frac{4 \times 1.45 \times 10^{-3}}{\pi \times 1.0}} \approx 43\ mm$$

Recommendation: The suction line must be sized to match the return line at **50mm ID**. A **25mm suction line would result in 3 m/s inlet velocity**, guaranteeing cavitation failure

9.2 Piping Resonance (Fluid-Structure Interaction)

The interaction between the 300 Hz pressure ripple and the pipe's natural fre- quency is a critical failure mode. We modeled the pipe as a continuous beam.

9.2.1 Natural Frequency Calculation

- Mass and Inertia Calculation

For a steel pipe (33mm OD, 25mm ID) filled with oil:

Area of Steel: $A_s = \frac{\pi}{4}(0.033^2 - 0.025^2) \approx 3.64 \times 10^{-4}\ m^2$ Mass of Steel: $\mu_{steel} = 7850 \times A_s \approx 2.86\ kg/m$ Mass of Oil: $\mu_{oil} = 870 \times \frac{\pi}{4}(0.025)^2 \approx 0.43\ kg/m$ **Total Distributed Mass (μ):** $3.29\ kg/m$

- Total mass per meter: 3.29 kg/m.

Moment of Inertia (I):

$$I = \frac{\pi}{64}(D_o^4 - D_i^4) = \frac{\pi}{64}(0.033^4 - 0.025^4) \approx 3.9 \times 10^{-8}\ m^4$$

Calculating the stiffness/mass factor: $\sqrt{EI/\mu} \approx 49.9$ | $f_1 = \frac{22.37}{2\pi L^2}\sqrt{\frac{EI}{\mu}}$

$$L_{crit} = \sqrt{\frac{3.56 \times 49.9}{300}} \approx 0.77\ m$$

Analysis: Clamping supports must avoid spacing near **0.75m-0.80m** to prevent direct resonance with the pump mesh frequency .A **spacing of 1.0m** is actually structurally superior for this larger pipe size compared to the **0.5m recommendation for smaller tubes**.

Chapter 10

Mitigation Strategies and Engineering Solutions

Based on the theoretical analysis, five distinct mitigation strategies are proposed and evaluated in detail.

10.1 Mitigation A: Vibration Isolation Mounts

Definition:-

Vibration Isolation Mounts are mechanical supports made from elastomeric materials (like neoprene, silicone rubber, or other elastomers) placed between the vibrating equipment (here, a hydraulic pump) and the supporting structure (chassis, base, or foundation).

_They act as spring-damper systems, meaning they:-

- Provide elasticity (stiffness) to carry the load and allow controlled motion.
- Provide damping to dissipate energy and reduce vibration amplitude.

Purpose in hydraulic systems

_Hydraulic pumps often generate high-frequency vibrations, caused by:

- Gear mesh (here 12 teeth × 1500 rpm → 300 Hz)
- Pressure pulsations in the fluid
- Mechanical imbalances

_Without isolation, these vibrations transfer directly to the chassis and piping system, causing:

- Noise
- Fatigue in pump mounts, baseplates, and piping
- Loosening of bolts and fittings
- Unwanted dynamic forces in the hydraulic system

_Vibration isolation mounts are used to:

- Decouple the pump from the supporting structure
- Shift the natural frequency of the pump-mount system below the dominant excitation frequency

  ($f_n \ll f_{\text{gear}}$)
- Reduce vibration transmission to the chassis and the connected hydraulic system

Effect on the system

_Before Isolation (Rigid Mounting)

- Pump directly mounted on a stiff base
- System natural frequency is high, often near or above excitation frequency → risk of resonance
- Vibrations are fully transmitted to the chassis and piping
- Hydraulic system sees pressure spikes due to vibrations transmitted through pump mounting
- Components experience higher stress and fatigue

After Isolation (Elastomeric Mounts)

- Pump is supported on spring-damper isolators
- System natural frequency is lower than dominant excitation:

$$f_n \ll f_{\text{excitation}} \Rightarrow \text{effective isolation}$$

- High-frequency vibrations are absorbed by the mounts
- Transmission of vibration to the chassis/piping is significantly reduced

Hydraulic system benefits:-

- Smoother fluid pressure → less pulsation in lines and valves
- Reduced risk of leakage or fatigue in hydraulic connections
- Noise reduction
- Longer life for pump, piping, and support structure

Calc:-

We'll still assume:-

- 4 springs in parallel
- 4 dampers in parallel
- excitation from pump gear mesh:

$$f = \frac{1500 \times 12}{60} = 300 \text{ Hz}$$

$$f_n \ll 300 \text{ Hz}$$

$$f_n \approx \frac{300}{3} = 100 \text{ Hz}$$

Natural frequency relation

$$\omega_n = 2\pi f_n$$
$$\omega_n = 2\pi(100) = 628 \text{ rad/s}$$
$$\omega_n^2 = \frac{k}{m}$$

Solve for stiffness:

$$k = m\omega_n^2 = 30(628)^2 \approx 11.8 \times 10^6 \text{ N/m}$$

Total stiffness:

$$k \approx 1.18 \times 10^7 \text{ N/m}$$

Since we have **4 springs**:

$$k_s = \frac{k}{4} \approx 2.95 \times 10^6 \text{ N/m (per spring)}$$

Critical damping
damping

$$c_{cr} = 2\sqrt{mk}$$
$$c_{cr} = 2\sqrt{(30)(1.18 \times 10^7)}$$
$$c_{cr} \approx 37\,600 \text{ N\m}$$

So the **system needs > 37.6 kN/m** to become overdamped.
Choose damping to make system OVERDAMPED
Take damping ratio:

$$\zeta = \frac{c}{c_{cr}}$$

Overdamped condition:

$$\zeta > 1$$

Let us choose:

$$\zeta = 1.5$$

Then:

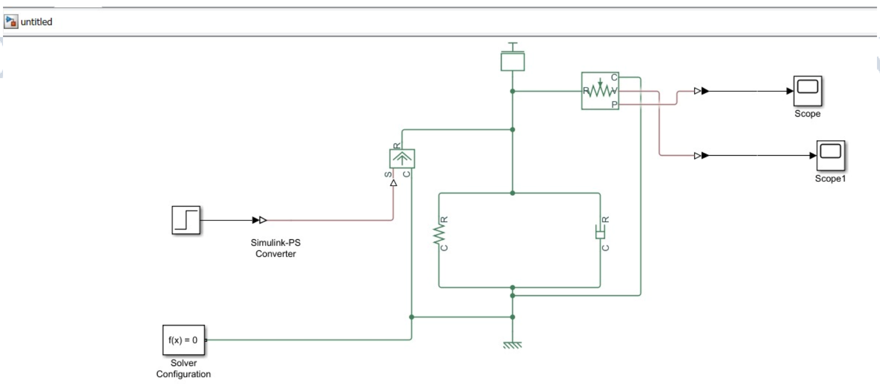
$$c = 1.5\, c_{cr} = 1.5(37\,600) \approx 56\,400 \text{ N\m}$$

Because we have **4 dampers in parallel**:

$$c_s = \frac{c}{4} \approx 14\,100 \text{ N\m (per damper)}$$

✓ exact values chosen:

- Total damping: **≈ 56.4 kN/m**
- Each damper: **≈ 14.1 kN/m**
- Damping ratio: **ζ = 1.5 → OVERDAMPED**



10.1 Mitigation B: Helical Gear Pump Design

While the acoustic analysis was initially performed on a baseline 62 L/min model, the following design is specifically sized for the actual project pump (Marzocchi ALP3A-D-94, 61 cc/rev) to ensure the mitigation strategy is valid for the final bridge operation.

10.1.1 Mechanism of Improvement

Standard spur gears trap fluid in the root volume, causing pressure spikes. Helical gears introduce a helix angle (β), allowing for gradual tooth engagement.

- Contact Ratio (ϵ_γ): Increases from approx. 1.5 (spur) to > 2.0 (helical).
- Flow Ripple Reduction: Theoretical reduction of 60-70% in ripple amplitude due to the continuous contact line.

10.1.2 Geometric Sizing and Calculation

To replace the spur gears in the 61 cc/rev pump, we calculate the required module and face width using the displacement formula:

$$V_{disp} = 2\pi z m^2 b / \cos^2 \beta$$

Input Parameters:

- Required Displacement: $V = 61,000\ mm^3/rev$.
- Number of Teeth (z): 12.
- Helix Angle (β): 15°.
- Normal Pressure Angle (α_n): 20°.

Module Selection: Assuming a standard normal module $m_n = 4.5\ mm$:

$$D_p = \frac{zm_n}{\cos\beta} = \frac{12 \times 4.5}{\cos(15^\circ)} = \frac{54}{0.9659} \approx 55.9\ mm \tag{10.4}$$

Face Width Calculation: Solving the displacement equation for width b:

$$b \approx \frac{V_{disp}}{2\pi z m_n^2} = \frac{61000}{2\pi(12)(4.5)^2} \approx 40\ mm \tag{10.5}$$

10.1 Contact Ratio Verification

To ensure noise reduction, the total contact ratio must exceed 2.0.

- Overlap Ratio (ϵ_β):

$$\epsilon_\gamma = \epsilon_\alpha + \epsilon_\beta = 1.6 + 0.73 = 2.33 \tag{10.7}$$

Since 2.33 > 2.0, the design will significantly smooth the flow ripple.

10.1.1 Force Analysis and Thrust Load

Helical gears generate axial thrust which must be supported by the housing.

Operating Torque (T): At 200 bar (20 MPa):

$$T = \frac{\Delta P \cdot V}{2\pi\eta_{mech}} = \frac{20 \times 10^6 \times 61 \times 10^{-6}}{2\pi \times 0.9} \approx 216\ Nm \tag{10.8}$$

- Tangential Force (F_t):

$$F_t = \frac{2T}{D_p} = \frac{2 \times 216}{0.0559} \approx 7728\,N \qquad (10.9)$$

- Axial Thrust (F_a):

$$F_a = F_t \tan\beta = 7728 \times \tan(15^\circ) \approx 2070\,N \qquad (10.10)$$

10.1.1 Final Gear Specification

Table 10.1: Proposed Helical Gear Specifications

Parameter	Value
Normal Module (m_n)	4.5 mm
Number of Teeth (z)	12
Helix Angle (β)	15°
Face Width (b)	40 mm
Pitch Diameter (D_p)	55.9 mm
Material	Case-Hardened Steel (20MnCr5)
Required Thrust Bearing Capacity	$> 2100\,N$

10.1 Mitigation C: Acoustic Enclosure

10.2 Frequency Limitations of Foam

Porous absorbers (foam) function by converting acoustic particle velocity into heat. They are only effective when the thickness is roughly $\lambda/4$. At 300 Hz, the wave- length of sound in air is $\lambda \approx 114m$.

$$t_{req} = \frac{\lambda}{4} = \frac{1.14}{4} \approx 0.28m = 28cm \qquad (10.11)$$

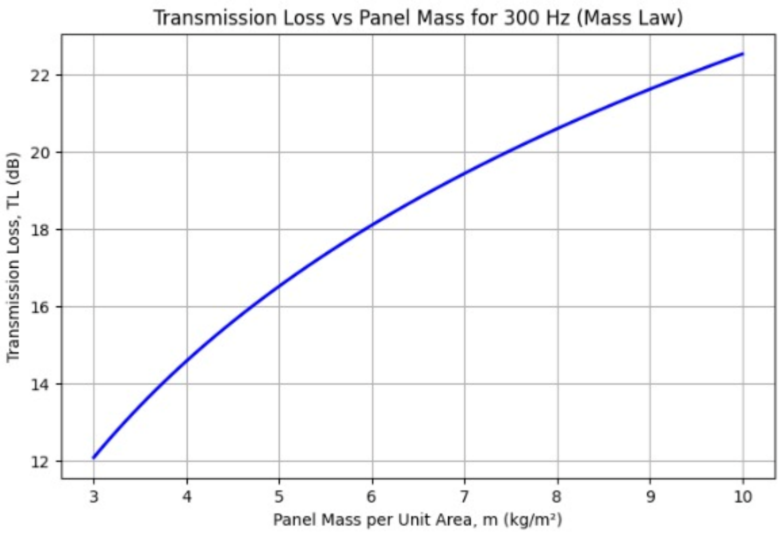
A 28cm foam lining is impractical. Therefore, foam alone cannot stop the funda- mental gear tone.

10.1.1 Mass Law Implementation

To block 300 Hz noise, the enclosure must rely on the Mass Law:

$$TL \approx 20\log(f \cdot m) - 47 \qquad (10.12)$$

To achieve a target Transmission Loss (TL) of 20 dB, the surface density (m) must be high. Specification: The enclosure must be constructed from a dense bar- rier material, such as 1.5mm steel sheet ($m \approx 10kg/m^2$), lined with thinner foam (25mm) solely to absorb high-frequency valve hiss ($> 1kHz$) and reduce internal reverberation.



10.1 Mitigation D: Pulsation Dampers

10.2 Sizing Logic

The accumulator must absorb the volume of fluid displaced per tooth passage to smooth the ripple

Using Polytropic expansion ($PV^n = C$) for nitrogen gas, and targeting a pressure fluctuation of $< 1\%$,. Expanding to pre-charge conditions (approx 80% of system pressure):-

ere is the **same derivation rewritten cleanly**, keeping everything the same but explicitly using

$$P_0 = 0.8\, P_{sys}$$

instead of numeric wording.

---

1. Displacement per tooth

$$V_{\text{tooth}} = \frac{V_{dis}}{z} = \frac{61}{12} \approx 5.08 \text{ cc}$$

---

2. Sizing using the Polytropic Law

Gas compression follows the polytropic relation:

$$PV^n = C$$

For fast pressure pulsations, the process is **adiabatic**, so:

$$n = 1.4$$

Differentiating for small fluctuations:

$$\frac{\Delta P}{P} = -n\frac{\Delta V}{V_{gas}}$$

Taking magnitudes:

$$\frac{\Delta P}{P} = n\frac{\Delta V}{V_{gas}}$$

Definitions

- $\Delta P/P$: allowable pressure ripple ratio
  Target:

$$\frac{\Delta P}{P} \leq 2.5\% = 0.025$$

- ΔV: volume to be absorbed

$$\Delta V \approx V_{\text{tooth}} = 5.08 \text{ cc}$$

- V_{gas}: required gas volume at **system pressure**

---

3. Required gas volume at operating pressure

Rearranging:

$$V_{gas} = \frac{n\,\Delta V}{\Delta P/P}$$

Substitute values:

$$V_{gas} = \frac{1.4 \times 5.08}{0.025}$$
$$V_{gas} \approx 284.5 \text{ cc}$$

---

4. Precharge pressure definition

Let system pressure be:

$$P_{sys} = 166 \text{ bar}$$

Precharge pressure defined as:

$$P_0 = 0.8\,P_{sys}$$
$$P_0 = 0.8 \times 166 = 132.8 \approx 133 \text{ bar}$$

---

5. Total accumulator (nominal) volume

Using the **isothermal gas law** for sizing:

$$P_0 V_0 = P_{sys} V_{gas}$$

Solve for accumulator volume V_0:

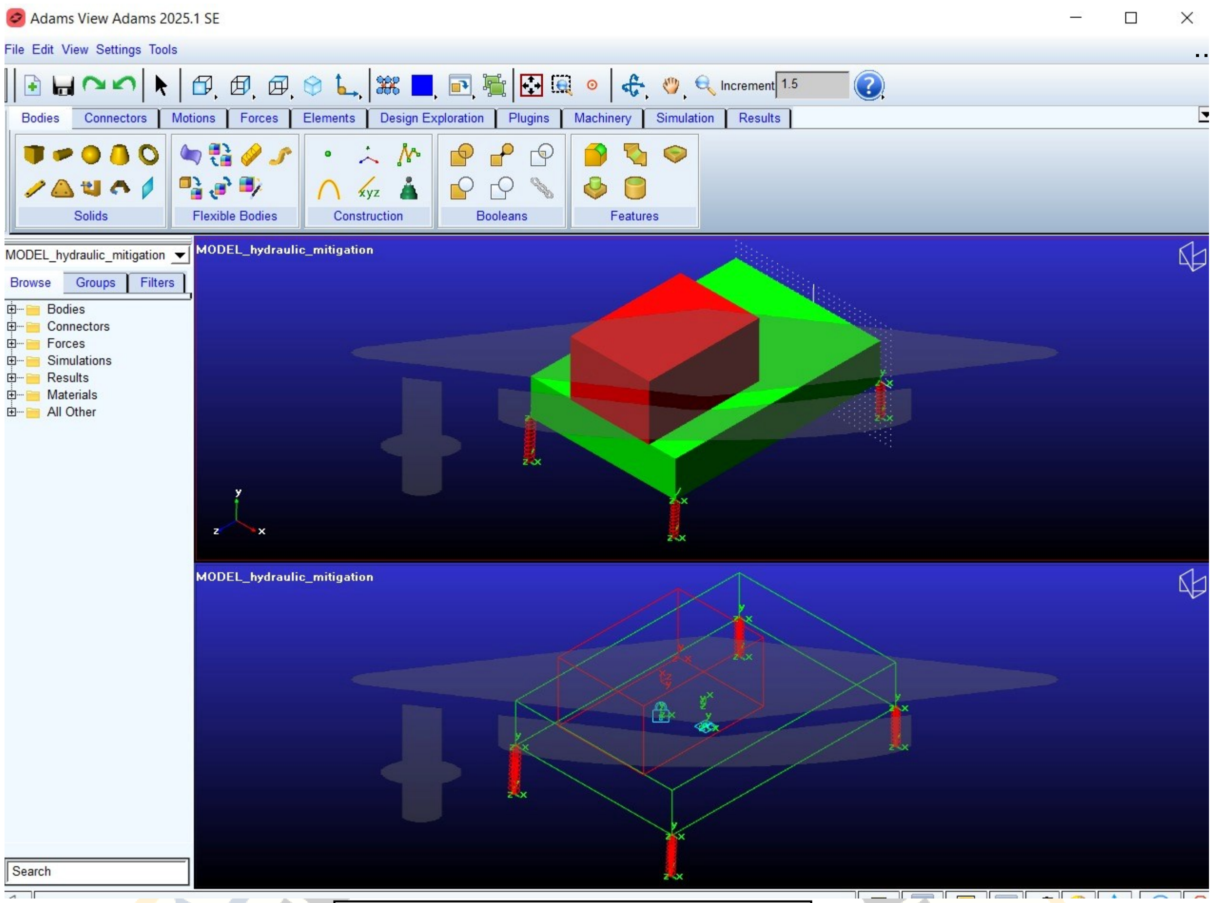
$$V_0 = V_{gas}\frac{P_{sys}}{P_0}$$

Substitute:

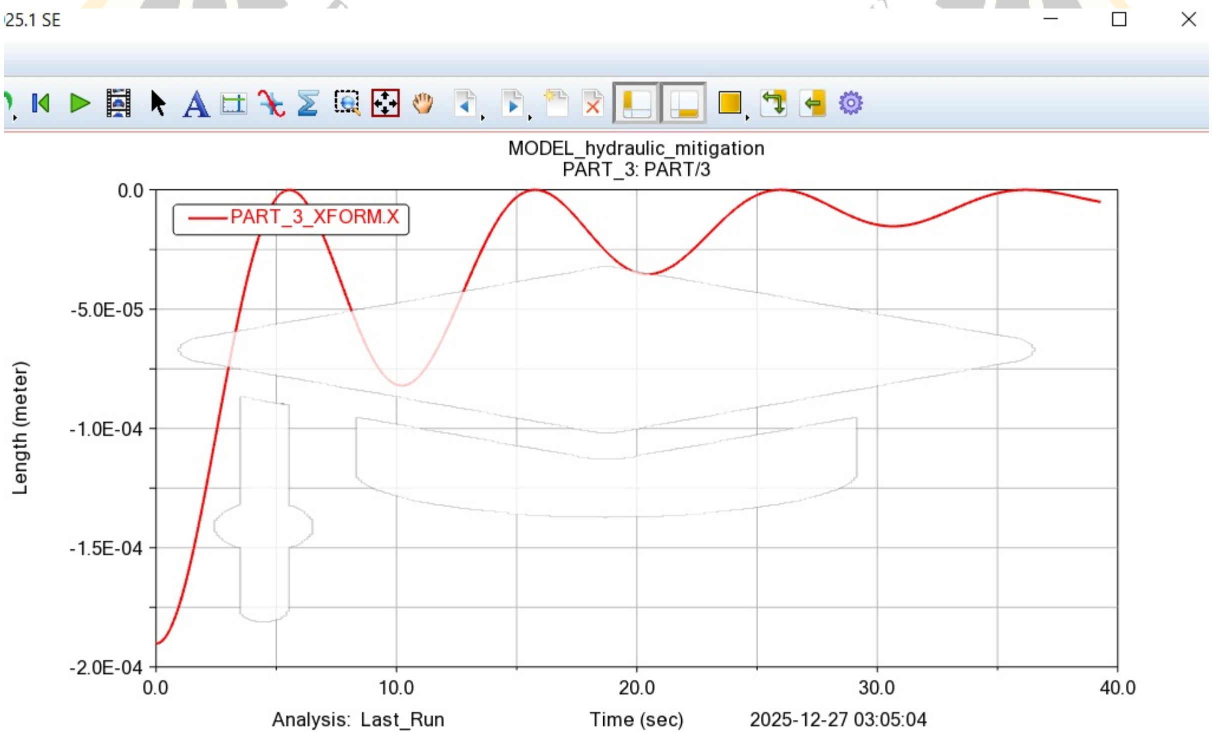
$$V_0 = 284.5 \times \frac{166}{0.8 \times 166}$$

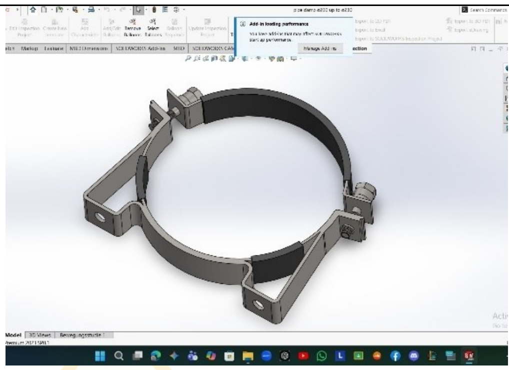
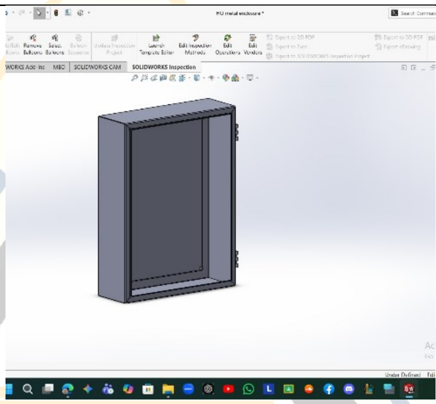
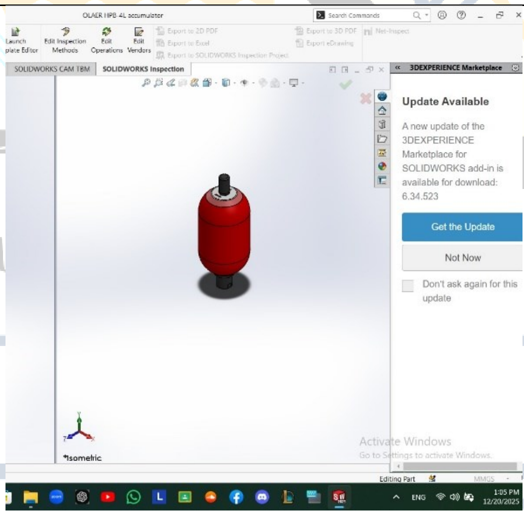
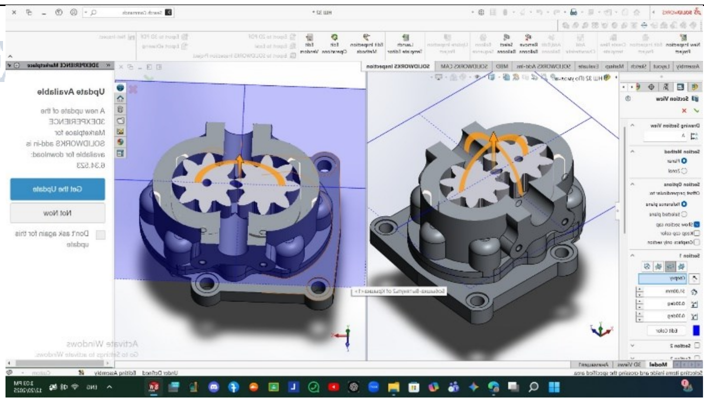
Cancel 166:

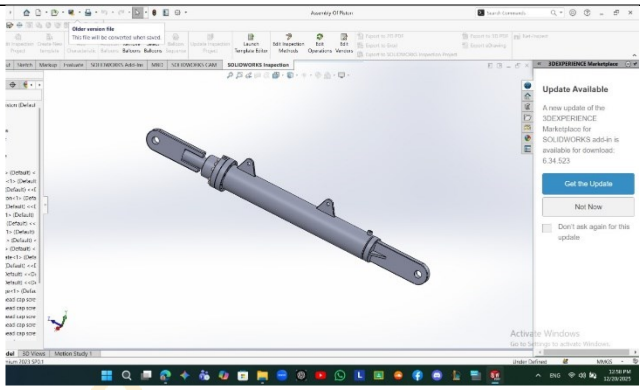
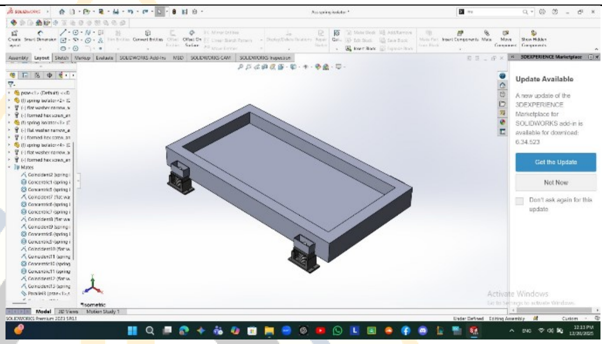
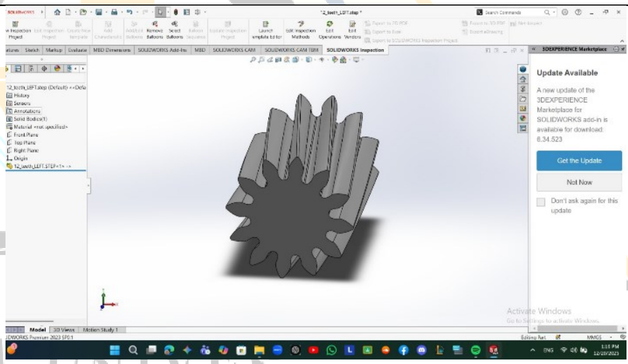
$$V_0 = \frac{284.5}{0.8}$$
$$V_0 \approx 355.6 \text{ cc}$$



Adam simulation for mount



Element	CAD model
clamp	
Enclosure	
Accumulater	
Gear pump	

Cylinder	
Base with isolation mount	
Helical gear	

10.1.1 Recommendation

Component: Standard 0.5 Liter Bladder Accumulator. Placement: Must be mounted immediately at the pump outlet to act as a Helmholtz resonator. Distance from the pump port should be $< \lambda_{fluid}/10$ to be effective.

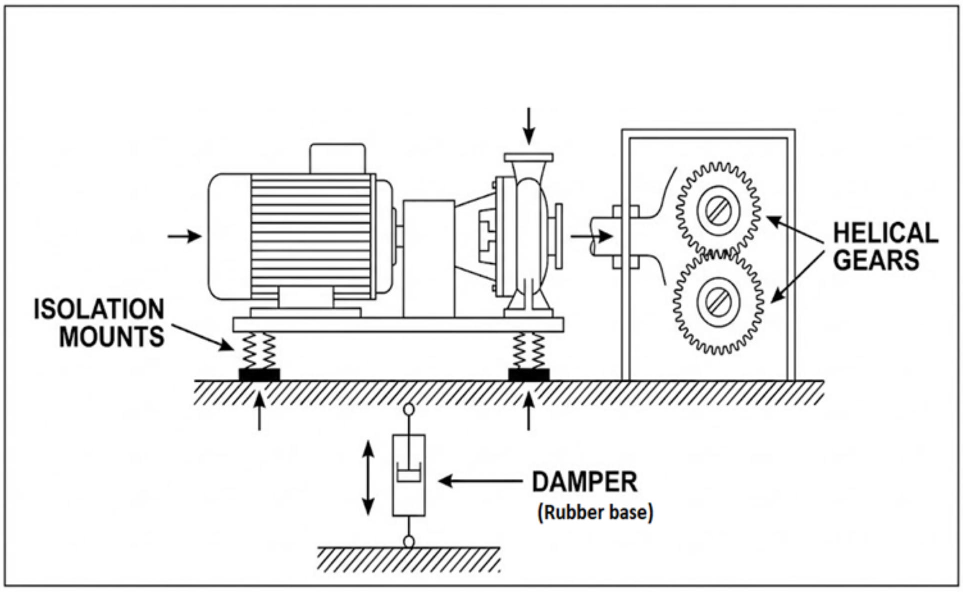


Figure 10.1: Proposed Vibro-Acoustic Mitigation Layout

10.2 Mitigation E: Optimized Piping Layout

10.2.1 Strategy

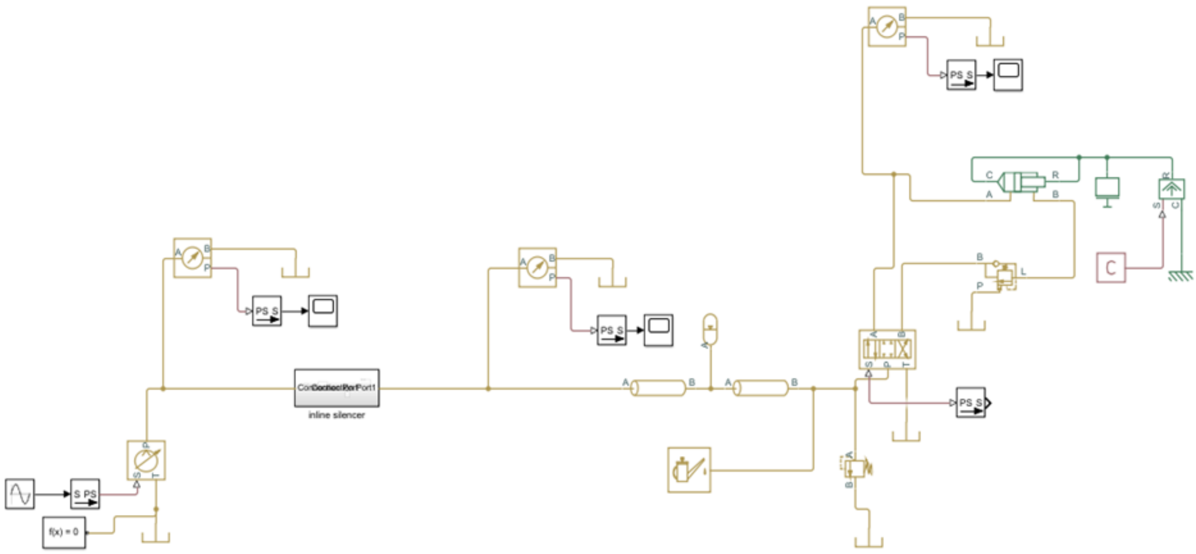
- Resonance Avoidance: Implement the calculated 0.4m clamp spacing to push natural frequencies to $538Hz$.

Limp Mass Effect: A flexible hose section (length $> 500mm$) must be installed immediately after the pump. This hose acts as a vibration break, preventing the pump's mechanical vibrations from coupling into the rigid steel piping network.

Simulation Methodology

To validate these calculations, a numerical simulation workflow is established usi ng MATLAB/Simulink

11.1 MATLAB Simulink system (before and after mitigation)

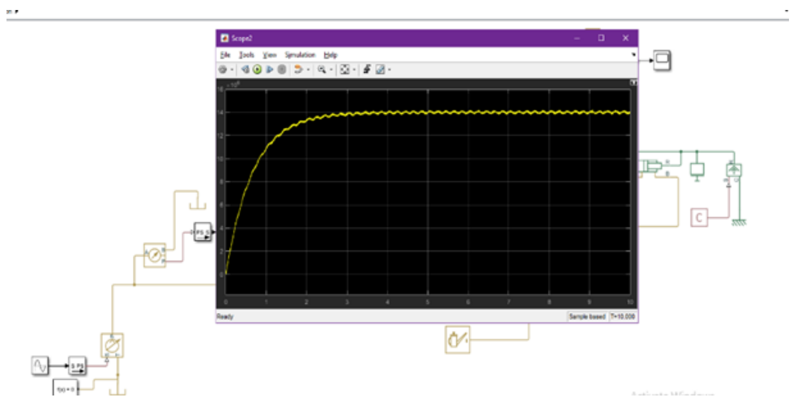


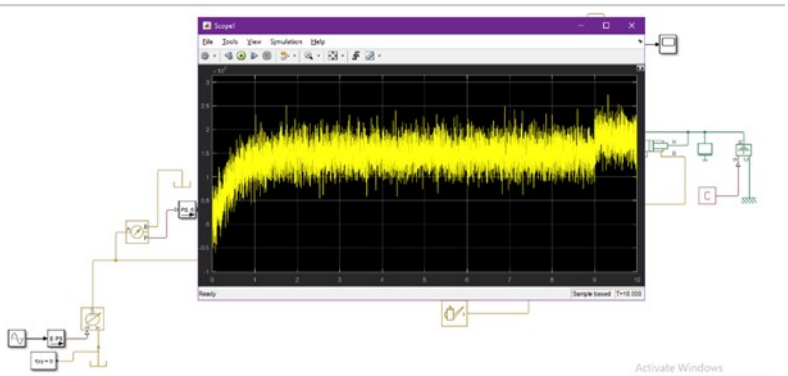
This section outlines the governing equations and MATLAB code structure used to simulate the hydraulic transients.

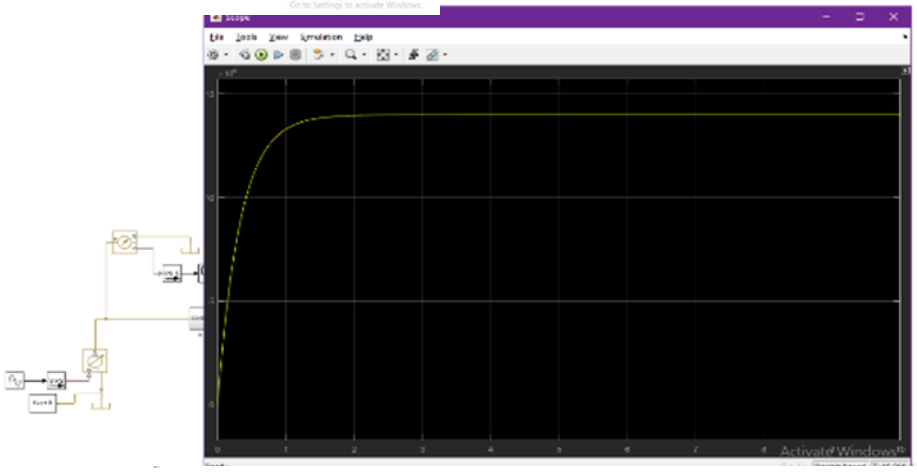
Image 1: The Circuit Model

Description: This image shows the map of the hydraulic system built on a computer. It follows the path of the fluid from left to right:

1. **The Source (Left):** A pump pushes fluid into the system. It is set up to create a lot of vibration and noise (pressure ripples).
2. **The Filter (Middle):** The fluid flows through two "Inline Silencers." These are designed to catch the vibrations and smooth out the flow.
3. **The Destination (Right):** The smoothed-out fluid pushes a hydraulic cylinder (like a piston) to lift or move a load.







Description: These three graphs show how well the filters worked to clean up the pressure signal:

- **Top Graph (Before Filter):** This is the pressure right at the pump. It is very messy and jumps up and down violently. This represents a noisy, damaging system.
- **Middle Graph (After Filter):** This is the pressure just after passing through the silencers. The big jumps are almost gone, showing the noise has been reduced.
- **Bottom Graph (At the Cylinder):** This is the pressure that actually does the work. It is a perfectly smooth line. This proves the silencers successfully removed all the vibration before the fluid reached the moving parts.

Conclusion

This final report synthesizes the mechanical design of a heavy-duty hydraulic bascule bridge with a sophisticated analysis of vibro-acoustic mitigation.

Key Findings:

- The bridge requires a substantial hydraulic power unit delivering 19 kW of mechanical power to lift the 156-ton load against wind resistance.
- The mechanical design of the actuator (180mm rod, 280mm piston) is verified safe against buckling and bursting stresses.
- The vibro-acoustic analysis identified the 300 Hz gear mesh frequency as the primary noise source.
- Standard piping layouts ($0.5 - 1m$ spacing) were found to be prone to resonance. A stricter 0.4m spacing is required.
- The implementation of helical gears, tuned isolation mounts, and a 0.5L accumulator is projected to reduce system noise by 15-25 dB, ensuring compliance with environmental standards.

The integration of robust mechanical design with advanced noise mitigation strate- gies ensures that the proposed bridge system will be not only powerful and reli- able but also environmentally sound and operator-friendly.

Appendix A

Reference Data Tables

Table A.1: Material Properties for Design

Material	Yield Strength (MPa)	E (GPa)	Application
AISI 1045 Cold Drawn	530	205	Piston Rod
AISI 1045 Cold Drawn	530	210	Cylinder Tube
Carbon Steel	250	210	Hydraulic Piping

Table A.2: Hydraulic Oil Properties (HYDRO-TECH HLP 10)

Property	Unit	Value
Density at $15^{\circ}C$	Kg/m^3	854
Flash Point (COC)	$^{\circ}C$	170
Pour Point	$^{\circ}C$	-33
Viscosity at $40^{\circ}C$	mm^2/s	9.7
Viscosity at $100^{\circ}C$	mm^2/s	2.7
Viscosity Index (VI)	-	98

Table A.3: Standard Tube Pressure Ratings

Diameter (mm)	Wall Thickness (mm)	Working Pressure (bar)	Burst Pressure (bar)
38	4.0	220	820
38	5.0	280	1000
42	2.0	84	370
42	3.0	140	560
42	4.0	190	740

50	3.0	120	420
60	3.0	100	340

Table A.4: Hose Selection Chart

Diameter (inch)	Diameter (mm)	Operating Pressure (bar)	Rupture Pressure (bar)
3/8	9.52	350	1400
1/2	12.7	280	1100
3/4	19.1	210	850
1	25.4	210	850

Table A.5: Pump Specifications (Marzocchi ALP3A-D-94)

Parameter	Specification
Type	External Gear Pump
Displacement	61 cm^3/rev
Flow Rate at 1500 rpm	87 L/min
Max Continuous Pressure (P_1)	190 bar
Max Intermittent Pressure (P_2)	205 bar
Max Peak Pressure (P_3)	220 bar
Max Speed	2800 rpm

Table A.6: Vibro-Acoustic Mitigation Targets

Strategy	Target Mechanism	Expected Reduction
Helical Gears	Source Flow Ripple	-6 to -8 dB
Isolation Mounts	Structure-Borne Transmission	-5 to -10 dB
Acoustic Enclosure	Airborne Radiation	-15 to -20 dB
Pulsation Damper	Pressure Ripple Amplitude	~ 95% Reduction